Optimizing Seat and Ball Combination of Actual Gas Lift Valve: An Experimental and
CFD Simulation Study

by

Ehsanul Kabir, M.S.

A Dissertation

In

Petroleum Engineering

Submitted to the Graduate Faculty
of Texas Tech University in
Partial Fulfillment of
the Requirements for
the Degree of

DOCTOR OF PHILOSOPHY

Approved

Dr. Hossein Emadi
Chair of Committee

Dr. Lloyd Heinze

Dr. Talal Gamadi

Dr. Mohamed Soliman

Dr. Mehdi Shahri

Dr. Fathi Elldakli

Mr. Matt Young

Mark Sheridan
Dean of the Graduate School

August, 2018

Copyright 2018, Ehsanul Kabir

ACKNOWLEDGEMENTS

This dissertation is the result of invaluable contributions of many people. I would like to take this opportunity to express my sincere acknowledgements towards them.

First and foremost, I want to thank my advisor Dr. Hossein Emadi. It has been an honor to be his mentee and first Ph.D. student. He has taught me, both consciously and unconsciously, innumerable lessons and insights on the workings of academic research. His technical and editorial advices were essential to the completion of this dissertation. I cannot thank him enough for the long hours he spent in revising the manuscripts for my papers and this dissertation. I appreciate all his contributions of time and ideas to make my Ph.D. experience productive and stimulating.

I would like to thank Dr. Soliman for his constant support throughout my Ph.D. pursuit. He has taught me how to be a prolific researcher. I am also thankful for the excellent examples he has provided as a successful educator. The experience I obtained during the three years I worked as his teaching assistant is going to be paramount in my future career.

I want to thank Dr. Fathi Elldakli for providing a solid foundation for my work. He has always made himself available whenever I needed his assistance. His positive perspective towards research and life will certainly benefit me for the rest of my career.

I am grateful to Mr. Matt Young from Flowco Solutions for his contributions towards this research work. He was kind enough to provide me with the gas lift valves, the seats and the balls that I needed to run the experiments. No doubt, his contributions are going to support this research in the future as well.

My sincere appreciation goes to my Ph.D. dissertation committee members, Dr. Lloyd Heinze, Dr. Talal Gamadi, and Dr. Mehdi Shahri for serving as my dissertation committee members and giving me feedback and suggestions to improve my dissertation.

I would like to thank my family for all their love, support, and encouragement. The enthusiasm of my parents, Z.M.A. Jabbar and Nadira Khan has given me the courage to set forth this demanding voyage. The sacrifices my grandparents made in raising me to become a person that I am today are second to none. Lastly, completing this work would not have

been possible at all without my wife, Nusrat Meharun's faithful and tremendous support throughout all the adversities and happiness.

TABLE OF CONTENTS

ACKNOWLEDGEMENTS.....	ii
TABLE OF CONTENTS	iv
ABSTRACT.....	vii
LIST OF TABLES	ix
LIST OF FIGURES.....	x
CHAPTER 1: INTRODUCTION	1
1.1 Effects of Ball Size	5
1.2 Effects of Port Top Diameter	6
1.3 Effects of Port Bottom Diameter	7
1.4 Effects of Seat Type.....	7
1.5 Research Objectives.....	8
1.6 Research Phases.....	8
1.6.1 Phase 1: Theoretical Calculations.....	8
1.6.2 Phase 2: Experiment.....	8
1.6.3 Phase 3: Simulation.....	8
1.6.4 Phase 3: Selecting the Optimum Combination.....	9
CHAPTER 2: LITERATURE REVIEW.....	10
2.1 Basics of Gas Lift	10
2.1.1 Continuous Flow Gas Lift.....	12
2.1.2 Intermittent Gas Lift.....	13
2.2 Gas Lift Valves.....	14
2.3 Types of Gas Lift Valves	15
2.3.1 Orifice Valves.....	15
2.3.2 PPO Valves (Figure 2-5).....	16

2.3.3 IPO Valves.....	16
2.3.4 IPO Valve Static Force Balance	17
2.3.5 IPO Valve Dynamic Force Balance	19
2.4 Gas Lift Models.....	21
2.5 GLV Dynamic Performance.....	32
2.6 Minimum Stem Travel for Fully Open Valve.....	33
2.6.1 Equations to Calculate Minimum Stem Travel	34
2.6.2 Effects of Ball and Seat Geometries on Minimum Stem Travel	36
2.7 CFD Simulation.....	37
CHAPTER 3: THEORETICAL CALCULATIONS	38
3.1 Case 1: Ball Diameter Larger than the Port Top Diameter.....	38
3.2 Case 2: Ball Diameter Smaller than the Port Top Diameter	39
3.2.1 First Subcase: Ball is Partially Inside the Seat (Figure 3-2).....	41
3.2.2 Second Subcase: Ball is Completely Outside the Seat (Figure 3-3).....	41
CHAPTER 4: EXPERIMENT AND SIMULATION.....	43
4.1 Experimental Procedures	43
4.1.1 Static Testing	43
4.1.2 Aging.....	45
4.1.3 Probe Testing.....	48
4.1.4 Dynamic Testing (Blowdown Test).....	51
4.2 CFD Modeling.....	55
4.2.1 Model Validation	57
4.3 Seats and Balls Used.....	59
CHAPTER 5: RESULTS.....	62
5.1 Optimizing Ball Size.....	62

5.1.1 Result Analysis	73
5.2 Optimizing Port Top Diameter.....	75
5.2.1 Result Analysis	83
5.3 Effects of Port Bottom Diameter.....	84
5.3.1 Result Analysis	90
5.4 Effects of Seat Type.....	91
5.4.1 Result Analysis	98
5.5 Selecting the Optimum Combination of Seat and Ball.....	98
5.6 Improvement over Current Industry Practice.....	100
CHAPTER 6: CONCLUSIONS AND RECOMMENDATIONS.....	102
6.1 Conclusions	102
6.2 Recommendations	103
APPENDIX A: DERIVATIONS	104
A.1 Larger Ball.....	104
A.2 Smaller Ball.....	107
A.2.1 When GLV is closed	107
A.2.2 When GLV is Open.....	110
APPENDIX B: SAMPLE CALCULATIONS.....	115
B.1 Probe Tester.....	115
B.2 Blowdown Tester	118
BIBLIOGRAPHY	123

ABSTRACT

Every Gas Lift Valve (GLV) performs as a variable orifice until port is fully open with maximum stem travel. As the stem travels away from the ball-seat contact point, the area open to flow increases. However, there is a limit to which the stem can travel, and the limiting factor is bellows stacking phenomenon. If bellows stacking takes place before the stem travel required for fully open port area is achieved, then the gas throughput of the GLV and consequently the overall GLV performance decreases.

This is almost always the case when conventional sharp-edge seats are used in GLVs. A modified design for the GLV seats (partly beveled, partly sharp-edge) was present in the literature for a long time which is supposed to enable the stem to travel less compared to sharp-edge seat design for the same flow area generated. However, this design of GLV seat remained surprisingly unexplored until recently when detailed theoretical calculations were made and experiments were run which showed a significant performance improvement compared to sharp-edge seats. Based on this modified design (partly beveled, partly sharp-edged), an optimized design (entirely beveled) was realized using computational fluid dynamics (CFD) simulation which is expected to improve the performance further.

The experiments with modified seat design were run using benchmark valve which excludes the bellows and nitrogen-charged dome sections. This essentially makes the results a little dubious in the context of actual GLV applicability, albeit the results being promising. Furthermore, the American Petroleum Institute (API) recommended practice for GLV stem tip (ball) sizing is to use ball diameters that are 1/16-inch larger than the port diameters of sharp-edge seats. The introduction of the beveled design of the seats facilitates to explore the effects of the ball size on GLV performance.

In this project, comprehensive experimental and simulation studies were conducted to determine the optimum combination of seat and ball. Experiments were conducted using an actual GLV. Geometric models have been deduced to calculate the generated upstream area (frustum area) open to flow for balls that are smaller in diameter than the port top diameter (PTD) of the beveled seats. This frustum area is a function of stem travel, and the

dimensions of the seat and ball. The experiments were run for multiple ball and seat sizes to cover the whole gamut of industrially used GLVs. A numerical model, built based on computational fluid dynamics (CFD), was validated first using experimental results. The difference between experimental and simulation runs for multiple cases was found to be maximum 5%. Finally, results from both simulations and experiments were utilized to determine the optimum seat and ball geometries.

From the actual GLV experiments and the simulations, it was concluded that the optimum PTD of the seat is 2/16-inch larger than the port bottom diameter (PBD) and the optimum ball diameter is 1/16-inch larger than the PBD. It was also concluded that, with the aforementioned optimum combination of the ball and the seat, the entirely beveled seats perform better than both the partially beveled seats and the sharp edge seats. This optimum combination of the ball and the seat resulted in a GLV gas throughput improvement of more than 27% over the currently used design in the industry for 5/16-inch port seat. For larger port seats, this improvement is expected to be even greater. These results, combined with previous results are expected to deliver us with an extensive GLV performance database as well as a complete perception of the effects of seat and ball geometries on GLV performance.

LIST OF TABLES

Table 2-1: Nieberding's Critical Pressure Ratio for Orifice Flow (Camco R-20)	27
Table 2-2: Nieberding's constant for throttling flow for different port sizes (R-20)	29
Table 4-1: Solution methods used in the numerical modeling	56
Table 4-2: Fluid flow model characteristics	57
Table 4-3: Model Validation (modified design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, Ball 5/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)	57
Table 4-4: Model Validation (modified design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, Ball 5/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)	58
Table 4-5: Modified design (MD) seats and corresponding balls used.....	60
Table 4-6: Optimized design (OD) seats and corresponding balls used	61
Table 4-7: Sharp edge (SE) seats and corresponding balls used	61
Table 5-1: Effects of ball size (1/16-inch larger and smaller)	62
Table 5-2: Effects of ball size (ball 2/16-inch larger and smaller)	64
Table 5-3: Mass flowrates from CFD simulations (optimizing ball size)	67
Table 5-4: Effects of port top diameter (ball 1/16-inch smaller)	75
Table 5-5: Effects of port bottom diameter	84
Table 5-6: Effects of seat type	92
Table 5-7: Optimized combinations of seat and ball.....	98
Table 5-8: Current industry practice vs. proposed design	100
Table B-1: Probe tester data.....	117
Table B-2: Blowdown tester fixed data.....	118
Table B-3: Blowdown tester experimental data.....	119

LIST OF FIGURES

Figure 1-1: Basic components of an actual (left) and a benchmark (right) GLV	1
Figure 1-2: Cross-sections of sharp edge (left), modified (center), and optimized (right) seat designs	2
Figure 1-3: Generated frustum area for larger ball	3
Figure 1-4: Bellows stacking phenomenon	4
Figure 1-5: Generated frustum area for smaller ball	6
Figure 1-6: Effect of port top diameter (larger ball)	7
Figure 1-7: Effect of port top diameter (smaller ball)	7
Figure 2-1: Configuration of a typical gas lift well.....	10
Figure 2-2: Flowing pressure gradient traverses above and below the depth of gas injection in a continuous flow gas lift well	13
Figure 2-3: Injection-gas cycle for gas lifting a liquid slug in an intermittent gas lift well	14
Figure 2-4: Major elements of a pressure regulator and a gas lift valve	15
Figure 2-5: PPO valve	16
Figure 2-6: Port area, bellows area, and injection and production pressures	17
Figure 2-7: Effective pressure concept (Kulkarni, 2005)	19
Figure 2-8: Different flow regimes	26
Figure 2-9: Upstream flow area	34
Figure 2-10: The geometric figure for the original derivation (Kulkarni, 2005).....	35
Figure 2-11: Geometric figures used for new equations (sharp-edge seats)	35
Figure 2-12: Same flow area at less stem travel with modified design (Elldakli, 2017) ..	36
Figure 3-1: Ball diameter larger than port top diameter.....	38
Figure 3-2: Ball diameter smaller than the port top diameter and ball is partially inside the seat.....	40
Figure 3-3: Ball diameter smaller than the port top diameter and ball is completely outside the seat	40
Figure 4-1: Tail plug of the GLV	43
Figure 4-2: Dome charging adapter.....	44

Figure 4-3: Stem valve	44
Figure 4-4: Schematic diagram of the ager.....	46
Figure 4-5: Actual ager.....	46
Figure 4-6: Schematic diagram of the probe tester	48
Figure 4-7: Actual probe tester	49
Figure 4-8: Typical data from probe test.....	51
Figure 4-9: Schematic diagram of the blowdown tester.....	52
Figure 4-10: Actual blowdown tester.....	53
Figure 4-11: Sample output of a blowdown test (modified design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, Ball 9/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)	55
Figure 4-12: Components of the experiment that were modeled.....	56
Figure 4-13: Model Validation (modified design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, Ball 5/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)	58
Figure 4-14: Model Validation (modified design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, Ball 5/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)	59
Figure 5-1: Effects of ball size (modified seat design, port top diameter 8/16-inch, ball 1/16-inch larger and smaller).....	63
Figure 5-2: Effects of ball size (optimized seat design, port top diameter 8/16-inch, ball 1/16-inch larger and smaller).....	63
Figure 5-3: Effects of ball size (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball up to 2/16-inch larger and smaller).....	65
Figure 5-4: Effects of ball size (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball up to 2/16-inch larger and smaller).....	66
Figure 5-5: Effects of ball size (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball up to 2/16-inch larger and smaller).....	66
Figure 5-6: Mass flowrates from CFD simulation (optimizing ball size)	67
Figure 5-7: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 5/16-inch).....	68
Figure 5-8: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 6/16-inch).....	68

Figure 5-9: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 8/16-inch).....	69
Figure 5-10: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 9/16-inch).....	69
Figure 5-11: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 5/16-inch).....	70
Figure 5-12: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 6/16-inch).....	70
Figure 5-13: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 7/16-inch).....	71
Figure 5-14: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 9/16-inch).....	71
Figure 5-15: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 6/16-inch).....	72
Figure 5-16: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	72
Figure 5-17: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 9/16-inch).....	73
Figure 5-18: Effects of port top diameter – gas throughput and mass flowrate (ball 1/16-inch smaller).....	76
Figure 5-19: Effects of port top diameter - gas throughput and frustum area (ball 1/16-inch smaller)	77
Figure 5-20: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 5/16-inch, ball 4/16-inch).....	77
Figure 5-21: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 6/16-inch, ball 5/16-inch).....	78
Figure 5-22: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 6/16-inch).....	78
Figure 5-23: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	79

Figure 5-24: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 6/16-inch, ball 5/16-inch).....	79
Figure 5-25: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 6/16-inch).....	80
Figure 5-26: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	80
Figure 5-27: Pressure distribution (modified seat design, port bottom diameter 5/16-inch, port top diameter 7/16-inch, ball 6/16-inch).....	81
Figure 5-28: Pressure distribution (modified seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	81
Figure 5-29: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	82
Figure 5-30: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 9/16-inch, ball 8/16-inch).....	82
Figure 5-31: Hexagonal part of the stem contacting the seat top.....	83
Figure 5-32: Effects of port bottom diameter (modified design seats)	85
Figure 5-33: Effects of port bottom diameter (optimized design seats).....	85
Figure 5-34: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	86
Figure 5-35: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	86
Figure 5-36: Pressure distribution (modified seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	87
Figure 5-37: Pressure distribution (modified seat design, port bottom diameter 6/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	87
Figure 5-38: Pressure distribution (optimized seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	88
Figure 5-39: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	88
Figure 5-40: Pressure distribution (optimized seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch).....	89

Figure 5-41: Pressure distribution (optimized seat design, port bottom diameter 6/16-inch, port top diameter 8/16-inch, ball 7/16-inch)	89
Figure 5-42: Combined effects of port bottom diameter and ball size (modified seat design, port top diameter 8/16-inch)	90
Figure 5-43: Combined effects of port bottom diameter and ball size (optimized seat design, port top diameter 8/16-inch)	91
Figure 5-44: Effects of seat type - gas throughput and mass flowrate (port top diameter 8/16-inch, ball 7/16-inch)	92
Figure 5-45: Effects of seat type - gas throughput and frustum area (port top diameter 8/16-inch, ball 7/16-inch)	93
Figure 5-46: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch)	94
Figure 5-47: Pressure distribution (optimized seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch)	94
Figure 5-48: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch)	95
Figure 5-49: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch)	95
Figure 5-50: Pressure distribution (modified seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch)	96
Figure 5-51: Pressure distribution (optimized seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch)	96
Figure 5-52: Pressure distribution (modified seat design, port bottom diameter 6/16-inch, port top diameter 8/16-inch, ball 7/16-inch)	97
Figure 5-53: Pressure distribution (optimized seat design, port bottom diameter 6/16-inch, port top diameter 8/16-inch, ball 7/16-inch)	97
Figure 5-54: Optimum combinations of seat and ball (gas throughput and mass flowrate)	99
Figure 5-55: Upstream and downstream flow areas of the optimum combinations of seat and ball	100
Figure 6-1: Seat with convex-edged port	103

Figure A-1: Frustum area as a function of stem travel for larger balls	104
Figure A-2: Ball diameter is smaller than port top diameter, ball is completely below the seat top, and the valve is closed	107
Figure A-3: Ball diameter is smaller than port top diameter, ball's top edge is above the seat top, and the valve is closed	107
Figure A-4: Ball diameter is smaller than port top diameter, initially ball was completely below the seat top, and the valve is open (Case 1).....	110
Figure A-5: Ball diameter is smaller than port top diameter, initially ball was partially below seat top, and the valve is open (Case 1).	110
Figure A-6: Ball is smaller than port top diameter, initially ball was partially below seat top, and the valve is open (Case 2).....	112
Figure A-7: Ball diameter is smaller than port top diameter, initially ball was partially below the seat top, and the valve is open (Case 2).....	113
Figure B-1: Probe tester result (modified design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball diameter 9/16-inch, test rack opening pressure 300 psig, injection pressure 450 psig)	115
Figure B-2: Determination of load rate (increasing pressure)	116
Figure B-3: Determination of load rate (decreasing pressure)	116
Figure B-4: Blowdown tester result (modified design, port bottom diameter 3/16"-inch, port top diameter 8/16-inch, ball diameter 9/16-inch, test rack opening pressure 300 psig, injection pressure 450 psig)	118

CHAPTER 1: INTRODUCTION

Artificial lift covers approximately 96% of the US oil well market of which roughly 10% uses gas lift for production (Salinas et al., 2014). Gas lift is a simple and flexible artificial lift method in which external gas is injected continuously or intermittently from the casing-tubing annulus into the tubing string through specially designed valves (Winkler and Blann, 2007). The objective of continuous gas lift is to reduce the flowing bottomhole pressure below the static bottomhole pressure, thereby increasing the inflow of produced fluids. For intermittent gas lift, the injected high-pressure gas periodically displaces liquid from tubing (Lea et al., 2008). Although sophisticated gas lift technologies such as smart GLV (Xu et al., 2013) and advanced gas-lift insert system (GLIS) (Aliyeva and Novruzaliyev, 2015) have been developed, still all GLVs are manufactured based on King's design (King, 1944). This design consists of a nitrogen-charged dome section and bellows assembly. Benchmark GLV is another type of GLV in which the nitrogen-charged dome and bellows assembly are absent. Laboratory testing is the primary use of this type of GLV. However, (Winkler and Camp, 1987) successfully used benchmark valve in field applications. The basic components of both types of GLV are shown in Figure 1-1:

Figure 1-1: Basic components of an actual (left) and a benchmark (right) GLV

For pressure charging and discharging purposes, a dome seal is present on the top of the GLV. A loading element is attached to the dome section (Brown, 1967). A nitrogen-charged bellows assembly is predominantly used as the loading element. However, a spring or a combination of both spring and bellows is also available. The bellows assembly acts very similar to a helical spring and is connected to the stem, which is attached to a ball at its end. Movement of all these sections takes place as a single unit. The ball is seated onto a sized-port when the GLV is closed. In order to prevent the backflow from either the tubing or casing, a check valve is installed on the downstream side of the port.

Three different types of seat designs are shown in Figure 1-2. Sharp edge seat design has been a standard practice in the industry for a long time. However, (Elldakli et al., 2014b) found out that a beveled seat design (modified design) results in as much as 30% improvement of gas throughput capacity of the benchmark GLV depending on the beveled angle with respect to the bottom edge of the seat. Based on the results of the beveled seat design, detail CFD simulations were run by (Elldakli and Soliman, 2017) and the modified seat design was optimized to give the best performance. The optimized design has completely beveled port as opposed to modified design which has partly beveled and partly sharp-edge port.

Figure 1-2: Cross-sections of sharp edge (left), modified (center), and optimized (right) seat designs

The injection pressure operated (IPO) GLV is primarily controlled by the injection pressure. There are two pressures which cause a valve to open: injection or casing pressure and production or tubing pressure. The injection pressure, operating on the effective bellows area minus the port area, produces the first opening force. The second opening force is produced by the production pressure acting on the valve port area. The closing force mainly comes from the nitrogen-charged dome pressure which acts on the effective

bellows area. The stem tip (ball) is also acted upon by a small fraction of the injection pressure which pushes the stem down. Ball surface area governs this fractional amount of closing force. When the total opening force exceeds the total closing force, the GLV initially begins to open. The opening mechanism is incremental. The difference between the opening and closing forces, and the bellows assembly load rate dictates the stem travel in the actual GLV system.

One of the most important parameters in each GLV is the upstream flowing area. Stem movement away from the seat generates this area. A larger upstream flow area is expected to deliver a better gas throughput. The lateral surface area of the frustum of a right circular cone defines the flow area for a partially open valve. This frustum area is generated between the ball surface and the valve seat and increases as the valve stem moves away from the seat (Figure 1-3). Therefore, a GLV is a variable orifice until maximum stem travel and in turns, a fully open port area is attained. Then it becomes equivalent to an orifice (pipe flow). The GLV fully opens when the frustum area becomes equal to the port bottom area and provides maximum gas throughput (Winkler and Camp, 1987). Since a GLV is a flow restriction, coefficient of discharge and flow coefficient also affect the gas throughput. However, as indicated in (Decker, 1993), flow coefficient is the preferred method of quantifying GLV's flow capacity.

Figure 1-3: Generated frustum area for larger ball

In an actual GLV, the bellows assembly performs the most important function by allowing the valve stem tip to move on and off the seat while maintaining the dome-charged pressure (Takacs, 2005). During the gas lift operation, the bellows is frequently exposed to the injection pressure (opening force) and dome-charged pressure (closing force). Therefore, the inner and outer convolutions of the bellows come in contact one another. When the GLV starts to open (the ball starts to move up), metal bellows connected to the stem begins to get compressed because of axial loading. At some point, the bellows assembly might get completely stacked (Figure 1-4). At this point, the ball can no longer move upward, and the frustum area ceases to increase. Therefore, the actual throughput flow in the GL V becomes less than the theoretically calculated value.

Figure 1-4: Bellows stacking phenomenon

A design that can generate a larger flow area with the same stem travel may solve this problem. That is how the modified design of the seats came into realization where a larger flow area with the same stem travel is created by enlarging the top port compared to the bottom port of the seat.

(Elldakli et al., 2014b) tested the modified design with a number of different size seats and balls that are 1/16-inch larger than the port top diameter (PTD). Later, the modified design was optimized based on computation fluid dynamics (CFD) simulation. The effect of larger ball size on GLV performance using sharp-edge seat was also studied (Elldakli et al., 2014a). The results from these tests were promising showing an improvement of 5%-58% in GLV performance. These tests were run using 1.5-inch benchmark valve. The physical difference between an actual GLV and the benchmark valve is the existence of the bellows assembly and nitrogen-charged dome. The benchmark valve is not equipped with a bellows or dome section, but the stem position in relation to the valve seat is manually adjustable. The basic components of a benchmark valve are shown in Figure 1-1.

The information from the benchmark valve could be applied to predict the stem travel for actual GLVs when the volumetric gas rate and upstream and downstream pressures are known. However, the effects ball size, port bottom diameter (PBD), PTD, and seat type (modified and optimized) on the performance of actual GLVs under the presence of bellows and nitrogen-charged dome still remain unexplored.

1.1 Effects of Ball Size

As recommended by (*Gas-Lift Valve Performance Testing API RP 11V2*, 2008), the diameter of the ball should be 1/16-inch larger than the PTD of the seat. When the ball is seated on the port top (ball diameter larger than PTD), the ball/seat contact area, which remains constant, is the major area of the frustum (Figure 1-3). The top plane of the frustum is the minor area, which decreases with an increase in stem travel as the ball moves away from the seat. When the ball is seated below the port top (ball diameter smaller than PTD but larger than PBD), both the major and the minor frustum areas change with the ball's upward movement (Figure 1-5).

Figure 1-5: Generated frustum area for smaller ball

The generated frustum area is controlled by three parameters: (i) the major area of the frustum governed by the PTD of the seat, (ii) the minor area of the frustum governed by the ball size and the PTD of the seat, and (iii) the stem travel. Because of larger size and being seated on top of the seat, larger balls inherently experience a smaller stem travel compared to smaller balls. However, as shown in Figure 1-3 and Figure 1-5, the larger ball might have larger major and minor frustum areas than those of the smaller ball depending on the location of the smaller ball. So, two opposing factors, working at the same time, determine the magnitude of the generated frustum area. Flow coefficient is also expected to have a significant effect on gas throughput. The combining effects of the generated frustum area and the flow coefficient of the ball/seal combination determines the overall gas throughput of the GLV.

1.2 Effects of Port Top Diameter

The PTD of a seat is associated with the ball size. The effects of PTD can be studied for both larger and smaller ball sizes. Increasing the PTD results in an increase in both the major and minor areas of the generated frustum (Figure 1-6 and Figure 1-7). However, when the ball size is larger than then PTD, increasing the PTD results in an increase in ball size. Thus, the effect of larger ball size is also applied. Also, increasing the PTD results in the reduction of the seat wall thickness which in turns adversely affects the sturdiness of the seat.

Figure 1-6: Effect of port top diameter (larger ball)

Figure 1-7: Effect of port top diameter (smaller ball)

1.3 Effects of Port Bottom Diameter

Being smaller than PTD, PBD determines the overall gas throughput of a GLV regardless of PTD and/or ball size. Thus, increase the PBD is expected to increase the gas throughput and vice versa.

1.4 Effects of Seat Type

The modified design of seat has been found to perform better than the sharp-edge design (Elldakli et al., 2014b). The modified design has been optimized to perform even better (Elldakli and Soliman, 2017). The modified design includes an acute change in the port geometry where the beveled and sharp-edge parts of the port meet. For optimized design,

the entire port is beveled. This results in a gradual change in pressure distribution inside the port of an optimized seat. That is why the optimized seats perform better than the equivalent modified seats.

1.5 Research Objectives

- Validate the applicability of the results obtained from benchmark valve to actual GLV regarding the effects of PTD, PBD, and seat type.
- Analyze the effects of ball size on GLV performance.
- Based on the analyses, propose an optimized combination of seat and ball.

1.6 Research Phases

This research consists of four phases as listed below.

1.6.1 Phase 1: Theoretical Calculations

In this phase, geometric models were derived to calculate frustum area as a function of ball and seat geometries, and stem travel for smaller balls. Frustum area equations were also derived for larger balls and compared with existing equations in this stage.

1.6.2 Phase 2: Experiment

This is the experimental phase and consists of three stages. In the first stage, different seats with the same PBD, PTD and seat type (modified and optimized) were tested with ball sizes up to 2/16-inch larger and 2/16-inch smaller than the PTD to determine the optimum ball size. In the second stage, different seats with the same PBD, optimum ball size and seat type were tested with increasing PTD to determine the optimum PTD. In the third stage, different seats with the same optimum ball size and PTD, and seat type were tested with increasing PBD to determine the optimum PBD. In the final stage of this phase, two different seat types with the same optimum PBD, PTD and ball size were tested to verify the optimum seat type.

1.6.3 Phase 3: Simulation

In this phase, the applicability of the existing CFD model was verified with actual GLV experimental results. Then the cases considered in Phase 2 were simulated using the model and the simulation results were compared with experimental results.

1.6.4 Phase 3: Selecting the Optimum Combination

Based on the outcomes of Phase 2 and Phase 3, an optimum combination of ball and seat was proposed in this phase.

CHAPTER 2: LITERATURE REVIEW

2.1 Basics of Gas Lift

Gas lift is a method of artificial lift that uses an external source of high-pressure gas for supplementing formation gas to lift the well fluids. The additional gas augments the formation gas and reduces the flowing bottomhole pressure, thereby increasing the inflow of produced fluids. For dewatering gas wells, the volume of injected gas is designed so that the combined formation and injected gas will be above the critical rate for the wellbore, especially for lower liquid producing gas wells. For higher liquid rates, much of the design procedure may more closely mirror producing oil well gas lift techniques. Parameters such as wellhead and bottomhole pressures, productivity index, and type of produced fluid affect the gas lift system design and consequently the injection pressure.

Figure 2-1: Configuration of a typical gas lift well

Although gas lift may not lower the flowing pressure as much as an optimized pumping system, there are several advantages of a gas lift system that often make gas lift the artificial lift method of choice. For gas wells in particular, when producing a low amount of liquids,

the producing bottomhole pressure with gas lift may compare well with other methods of dewatering. For higher liquid rates, the achievable producing Bottom Hole Pressure (BHP) may be higher than pumping techniques. In fact, there are mainly four categories of wells in which a gas lift system can be considered:

High productivity index (PI), high bottom-hole pressure wells

High PI, low bottom-hole pressure wells

Low PI, high bottom-hole pressure wells

Low PI, low bottom-hole pressure wells

Wells having a PI of 0.50 STB/day/psi or less are classified as low productivity wells. Wells having a PI greater than 0.50 are classified as high productivity wells. High bottom-hole pressures will support a fluid column equal to 70% of the well depth. Low bottom-hole pressures will support a fluid column less than 40% of the well depth.

Of all artificial lift methods, gas lift most closely resembles natural flow and has long been recognized as one of the most versatile artificial lift methods. Because of its versatility, gas lift is a good candidate for removing liquids from gas wells under certain conditions. Crooked/deviated holes present no problem. Well depth is not a limitation. It is also applicable to offshore operations. Lifting costs for a large number of wells are generally very low. However, it requires lift gas within or near the oil fields. It is usually not efficient in lifting small fields with a small number of wells if gas compression equipment is required.

Another significant advantage that gas lift has over other types of artificial lift is its adaptability to changes in reservoir conditions. It is relatively simple to alter a gas lift design to account for reservoir decline or an increase in fluid (water) production that generally occurs in the latter stages of the life of the field. Changes to the gas lift installation can be made from the surface without pulling tubing by replacing the gas lift valves via wireline and reusing the original downhole components.

The primary limitation for gas lift operations is the lack of formation gas or an injection-gas source. Wide well spacing and lack of space for compressors on offshore platforms

may also limit the application of gas lift. Poor compressor maintenance can increase compressor downtime and add to the cost of gas lift gas, especially with small field units. Compressors are expensive and must be properly maintained. Generally, gas lift is not as suitable as some other systems for single-well installations and widely spaced wells. The use of wet gas without dehydration reduces the reliability of gas lift operations.

The two fundamental types of gas lift used in the industry today are continuous flow and intermittent flow. This is the conventional breakdown. However, one could say there are gas lifted gas wells and there are gas lifted oil wells. Gas wells can also be lifted by continuous or intermittent gas lift although most of the gas wells are being lifted by continuous flow.

2.1.1 Continuous Flow Gas Lift

The vast majority of gas lift wells are produced by continuous flow, which is very similar to natural flow. In continuous-flow gas lift, the formation gas is supplemented with additional high-pressure gas from an outside source. Gas is injected continuously into the production conduit at a maximum depth that depends upon the injection-gas pressure and well depth. The injection gas mixes with the produced well fluid and decreases the density and, subsequently, the flowing pressure gradient of the mixture from the point of gas injection to the surface. The decreased flowing pressure gradient reduces the flowing bottomhole pressure below the static bottomhole pressure thereby creating a pressure differential that allows the fluid to flow into the wellbore. Figure 2-2 illustrates this principal.

Figure 2-2: Flowing pressure gradient traverses above and below the depth of gas injection in a continuous flow gas lift well

2.1.2 Intermittent Gas Lift

Intermittent flow is the periodic displacement of liquid from the tubing by the injection of high-pressure gas. When the gas lift valve opens, high pressure injection gas enters the tubing and rapidly expands. This action forces the liquid slug from the tubing. The disadvantage of intermittent-flow gas lift is the “on/off” need for high-pressure gas, which presents a gas-handling problem at the surface and causes surging in the flowing bottomhole pressure that cannot be tolerated in many wells producing sand. Because of the intermittent production of the well, intermittent-flow gas lift is not capable of producing at as high a rate as continuous-flow gas lift. Intermittent flow should not be considered unless the flowing bottomhole pressure is low, and the well is gas lifting from the bottom valve.

Figure 2-3: Injection-gas cycle for gas lifting a liquid slug in an intermittent gas lift well

2.2 Gas Lift Valves

Before the introduction of the single-element, unbalanced, bellows charged gas lift valve in 1944 by King W.F. (King, 1944), there were several unsuccessful attempts were made to lift the oil by using air as the lift gas. This valve revolutionized the gas lift applications and installation design methods. Presently, the unbalanced single-element bellows charged valve remains the most widely used type of gas lift valve for gas lifting wells.

The heart of any gas lift system is the gas lift valve. Because of their functional similarity they are basically downhole pressure regulators. The resemblance between these two devices is apparent in Figure 2-4. In case of gas lift valve, an opening force is combined casing and tubing pressure which exceeds the bellows pressure force. The closing force can be bellows pressure acting on bellows area (A_b) or a spring force or a combination of both.

Figure 2-4: Major elements of a pressure regulator and a gas lift valve

2.3 Types of Gas Lift Valves

Gas lift valves fall into one of three major categories:

1. Orifice valves
2. Production pressure operated (PPO) valves
3. Injection pressure operated (IPO) valves

2.3.1 Orifice Valves

Orifice valves cannot be considered as real valves because they do not open and close. Orifice valves are simply orifices, or holes, providing a communicating port from the casing to the tubing. Because they do not actually function as valves, orifice valves are used only as operating valves to provide the correct injection flow area as required by the valve design and to properly disperse the injected gas to minimize the formation of slugs. Orifice valves typically are used only for continuous flow applications. The valve includes a check to prevent tubing to casing flow.

2.3.2 PPO Valves (Figure 2-5)

Production pressure operated valves (sometimes called tubing pressure valves) are operated primarily by changes in pressure of the production fluid. Unloading is then controlled primarily by the reduction in hydrostatic pressure in the production stream by injecting lift gas. PPO valves are used typically:

- Where the production fluid is produced through the annulus.
- In dual completions where two gas lift systems are installed in the same well to produce two differently pressured zones.
- For intermittent lift.

PPO valves are ideal for intermittent lift applications since the valve is designed to remain closed until a sufficient fluid load is present in the tubing, at which time the valve opens, producing the liquid.

Figure 2-5: PPO valve

2.3.3 IPO Valves

An injection pressure operated gas lift valve is mostly sensitive to changes in casing pressure. Opening and closing of a valve is controlled by varying casing pressure and to a

lesser degree, the tubing pressure effect. The injection pressure plays an important role in successfully unloading the well. A pressure operated valve is shown in Figure 1-1.

IPO valves have one or more entrance ports through which injection gas can enter the valve chamber from either the casing or tubing. The chamber contains the bellows. The dome above the bellows contains nitrogen gas under pressure which is put in the valve when the bellows is charged in a controlled condition shop.

The valve stem extends to the ball which seals on a seat (port). Fluid pressure is exerted against the ball on the seat from below. Injection gas pressure, exerted on the outside of the bellows, and production or fluid pressure on the port area, causes the valve to open by lifting the ball off of the seat allowing injection gas to enter the production fluids through the port, the check valve and into the fluid stream. The bellows and attached stem lowers onto the seat to close the port.

2.3.4 IPO Valve Static Force Balance

Figure 2-6: Port area, bellows area, and injection and production pressures

The opening force is a combined pressure force of casing and tubing pressure. Injection pressure acts on the effective port area ($A_b - A_p$) while production pressure acts on port area (A_p). Therefore, the opening force can be expressed as,

$$P_i \times (A_b - A_p) + P_p \times A_p \quad (2-1)$$

Here,

A_p = Port area

A_b = Total bellows area

P_i = Injection pressure

P_p = Production pressure

The closing force for most injection pressure operated gas lift valves is obtained from a pressure charged bellows. Some valves use a spring with an uncharged bellows to produce the closing force. Therefore, the closing force can be written as,

$$P_{bt} \times A_b \quad (2-2)$$

Here,

P_{bt} = Nitrogen-charged dome pressure

Basic force balance can be expressed by equating these two forces.

$$P_{bt} \times A_b = P_i \times (A_b - A_p) + P_p \times A_p \quad (2-3)$$

If the valve is open then,

If $P_{bt} \times A_b < P_i \times A_p$Stays Open

If $P_{bt} \times A_b > P_i \times A_p$Will close

And if the valve is closed then,

If $P_{bt} \times A_b < P_i \times (A_b - A_p) + P_p \times A_p$ Will reopen

If $P_{bt} \times A_b > P_i \times (A_b - A_p) + P_p \times A_p$ Stays closed

So, the valve closing pressure is P_{bt} (dome pressure) and the opening pressure is

$$P_i = \frac{P_{bt} - P_p \times R}{1 - R} \quad (2-4)$$

Where,

$$R = \frac{A_p}{A_b} \quad (2-5)$$

The valve spread is defined as,

$$P_i - P_{bt} = \frac{R(P_{bt} - P_p)}{1 - R} \quad (2-6)$$

2.3.5 IPO Valve Dynamic Force Balance

One assumption of the static force balance is a closed valve pops open when opening pressure exceeds closing pressure. This means at the instance of opening the entire flow area becomes available for gas passage, which is clearly not the actual case. Because of this inherent limitation of static force balance, a dynamic force balance equation is required to determine valve's stem position while open and flowing. The pressure acting on the valve stem is not simply injection or production pressure. Various methods are available in literature to determine that "effective pressure" acting on stem. A simple assumption that the ball sees only tubing pressure when ball is on seat or stem travel = 0 inch and casing pressure acts on ball when maximum stem travel occurs. This is illustrated in Figure 2-7. The forces acting on the ball will be upward or opening forces and closing or downward forces.

Figure 2-7: Effective pressure concept (Kulkarni, 2005)

Downward forces acting on stem tip are:

- Load Rate acting on Bellows area
- P_{bt} acting on Bellows area

Upward forces acting on stem tip are:

- Casing pressure acting on (Bellows area - Port area)
- Tubing pressure acting on fraction of Port area
- Casing pressure acting on fraction of Port area

Balancing these forces, we get

$$\begin{aligned} LR \times y \times A_b + P_{bt} \times A_b \\ = P_c \times (A_b - A_p) + P_t \times f \times A_p \\ + P_c \times (1 - f) \times A_p \end{aligned} \quad (2-7)$$

Here,

LR = Load rate

y = Stem travel

A_b = Total bellows area

P_{bt} = Nitrogen-charged dome pressure

A_p = Port area

P_t = Tubing pressure

P_c = Casing pressure

$$f = 1 - \frac{y}{y_{max}}$$

y_{max} = (Multiplying factor) $\times$ $y_{fully\ open}$

$y_{fully\ open}$ = Minimum stem travel to fully open the GLV

Case 1 ($y < y_{max}$)

$$\begin{aligned}
LR \times y \times A_b + P_{bt} \times A_b \\
&= P_c \times (A_b - A_p) + P_t \times \left(1 - \frac{y}{y_{max}}\right) \times A_p + P_c \times \left(\frac{y}{y_{max}}\right) \times A_p \\
y &= \frac{P_c \times (1 - R) + P_t \times R - P_{bt}}{LR + \frac{R \times (P_t - P_c)}{y_{max}}} \quad (2-8)
\end{aligned}$$

Case 2 ($y > y_{max}$)

$$y = \frac{P_c \times A_b - P_{bt} \times A_b}{LR} \quad (2-9)$$

2.4 Gas Lift Models

Various attempts were made to predict flow rate performance of gas lift. Thanks to researchers like Winkler, Decker K.L., Neely A.B, Decker L.A. and associations like TUALP, API. Because of them, valve testing and modeling has advanced. But unfortunately, after so much of discussion and deliberation on this topic, the question remains the same. What is the actual gas lift valve performance? Some of the researchers treated the gas lift valve as an orifice and used Thornhill-Craver equation to calculate the gas throughput while others treated gas lift valve as a convergent-divergent nozzle or a control valve and calculated the gas throughput accordingly.

(Montgomery et al., 1974) showed that gas lift valve performance is complex. They also indicated that valve should be modeled as variable orifice venturi in which the flow area is controlled as a complicated function of both casing and tubing pressure.

(Decker, 1976) made the first attempt to analytically model the bellows operated gas lift valve performance. He modeled the pressure response by considering thermodynamic and mechanical factor. He solved the force balance equation considering these two factors.

$$\int_0^x \left(\frac{d\bar{P}}{dx}\right) dx = \frac{K_s}{A_b} \int_0^x dx + \int_0^x \left(\frac{dP_D}{dx}\right) dx + \int_0^x \left(\frac{d\bar{P}}{dx}\right)_{fraction} dx \quad (2-10)$$

Where,

$d\bar{P}$ = Incremental change in upstream pressure, psig

dx = Incremental stem travel, inch

K_s = Spring rate, lb_f/inch

A_b = Total bellows area, inch²

$d\bar{P}_D$ = Incremental change in downstream pressure, psig

Earlier models used Thornhill-Craver equation without accounting for variable area and gas compression in actual gas lift valve operation. (Winkler and Eads, 1993) were the first ones to notice the variable area to flow effect. They treated the valve as a variable orifice backpressure regulator and modified the Thornhill-Craver equation for variable area and gas compressibility. Below is the Thornhill-Craver equation for sharp-edge seats:

$$C_d = \frac{Q_{gi} \times \sqrt{Z \times SG_g \times T}}{155.5 \times A_{eff} \times P_{iod} \times \sqrt{2 \times g \times \left(\frac{K}{K-1}\right) \times \left(r^{\frac{2}{K}} - r^{\frac{K+1}{K}}\right)}} \quad (2-11)$$

Where,

Q_{gi} = Volumetric gas flowrate, MSCF/D

A_{eff} = Effective orifice area open to flow, in²

C_d = Discharge coefficient

P_{iod} = Upstream pressure, psig

g = Gravitational acceleration, 32.174 ft/sec²

$K = C_p/C_v$

$r = P_{pd}/P_{iod}$

P_{pd} = Downstream pressure, psig

SG_g = Specific gravity of gas at valve (air=1)

T = Upstream temperature at the inlet of the valve, °R

Z = Compressibility factor at valve conditions

(Winkler and Camp, 1987) also suggested a criterion for testing valves. They presented new discharge coefficients obtained from testing a benchmark valve. They also pointed out that a simple mathematical model to predict gas lift valves is unlikely because of many factors affecting valve performance, including the effect of the operating pressure level on bellows assembly load rate and linear stem travel for many valves.

(Decker, 1993) wrote an excellent paper describing the method for conducting gas lift valve performance test. He proved that the use of static force balance in determining stem travel and opening and closing pressure can lead to erroneous results. He also pointed out the use of C_d as a single number in Thornhill-Craver equation is unable to account for changes in flow capacity. He used flow coefficient (C_v) concept instead of C_d and presented flow coefficients as a function of stem travel based on ISA (International Standard on Auditing) testing procedure. His calculation procedure can be summarized as follows:

Step 1:

Solve static force balance equation for stem travel for subject valve at the operating pressure and temperature conditions.

$$dx_s = \frac{0.193 \times (P_{tf} - P_{vct}) + 0.037 \times P_{cf}}{k} \quad (2-12)$$

Step 2:

Find valve closing pressure, valves pressure spread and the pressure ratio correlation for the operating conditions.

$$P_{vc1} = P_{vct} - 0.192 \times P_{cf} \quad (2-13)$$

$$P_s = P_{cf} - P_{vc1} \quad (2-14)$$

$$P_r = \frac{dP}{P_s} \quad (2-15)$$

Step 3:

Compute the modified travel ratio (T_m) as a function of the pressure ratio.

$$T_{rm} = C_1 \times P_r + C_2 \text{ for } P_r, P_{rmax} \quad (2-16)$$

$$T_{rm} = C_3 \times P_r^2 + C_4 \times P_r + C_5 \text{ for } P_r, P_{rmax} \quad (2-17)$$

Step 4:

Compute the dynamic stem travel as follows:

$$dx_d = \frac{T_{rm} \times dx_s}{\left(\frac{P_s}{P_{vct}}\right)^{0.5}} \quad (2-18)$$

Step 5:

Compute the flow coefficient and critical pressure ratio using dynamic stem travel.

$$C_v = C_1 \times dx_d^3 + C_2 \times dx_d^2 + C_3 \times dx_d + C_4 \quad (2-19)$$

$$X_t = C_5 \times dx_d^3 + C_6 \times dx_d^2 + C_7 \times dx_d + C_8 \quad (2-20)$$

Step 6:

Flow rate for given operating conditions and for specific valve and choke size can be calculated as follows:

$$Q_{gi}(MSCF/D) = 32.64 \times C_v \times y \times P_{cf} \times \sqrt{\frac{X'}{T_{inj} \times S_g \times Z_{inj}}} \quad (2-21)$$

Because of the industry requests, API published “Recommended Practice” on gas lift valve performance testing in March 2001 (*Gas-Lift Valve Performance Testing API RP 11V2*, 2008) which was reaffirmed in April 2008. API, in their simplified model approach followed the same test procedure as Decker K.L. stated in his paper (Decker, 1993). Instead of dynamic stem travel they used a static force balance for the calculation of the stem travel. The API procedure can be outlined as follows:

Step 1:

Determine the stem position using following equation:

$$P_{vct} \times A_b + B_{lr} \times A_b \times dx = P_{iod} \times (A_b - A_s) + P_{pd} \times A_s$$

$$dx = \frac{P_{iod} \times (A_b - A_s) + P_{pd} \times A_s - P_{vct} \times A_b}{B_{lr} \times A_b} \quad (2-22)$$

Step 2:

Determine C_v and X_t for a given valve and given operating conditions:

$$C_v = 219.3 \times dx^3 - 149.04 \times dx^2 + 22.58 \times dx \quad (2-23)$$

$$X_t = 734.4 \times dx^3 - 178.3 \times dx^2 + 15.12 \times dx \quad (2-24)$$

Step 3:

Compute the flow rate using the following equation:

$$Q_{gi}(MSCF/D) = 32.64 \times C_v \times y \times (P_{iod} + 14.7) \times \sqrt{\frac{X}{(T_v + 460) \times S_g \times Z}} \quad (2-25)$$

(*Gas-Lift Valve Performance Testing API RP 11V2*, 2008) and Decker's method (Decker, 1993) relies on determining the various constants e.g. C_v , X_t as per defined test procedure for each individual valve.

An industry sponsored research program at Tulsa University (TUALP) started in 1983 to investigate this area of gas lift. The various researchers approach this problem differently as some of them developed unified model to determine the gas passage through valve ((Bertovic, 1995), (Nuenz, 1994), and (Faustinelli, 1997)). Some of them developed orifice and throttling flow models ((Nieberding, 1988), (Hepguler, 1988), (Acuna, 1989) and (Sagar, 1991)). TUALP researchers distinguished the flow performance family in orifice, transition and throttling flow. A typical plot is shown in the following Figure 2-8.

Figure 2-8: Different flow regimes

(Biglarbigi, 1985) was the first TUALP researcher to complete the thesis on gas lift valve performance. He analyzed the flow performance of nitrogen and spring-loaded valves simulating downhole conditions by adapting constant injection pressure test (CIPT). He developed statistical model for predicting valve performance based on tests conducted on Camco R-20 valve. His thesis emphasized on developing the facilities and procedures for valve testing. Unfortunately, an error in the static test facility made all his test data invalid.

(Nieberding, 1988) continued Biglarbigi's work (Biglarbigi, 1985). He modified the static test facility by correcting the error. He developed statistic models for throttling and orifice flow based on test data of Camco R-20 valve. He suggested a criterion based on test rack opening pressure to differentiate between throttling and orifice flow. When injection pressure is above the test rack opening pressure, P_{tro} , the valve will behave like an orifice and if injection pressure is lower than the test rack opening pressure, P_{tro} , but larger than valve closing pressure, P_{vc} the valve will behave like a throttling valve. For orifice flow he developed following correlation to predict gas passage.

$$Q_{gs} = 1240.3 \times A_p \times C_d \times y \times \sqrt{\frac{P_{inj} \times (P_{inj} - P_{pd})}{(1 - \beta^4) \times T_{inj} \times Z_{inj} \times \gamma_g}} \quad (2-26)$$

Where,

Q_{gs} = Volumetric gas flowrate, MSCF/D

A_p = Port area, in²

C_d = Discharge coefficient

y = Gas expansion factor

P_{inj} = Upstream pressure, psig

P_{pd} = Downstream pressure, psig

β = Square root of ratio of port to inlet area

Nieberding's empirical values for constants are tabulated in the following table.

Table 2-1: Nieberding's Critical Pressure Ratio for Orifice Flow (Camco R-20)

Port size (in ²)	R_{crit}
0.1875	0.63
0.2500	0.64
0.3125	0.60
0.3750	0.58
0.4375	0.45
0.5000	0.46

The discharge coefficient was calculated based on following regression correlation.

$$C_d \times y = M \times \frac{P_{inj} - P_{pd}}{k \times P_{inj}} + B \quad (2-27)$$

$$M = 2.03 \times d_p - 1.28 \quad (2-28)$$

$$B = -1.54 \times d_p + 1.24 \quad (2-29)$$

Based on test rack opening pressure, he developed following equation to describe throttling flow performance.

$$Q_{gs} = Q_{gs,max} \times \frac{N \times (1 - N)}{N_{max}^2 - N \times (2 \times N_{max} - 1)} \quad (2-30)$$

$$Q_{gs} = Q_{gs,max} \times \frac{N \times (1 - N)}{N_{max}^2 - N \times (2 \times N_{max} - 1)} \quad (2-31)$$

$$N_{max} = \frac{P_{pd,max} - P_{pdc}}{P_{inj} - P_{pdc}} \quad (2-32)$$

$$P_{pdc} = M_1 \times P_{pdcfb} + B_1 \quad (2-33)$$

$$P_{pdcfb} = \frac{P_{vct} \times A_b - P_{inj} \times (A_b - A_p)}{A_p} \quad (2-34)$$

$$Q_{gs,max} = a \times d_p^b \times P_{inj}^c \times (P_{inj} - P_{d,max})^d \times T_{inj}^e \times \gamma_g^f \quad (2-35)$$

Where,

$Q_{gs,max}$ = The maximum gas flow rate on throttling flow curve, MSCF/D

N = Dimensionless downstream production pressure

N_{max} = Dimensionless downstream production pressure at maximum flow rate

$P_{d,max}$ = Downstream production pressure at maximum flow rate, psig

P_{pdc} = Production closing pressure, psig

P_{pdcfb} = Production closing pressure according to force balance, psig

d_p = Diameter of port, inch

A_p = Port area, in²

A_b = Bellows area, in²

γ_g = Gas specific gravity

T_{inj} = Upstream gas temperature, °F

Table 2-2: Nieberding's constant for throttling flow for different port sizes (R-20)

Constant	0.1875"	0.25"	0.3125"	0.375"	0.4375"	0.5"
M1	0.64	0.66	0.87	1.09	1.15	1.23
B1	347	276	157	-16	-20	-23
Nmax	0.678	0.665	0.639	0.476	0.400	0.259
a	0.65					
b	0.26					
c	0.26					
d	1.32					
e	0.21					
f	0.32					

Nieberding also tested the valves in methane to make sure that the same test procedure works other than air as anticipated.

(Hepguler, 1988) also completed his thesis in same year. He pursued a different approach. He adapted the test facility of (Biglarbigi, 1985) to measure the internal parameters of the valve needed to develop a mechanistic model. His goal was to significantly reduce the number of tests that must be made to fully characterize a valve. Hepgular measured the internal pressures and temperature during dynamic testing and stem displacements and the load rate in static tests. He observed pressure drop through valve when gas flow rate exceeded 1.4 MMSCF/D. He developed a mechanistic model for valve performance. Unfortunately, the model was not as accurate as the empirical testing process.

(Acuna, 1989) performed tests on three 1-inch gas lift valves and developed a statistical model by normalizing the valve data using Nieberding's orifice flow model (Nieberding, 1988) rather than a dynamic model. He also developed second model on the assumption that throttling flow region of the curve was a linear function of valve closing pressure. The Acuna's final form of equation is,

If $P_{pd} < P_{pd,max}$, then

$$Q_{gi} = Slope \times (P_{pd} - P_{pdc}) \quad (2-36)$$

$$Slope = m \times P_{vct} + b \quad (2-37)$$

Where, m and b are statistically determined coefficients.

If $P_{pd} > P_{pd,max}$, then

$$Q_{gi} = (Correction\ Factor) \times Q_{g,orifice} \quad (2-38)$$

$$Correction\ Factor = \frac{Q_{gs,max}(Straight\ Line\ Throttling\ Model)}{Q_{gs,max}(TUALP\ Orifice\ Model)} \quad (2-39)$$

Where, correction factor is a function of the maximum flow rate in throttling and orifice flow.

(Sagar, 1991) performed comprehensive theoretical and experimental study to develop an improved mechanistic model of a pressure operated nitrogen gas lift valve. He conducted experiments on Merla-made N-15R valve and developed empirical equations quantifying bellows behavior.

$$LR = 0.3124 \times P_{vc} + 82.49 \quad (2-40)$$

Where, LR = Load rate psi/inch

$$y_{s,max} = \frac{P_{inj} - P_{vc} - 5.8}{0.3124 \times P_{vc} + 82.49} \quad (2-41)$$

Where,

P_{inj} = Upstream gas injection pressure, psig

P_{vc} = Valve closing pressure, psig

He used a Quasi-one-dimensional flow theory for a convergent-divergent nozzle to model and described the complex high velocity gas flow through the ball seat configuration. He also observed that pressure operated valve is sensitive to performance parameters. A small variation of 1% in injection pressure and temperature can change the maximum flow rate by as much as 30%.

(Rodriguez, 1992) extended the models of both (Nieberding, 1988) and (Acuna, 1989) by comparing their application to several pressure operated gas-lift valves. He acquired the performance data on those valves and compared the two models. Rodriguez empirically proved that the constant injection pressure test (CIPT) and constant production pressure test (CPPT) methods are equivalent for valve performance testing. His equation was:

$$Q_{gs} = 1240.3 \times A_p \times C_d \times y \times \sqrt{\frac{P_{iod} \times (P_{iod} - P_{pd})}{T_{inj} \times Z_{inj} \times \gamma_g}} \quad (2-42)$$

He used $P_{inj} = P_{vOT}$ as the transition boundary and modified Nieberding's orifice model by implementing the term $(1/(1-\beta^4))^{0.5}$ in $C_d \times y$.

(Nuenz, 1994) first attempted a unified model which treated the both, orifice and throttling flow performance by a single equation. She analyzed seven different valves for performance testing. She showed linear relationship between $C_d \times Y \times A_f$ and the valve's throttling behavior. She defined the ratio A_{eff}/A_p as effective area factor, A_f . For orifice flow, she used $A_f = 1$ and for throttling flow she introduced the term, dimensionless stem position, N_s , which is sensitive to changes in valves closing pressure. The final form of gas passage equation was:

$$Q_{gs} = 1240.3 \times A_p \times C_d \times y \times A_f \sqrt{\frac{P_{iod} \times (P_{iod} - P_{pd})}{T_{inj} \times Z_{inj} \times \gamma_g}} \quad (2-43)$$

(Bertovic, 1995) conducted an experimental and theoretical study of the flow performance for the McMurry JR-STDN 1-inch valve. His unified model described all three flow regimes, orifice, throttling and transition. His model used six coefficients to predict all three flow regimes. However, he incorrectly used the theoretical adiabatic equation from the minimum area into expansion area.

(Faustinelli, 1997) also developed a unified model for all three flow regimes for gas lift valve performance. Compared to previous models, his model predicts volumetric flow rates more accurately. Also, his model required only four empirical parameters to be determined to calculate gas passage through valve. In addition, a numerical model for the influence of production and fluid temperature on bellows temperature was developed, which can predict the dome temperature by following equation,

$$T_d \cong 0.3 \times T_{pd} + 0.7 \times T_{inj} \quad (2-44)$$

Where,

T_d = Valve dome temperature, °F

T_{pd} = Production fluid temperature, °F

T_{inj} = Injection gas temperature, °F

Both, Bertovic's (Bertovic, 1995) and Faustinelli's (Faustinelli, 1997) models assumed one-dimensional flow, which may fail to predict transition flow. They assumed that the ball-seat area is the function of only the dimensionless stem position. Also, their coefficients correspondent to only three different valve closing pressure, which results in their models being unusable for other valve closing pressures.

2.5 GLV Dynamic Performance

For a long time, GLVs were considered as fixed orifice and the Thornhill-Craver (T-C) equation (Beggs, 1984) was used to calculate the gas flow rate through GLVs. The original T-C equation (Cook and Dotterweich, 1946) was developed for 6-inch bean choke and was not meant to be used for GLVs. A GLV acts as a fixed orifice only when the valve is fully open which rarely happens. Consequently, using the T-C equation would produce erroneous results as indicated by (Decker, 2008) and (Almeida, 2011).

One of the first approaches to investigate the orifice nature of the GLV was by (Montgomery et al., 1974). In this study, the GLV was identified as a venturi with variable orifice. This research was further advanced by TUALP (Tulsa University Artificial Lift Project). Several papers related to the modeling of dynamic nature of GLV were published under TUALP between 1992 and 1993. Finally, in 2001, the API Recommended Practice 11V2 (*Gas-Lift Valve Performance Testing API RP 11V2*, 2008) was introduced combining the results of TUALP publications to provide a complete method for dynamic testing of GLVs. The API method is not included in this paper to avoid superfluity. A brief description of this method can also be found in (Takacs, 2005).

In this study, a pressure decay method, which is also called the "Blowdown test" is used to determine the GLV dynamic performance. The objective of this method is to rapidly measure the flow rate capabilities of the valves. As indicated in (Shahri, 2011), this method does not substitute the current method mentioned in the API Recommended Practice 11V2 (*Gas-Lift Valve Performance Testing API RP 11V2*, 2008) and is recommended where the flow regime of gas injection is critical or supersonic. The advantage of using blowdown test as the dynamic testing is its rapidity. Since the objective of this work requires

significant number of flow tests, the blowdown test method was chosen. The validity of this method was verified by (Shahri and Winkler, 2011) and (Kulkarni, 2005). The detail methodology is explained in Section 4.1.4.

2.6 Minimum Stem Travel for Fully Open Valve

The minimum stem travel required to fully open an unbalanced, single-element gas lift valve (GLV) is an important parameter in GLV performance calculations. The value of this parameter dictates whether the valve is fully or partially opened, thus determining the amount of gas passage through the GLV. The geometries of the ball and the seat are the main factors affecting this parameter.

The flow area for a partially open valve is defined by the lateral surface area of the frustum of a right circular cone (Figure 2-9). This frustum area is generated between the ball surface and the valve seat and increases as the valve stem moves away from the seat. The major area of the frustum is the ball/seat contact area, which remains constant. The minor area, which is the top plane of the frustum, decreases with an increase in stem travel as the ball moves away from its seat. When the frustum area becomes equal to the port area, which is constant for a sharp-edge seat and variable for a beveled seat, the GLV fully opens and provides maximum gas throughput. However, it is recommended to have a stem travel that allows at least 125% fully open GLV in order to eliminate the effect of the presence of the ball (Shahri, 2011).

Figure 2-9: Upstream flow area

2.6.1 Equations to Calculate Minimum Stem Travel

The following equation was given by (Kulkarni, 2005) to calculate the minimum stem travel required for a fully open GLV based on sharp-edge seat geometry.

$$y = r_p \times \tan \left\{ \cos^{-1} \left(\frac{-r_p^2 + \sqrt{r_p^4 + 4 \times r_b^2 \times r_p^2}}{2 \times r_b^2} \right) \right\} - \sqrt{r_b^2 - r_p^2} \quad (2-45)$$

This equation was modified by (Elldakli et al., 2014b) to fit the beveled seat design. However, only the port top diameter was considered when comparing the port area to the frustum area.

$$y = r_{pt} \times \tan \left\{ \cos^{-1} \left(\frac{-r_{pb}^2 + \sqrt{r_{pb}^4 + 4 \times r_b^2 \times r_{pt}^2}}{2 \times r_b^2} \right) \right\} - \sqrt{r_b^2 - r_{pt}^2} \quad (2-46)$$

The geometric figure used to derive the original equation is presented in Figure 2-10.

Figure 2-10: The geometric figure for the original derivation (Kulkarni, 2005)

However, the issue with these equations is these are based on only one position of the ball. This means that this particular figure might not be applicable for other combinations of position of the ball and/or geometry of the seat. This is illustrated in Figure 2-11.

Figure 2-11: Geometric figures used for new equations (sharp-edge seats)

In Figure 2-11, two different positions of the ball are shown. The left one shows that the center of the ball's changed position is still under the area of the balls initial position. In the right one, the center of the ball's changed position is outside of the area of the ball's initial position. If Figure 2-10 is used to calculate stem travel, then according to Figure

2-11 which is drawn to scale, the stem travel would be calculated as AF whereas the actual stem travel is AL (change in the position of the ball's center, marked in yellow). So, for the first case, the calculated stem travel would be an overestimation and for the second case, an underestimation.

Also, Equation (2-45) is only applicable for sharp-edge seat. Although Equation (2-45) was modified to give Equation (2-46) to fit the beveled seat geometry, only the port top area of the beveled seat was considered. As opposed to the sharp-edge seat design, the port area is not constant in beveled seat. So, in the development of the respective equation, the average port area should be considered rather than just the port top area.

2.6.2 Effects of Ball and Seat Geometries on Minimum Stem Travel

(Elldakli, 2017) investigated the effects of ball and seat geometries on the minimum stem travel and also on GLV gas throughput using benchmark valve and CFD. He found out that with increasing PTD, the minimum stem travel required to achieve a fully open valve, decreases (Figure 2-12).

Figure 2-12: Same flow area at less stem travel with modified design (Elldakli, 2017)

He also found out that GLV gas throughput increases with increasing PBD along with 1/16-inch larger ball sizes, and with ball sizes even larger. Based on these findings, he optimized the port geometry of the seat which is illustrated in Figure 1-2.

2.7 CFD Simulation

CFD uses numerical analysis, data structure, and computational software to solve, visualize, and analyze problems that involve fluid flows. CFD modeling is based mainly on three governing equations: conservation of mass, conservation of momentum, and conservation of energy. These equations are simultaneously solved to predict fluid flow behaviors for defined set of boundary conditions. The entire process is divided into three steps:

- (i) Pre-processing: In this step, the geometry is created and cleaned up to define the domain of interest. The entire domain is then divided into small segments known as mesh or grids.
- (ii) Solver: The boundary conditions are specified in this step. The numerical methods to be used are defined, and additional models required by physics are also included in this step.
- (iii) Post-processing: This step involves results generation like contour and vector plots. Average values are calculated, and report is generated in this step.

The application of CFD technique to characterize GLV is somewhat limited in the industry. (Turzo and Takacs, 2009) studied the GLV performance by developing a numerical model. They validated the model using the (*Gas-Lift Valve Performance Testing API RP11V2*, 1995) specifications. The model was based on five governing equations: the Navier-Stokes equation, the continuity equation, the equation for conservation of energy, the equation of state for the flowing fluid, and the equation describing the change in enthalpy. According to the model, a high-pressure zone is evident at the stem tip, which was not explained by the authors.

CHAPTER 3: THEORETICAL CALCULATIONS

Geometric models have been derived to calculate the frustum area as a function of stem travel. There can be two cases for calculation of frustum area as a function of stem travel:

Case 1: Ball diameter larger than the port top diameter

Case 2: Ball diameter smaller than the port top diameter

3.1 Case 1: Ball Diameter Larger than the Port Top Diameter

The geometric model for this case was present in the literature (Kulkarni, 2005). However, there was an error associated with the derivation of the model as pointed out in Section 2.6.1. A geometrically correct model has been derived in this study.

Figure 3-1: Ball diameter larger than port top diameter

In this case (Figure 3-1), the major area of the frustum is based on the PTD and it remains constant regardless of the stem movement. The minor frustum area changes as the stem moves upward and is dependent on PTD, stem travel, and ball diameter.

$$Frustum\ area = \frac{\pi z PTD}{2} \left(2 - \frac{z}{z + r_b} \right) \quad (3-1)$$

In this equation,

$$z = \sqrt{x^2 + \left(\frac{PTD}{2} \right)^2} - r_b \quad (3-2)$$

$$x = y + \sqrt{r_b^2 - \left(\frac{PTD}{2} \right)^2} \quad (3-3)$$

Here,

h_2 = Height of the seat, inch

h = Depth of the beveled part of the seat measured from the seat top, inch

PTD = Port top diameter, inch

r_b = Ball radius, inch

y = Stem travel, inch

The details of the derivations are presented in Section A.1.

3.2 Case 2: Ball Diameter Smaller than the Port Top Diameter

Since smaller balls have not been introduced in the industry yet, the geometric models for this case is not present in the literature. In this study, a rigorous geometric model for smaller balls has been developed from the ground up.

When the ball diameter is smaller than PTD, the ball contacts the seat at a level lower than the top edge of the seat, and the center of the ball is below the seat top. In this case, there can be two subcases: (i) the ball is partially inside the seat (Figure 3-2), and (ii) the ball is entirely out of the seat (Figure 3-3). For the first subcase, the perpendicular drawn from the center of the ball to the edge of the seat is inside the seat, and both the major and minor areas of the frustum change as the stem moves. The second subcase is similar to that of the larger ball where the perpendicular drawn from the center of the ball to the edge of the seat is outside the seat. In this subcase, the major frustum area is constant and based on the PTD, and the minor frustum area changes with the stem movement.

Figure 3-2: Ball diameter smaller than the port top diameter and ball is partially inside the seat

Figure 3-3: Ball diameter smaller than the port top diameter and ball is completely outside the seat

3.2.1 First Subcase: Ball is Partially Inside the Seat (Figure 3-2)

For this subcase, the generated frustum area is a function of PBD, PTD, ball diameter, stem travel, depth of the beveled part of the seat, and the diameter of the contact between the ball and the seat (CCD).

$$Frustum\ area = \frac{y\ r_b}{2a} \left\{ CCD + 2\ r_b \left(1 + \frac{y}{a} \right) \sin(d) \right\} \quad (3-4)$$

In this equation,

$$a = r_b \cos(b) + e + f \quad (3-5)$$

$$b = \tan^{-1} \left(\frac{2h}{PTD - PBD} \right) \quad (3-6)$$

$$e = \frac{h(CCD - PBD)}{PTD - PBD} \quad (3-7)$$

$$f = \frac{h \times PBD}{PTD - PBD} \quad (3-8)$$

Here,

y = Stem travel, inch

CCD = Contact circle diameter, inch

r_b = Ball radius, inch

h = Depth of the beveled part of the seat measured from the seat top, inch

PTD = Port top diameter, inch

PBD = Port bottom diameter, inch

3.2.2 Second Subcase: Ball is Completely Outside the Seat (Figure 3-3)

For this subcase, the generated frustum area is a function of PBD, PTD, ball diameter, stem travel, and the diameter of the contact between the ball and the seat (CCD).

$$Frustum\ area = \frac{j - r_b}{2} \left(PTD + r_b \frac{PTD}{j} \right) \quad (3-9)$$

In this equation,

$$j = \sqrt{(y - |g - r_b \cos(d)|)^2 + \left(\frac{PTD}{2}\right)^2} \quad (3-10)$$

$$g = \frac{h(PTD - CCD)}{PTD - PBD} \quad (3-11)$$

$$d = \tan^{-1}\left(\frac{2h}{PTD - PBD}\right) \quad (3-12)$$

Here,

r_b = Ball radius, inch

PTD = Port top diameter, inch

PBD = Port bottom diameter, inch

CCD = Contact circle diameter, inch

The details of the derivations are presented in Section A.2.

CHAPTER 4: EXPERIMENT AND SIMULATION

4.1 Experimental Procedures

The main aim of the entire experimental program is to test gas lift valve under dynamic conditions by employing “blowdown” or “pressure decay” concept as explained in detail by (Kulkarni, 2005) and (Shahri, 2011). The entire program is divided into the following four parts:

- 1) Static testing
- 2) Aging
- 3) Probe testing
- 4) Dynamic testing (Blowdown)

4.1.1 Static Testing

The objective of static testing is to set the pressure in the nitrogen-charged dome and to measure the test rack opening pressure. The testing procedure is:

- 1) After a seat and a ball are selected and installed in the GLV, the tail plug of the GLV is removed (Figure 4-1).

Figure 4-1: Tail plug of the GLV

- 2) Using correct sizes of adapters (Figure 4-2) and high pressure hoses the dome is overcharged with at least 50 psi or more. The approximate dome pressure or set pressure is read from pressure gauge of nitrogen bottle.

Figure 4-2: Dome charging adapter

- 3) The dome temperature is recorded using infrared temperature gun.
- 4) The GLV is placed in the testing block (probe tester).
- 5) The stem valve (Figure 4-3) is depressed to exit some nitrogen to lower the dome pressure in order to achieve the exact dome pressure.

Figure 4-3: Stem valve

- 6) Keeping the downstream side of the valve open to atmosphere, the injection or upstream pressure is gradually increased with the help of pressure regulator until enough pressure is reached to open the valve. At this pressure, quick release of nitrogen gas is observed at the downstream side of valve.
- 7) This Pressure is recorded as the test rack opening pressure (P_{TRO}).
- 8) Using Equation (4-1), the valve closing pressure (P_{vc}) is calculated.

$$P_{vc} = P_{TRO} \times C_t \times \left(1 - \frac{A_p}{A_b}\right) \quad (4-1)$$

Here,

P_{vc} = Valve closing pressure, psig

P_{TRO} = Test rack opening pressure, psig

C_t = Nitrogen temperature correction factor, dimensionless

A_p = Port area, inch²

A_b = Bellows area, inch²

4.1.2 Aging

Aging or hydraulic stabilization is done to reduce the hysteresis effect associated with the bellows assembly. The main parts of this apparatus are an air-driven water pump, a GLV holder, and some pop joints. A schematic diagram and an actual diagram of the apparatus are given in Figure 4-4 and Figure 4-5 respectively. This apparatus is called ager or GLV hydro tester.

Figure 4-4: Schematic diagram of the ager

Figure 4-5: Actual ager

The procedure for using the ager is as follows:

- 1) After setting the opening pressure of the GLV using static testing, the GLV is put into a 60°F water bath and kept there for at least 15 minutes.
- 2) The GLV is then inserted into the ager GLV chamber. During this step, the GLV should not be held from the dome section to eliminate the possibility of heat transfer from hand.
- 3) Close the lid of the GLV chamber.
- 4) The pump is started using the following procedure
 - a. Make sure all the lines are depressurized.
 - b. Close air shut-off valve.
 - c. Turn on driving air supply.
 - d. Adjust air pressure regulator to 25 psi starting pressure.
 - e. Open the valve in the hydraulic circuit and the water shut-off valve to allow free flow of water from reservoir.
 - f. Slowly open the air shut-off valve to start the pump operating.
 - g. After pump has been primed, close valve in hydraulic circuit.
 - h. Check pump and air circuit for leaks in lines, fittings, etc.
 - i. With pump and circuit operating properly, readjust air pressure regulator until desired pump discharge pressure (5000 psi) is reached.
 - j. Read and record the pressure off the gauge attached to the GLV chamber.
- 5) The GLV is kept inside the GLV chamber under 5000 psi pressure for at least 15 minutes.
- 6) The pump is stopped using the following procedure:
 - a. Close air shut-off valve. Normally after driving air supply has been adjusted, the pump can be on-off controlled or reduced in pumping rate at the air shut-off valve.
 - b. After stopping pump, bleed off hydraulic pressure before disconnecting the hydraulic circuit.
 - c. Close the water shut-off valve.
- 7) Open the lid of the GLV Chamber and remove the GLV. Then close the lid.
- 8) The GLV is immersed in the water bath again for at least 15 minutes.

- 9) The GLV is removed from the water bath and using static testing, the test rack opening pressure is measured.
- 10) If the test rack opening pressure has been changed by 5 psi or more, then steps 2 through 9 are repeated until the pressure does not change by maximum of 5 psi.

4.1.3 Probe Testing

The purpose of the GLV probe test is to determine the relative “stiffness” of a gas-lift valve and to determine the maximum effective travel of the stem. When gas pressure is admitted to the tester, it must act on the full area of the valve bellows to lift the stem off the seat. When this pressure is increased, the stem lifts further from the seat. By using the probe tester, an accurate measure of the stem travel versus pressure can be determined and the results are tabulated and plotted. A schematic diagram and an actual diagram of the probe tester are given in Figure 4-4 and Figure 4-5 respectively.

Figure 4-6: Schematic diagram of the probe tester

Figure 4-7: Actual probe tester

When the pressure is plotted as the ordinate and the stem travel as the abscissa, a relatively straight line will be generated for the effective stem travel. The slope of this line is an indication of the “stiffness” of the valve. The numerical value of the slope is called the Bellows Assembly Load Rate and is measured in psig/inch. The higher the load rate, the “stiffer” the valve and inversely, the lower the load rate, the “softer” the valve. The stepwise procedure is given below:

- 1) After setting the test rack opening pressure using the static testing and aging, the valve is inserted in the test sleeve and a micrometer is added to the downstream side of the valve.
- 2) By adjusting the knob attached to the micrometer, the micrometer probe is positioned such that it barely touches the ball. The corresponding micrometer reading is recorded.
- 3) A multimeter is attached to the two sides of the nylon bushing. A resistance reading of zero ensures the continuity of the circuit.

- 4) The pressure is increased until the probe no longer touches the stem tip. A drastic increase in resistance in the multimeter reading ensures this. This is the pressure at which the valve opens when test pressure is applied over the full bellows area.
- 5) The micrometer probe is positioned using the knob such that it barely touches the ball again which is demonstrated by zero resistance in the multimeter reading again. The corresponding micrometer reading is recorded.
- 6) Steps 2 through 5 are repeated for several increasing pressures until the stem no longer moves with increasing pressure.
- 7) Steps 2 through 5 are repeated for several decreasing pressures until the initial micrometer reading is reached.
- 8) The pressures vs stem travels are plotted on the same graph for both increasing and decreasing pressures.
- 9) There will be two regions in the plot: a left region with lower slope and a right region with higher slope. Some best fit lines are drawn through the data points in the region with lower slope.

The end of the lower slope region indicates the maximum linear stem travel and the slope indicates the load rate. The start of the higher slope region indicates starting of bellows stacking and the region where the increasing/decreasing pressures result in zero stem movement is the region where the bellows got completely stacked. A typical graph from probe testing is presented in Figure 4-8. If the test is done with the same valve, except that opening pressure (dome charge or spring setting) is varied, then the effect of dome charge pressure or spring setting on the bellows assembly load rate can be compared for the same type valve when set for different opening pressures.

Figure 4-8: Typical data from probe test

4.1.4 Dynamic Testing (Blowdown Test)

The objective of dynamic testing (also called blowdown or pressure decay test) is to rapidly measure the flow rate capabilities of the valves under simulated well conditions. As indicated in (Shahri, 2011), this method does not substitute the current method mentioned in (*Gas-Lift Valve Performance Testing API RP 11V2*, 2008) and is recommended for high production well where the flow regime is critical or supersonic.

The methodology behind this technique is simply discharging a certain volume of gas at a certain time until the upstream pressure reaches the final downstream pressure which is ambient pressure. The initial pressure is largely greater than the test rack opening pressure to ensure a fully open GLV.

The apparatus for this test includes some compartments such as source of high pressure nitrogen gas, upstream and downstream regulators attached to the high-pressured source of

gas, an extra empty volume with known internal capacity, an encapsulated vessel which holds the GLV, the GLV, high-speed pressure transducers, high speed temperature recorder, and a data-acquisition system (DAQ) which might be integrated with the pressure transducers. A schematic diagram and an actual diagram of the blowdown tester are given in Figure 4-9 and Figure 4-10 respectively.

Figure 4-9: Schematic diagram of the blowdown tester

Figure 4-10: Actual blowdown tester

The procedure for running the dynamic testing is as follows:

- 1) The GLV is attached inside the encapsulating vessel and the vessel is closed.
- 2) The upstream pressure is set at a very high value compared to the test rack opening pressure.
- 3) The main feeding valve that is attached to the high-pressure nitrogen source is shut-in.
- 4) Wait until the upstream pressure is stabilized.
- 5) The temperature is recorded using laser temperature gun.
- 6) The pressure and corresponding time recording is started.
- 7) The downstream valve is kicked open as fast as possible.
- 8) The temperature is recorded again using laser temperature gun.

Temperature is not expected to change much for such a short period of time and for small volume of gas discharged. However, a temperature variation of as small as $\pm 2^{\circ}\text{F}$ was

reported to change the flowrate by almost 30% (Sagar, 1991). That is why temperatures were measured and recorded both immediately before and after discharging the gas using an electronic laser thermometer. The reason for starting data recording before opening the downstream valve is to be able to detect the starting point of pressure decay with time. The output of this recording is pressure versus time as demonstrated in Figure 4-11. Once the pressure decay data (pressure vs. time) is recorded, the data need to be curve fitted because of being scattered. Regression curve-fit method was implemented on the data using different techniques to find the best fit. First order exponential decay and second order exponential decay were the best fitting techniques for most cases and these techniques are also recommended by (*Gas-Lift Valve Performance Testing API RP 11V2*, 2008). After curve fitting, the gas flow rate can be calculated using Equation (4-2) and Avogadro's rule. A sample output of the blowdown test is demonstrated in Figure 4-11. The gas compressibility factor was calculated at each pressure and temperature based on available correlations and found to be close to one. Some constant inputs were used in this analysis that were dominantly dependent on the physical location and testing facility such as the gas constant, ratio of specific heats of the active gas, the capacity of the storage facility, and the specific gravity of the working gas.

$$\frac{dn}{dt} = \frac{V}{Z \times R \times T_{up}} \frac{dP_{up}}{dt} \quad (4-2)$$

Here,

n = Number of lb-moles of gas, lb-moles

t = Time, second

V = Volume of gas, ft³

Z = Compressibility factor at valve conditions, dimensionless

R = Ideal gas constant, 10.732 psi.ft³/lb-mole/R

T_{up} = Upstream temperature at the inlet of the valve, R

P_{up} = Upstream pressure at the inlet of the valve, psig

Figure 4-11: Sample output of a blowdown test (modified design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, Ball 9/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)

4.2 CFD Modeling

The CFD model used in this study was initially built by (Elldakli, 2017) using a commercially available software package (Fluent from ANSYS). The model geometry is based on the experimental apertures where the fluid (gas) flow takes place (Figure 4-12). After building the geometry, the entire fluid flow space is divided into small finite volumes or cells that are interconnected. Polyhedral cells with hybrid mesh were used to generate the flow space. The number of cells used was 300,000. Finer and higher number of cells are used in the spatial distributions where fluid flow might be critical. The next step in the modeling is to setup the solution criteria. A steady state flow solver is used with smooth wall boundary conditions. Nitrogen and aluminum are used as the modeling fluid and solid respectively. Experimental injection pressure is used as the inlet pressure in the solution setup. Since the outlet of the experimental setup is opened to atmosphere, the atmospheric

pressure is used as the outlet pressure of the model. One of the inputs to the model is the vertical distance of the center of the ball measured from the seat top. The stem travel measured from the probe tester (Section 4.1.3) was used calculate this distance. The simulation was run multiple times with different numbers of iterations and it was found that 500 iterations are sufficient to reach satisfactory convergence. The rest of the solution methods and fluid flow model characteristics used in the modeling are presented in Table 4-1 and

Table 4-2 respectively.

Figure 4-12: Components of the experiment that were modeled

Table 4-1: Solution methods used in the numerical modeling

Pressure-Velocity Coupling	
Scheme	Simple
Spatial Discretization	
Gradient	Least Square Cell Based
Pressure	Linear
Density	Second Order Upwind
Momentum	Second Order Upwind
Modified Turbulent Viscosity	Second Order Upwind

Table 4-2: Fluid flow model characteristics

Criteria	Definition
Flow solver	Steady-state
Wall boundary conditions	Smooth wall
Turbulence model	Standard k - ϵ
Fluid model	Nitrogen (compressible gas)

4.2.1 Model Validation

The CFD model was validated by (Elldakli, 2017) using benchmark valve. In this study, the same model was validated by matching experimental data for numerous cases using actual GLV. For all the cases, the differences between the simulation results and the experimental results were 5% or less. For the sake of brevity, only the matchings of the following two cases are presented here.

Table 4-3: Model Validation (modified design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, Ball 5/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)

Time (t) [sec]	Pressure (Simulation) [psi]	Pressure (Experiment) [psi]	% Difference
0	450	428	5
0.1	436	417	5
0.2	424	406	4
0.3	412	395	4
0.4	401	384	4
0.5	390	374	4
0.6	380	365	4
0.7	371	356	4
0.8	362	347	4
0.9	354	339	4
1	346	331	4

Table 4-4: Model Validation (modified design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, Ball 5/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)

Time (t) [sec]	Pressure (Simulation) [psi]	Pressure (Experiment) [psi]	% Difference
0	450	431	4
0.1	425	411	4
0.2	402	391	3
0.3	381	372	3
0.4	361	353	2
0.5	342	335	2
0.6	325	318	2
0.7	309	302	2
0.8	293	287	2
0.9	279	274	2
1	265	261	2

Figure 4-13: Model Validation (modified design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, Ball 5/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)

Figure 4-14: Model Validation (modified design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, Ball 5/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig)

4.3 Seats and Balls Used

The seat and ball combinations used in this study are listed in Table 4-5, Table 4-6 and Table 4-7. For all cases, the test rack opening pressure was set at 300 psig and the injection pressure was 450 psig.

Table 4-5: Modified design (MD) seats and corresponding balls used

Seat Type	PBD [inch]	PTD [inch]	Ball Diameter [inch]
MD	3/16	4/16	5/16
			6/16
			7/16
		5/16	4/16
			6/16
			7/16
		6/16	5/16
			7/16
			8/16
		7/16	5/16
			6/16
			8/16
			9/16
		8/16	7/16
			9/16
	4/16	5/16	6/16
			7/16
		6/16	5/16
			7/16
			8/16
		7/16	5/16
			6/16
			8/16
			9/16
		8/16	6/16
			7/16
			9/16
	5/16	6/16	7/16
			8/16
		7/16	6/16
			8/16
			9/16
		8/16	7/16
			9/16
	6/16	7/16	8/16
			9/16
		8/16	7/16
			9/16
	7/16	8/16	9/16

Table 4-6: Optimized design (OD) seats and corresponding balls used

Seat Type	PBD [inch]	PTD [inch]	Ball Diameter [inch]
OD	3/16	8/16	7/16
			9/16
		9/16	7/16
			8/16
	4/16	8/16	6/16
			7/16
			9/16
		9/16	7/16
			8/16
			6/16
	5/16	8/16	7/16
			9/16
			6/16
	6/16	8/16	7/16
			9/16

Table 4-7: Sharp edge (SE) seats and corresponding balls used

Seat Type	Port Diameter [inch]	Ball Diameter [inch]
SE	3/16	4/16
	5/16	6/16

CHAPTER 5: RESULTS

The seat and ball combinations listed in Table 4-5, Table 4-6 and Table 4-7 were used to

- Optimize ball size
- Optimize port top diameter
- Optimize port bottom diameter
- Optimize seat type and
- Select the optimum combination of seat and ball

5.1 Optimizing Ball Size

Ball diameter 1/16-inch larger than the PTD of the seat is the API recommended ball size (*Gas-Lift Valve Performance Testing API RP 11V2*, 2008). Therefore, the performances of 1/16-inch larger ball sizes were first compared with 1/16-inch smaller ball sizes using the following cases (Table 5-1, Figure 5-1 and Figure 5-2).

Table 5-1: Effects of ball size (1/16-inch larger and smaller)

Seat Type	PBD	PTD	Ball Diameter	Gas Throughput	
	(inch)		(inch)	(inch)	(MSCF/D)
MD	3/16	8/16	7/16	417	2
			9/16	408	
	4/16		7/16	732	20
			9/16	584	
	5/16		7/16	1083	41
			9/16	640	
	6/16		7/16	1351	52
			9/16	651	
OD	3/16	8/16	7/16	423	2
			9/16	415	
	4/16		7/16	745	13
			9/16	651	
	5/16		7/16	1150	39
			9/16	704	
	6/16		7/16	1851	60
			9/16	734

Figure 5-1: Effects of ball size (modified seat design, port top diameter 8/16-inch, ball 1/16-inch larger and smaller)

Figure 5-2: Effects of ball size (optimized seat design, port top diameter 8/16-inch, ball 1/16-inch larger and smaller)

For all the eight seats tested here (Table 5-1), the gas throughput of the GLV decreased as a result of increasing the ball size. This decrease was more prominent (more than 50%) for larger port seats compared to smaller port seats (2%). These results lead to investigate the effects of even smaller balls. Now, ball sizes up to 2/16-inch larger and smaller were tested using three different seats.

Table 5-2: Effects of ball size (ball 2/16-inch larger and smaller)

Type	PBD	PTD	Ball Diameter	Frustum Area		Gas Throughput	
	(inch)	(inch)	(inch)	(inch ²)	% decrease	(MSCF/D)	% decrease
MD	3/16	7/16	5/16	0.0468		427	
			6/16	0.0460	2	426	0
			8/16	0.0334	27	395	7
			9/16	0.0096	71	97	75
	4/16	7/16	5/16	0.0456		752	
			6/16	0.0441	3	747	1
			8/16	0.0322	27	466	38
			9/16	0.0091	72	129	72
OD	4/16	8/16	6/16	0.0431		752	
			7/16	0.0423	2	745	1
			9/16	0.0350	17	651	13

In all the cases considered in Table 5-2, some common trends are observed. These trends are:

- (i) With decreasing ball size, the flow rate increases.
- (ii) The increase in flowrate is most significant when the ball size decreases from 2/16-inch larger to 1/16-inch larger than the PTD.

(iii) The increase in flowrate is least significant when the ball size decreases from 1/16-inch smaller to 2/16-inch smaller than the PTD. For these cases, the flow rate remained almost unchanged.

(iv) The lowest flowrate occurs when the ball size is 2/16-inch larger than the PTD.

In order to investigate the reason(s) behind these observations, both the gas throughputs and frustum areas were plotted on the same charts (Figure 5-3, Figure 5-3 and Figure 5-5).

The frustum areas were calculated using measured stem travels, and Equation (3-1), Equation (3-4) and Equation (3-9) as applicable.

Figure 5-3: Effects of ball size (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball up to 2/16-inch larger and smaller)

Figure 5-4: Effects of ball size (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball up to 2/16-inch larger and smaller)

Figure 5-5: Effects of ball size (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball up to 2/16-inch larger and smaller)

Furthermore, simulations were run for the same cases as in Table 5-2 to determine the mass flowrates and observe the pressure distributions.

Table 5-3: Mass flowrates from CFD simulations (optimizing ball size)

Seat Type	PBD	PTD	Ball Diameter	Mass Flow Rate
	(inch)	(inch)	(inch)	(lb _m /sec)
MD	3/16	7/16	5/16	0.5640
			6/16	0.5625
			8/16	0.5146
			9/16	0.4810
	4/16		5/16	1.0251
			6/16	1.0240
			8/16	0.6977
			9/16	0.4876
OD	4/16	8/16	6/16	0.4955
			7/16	0.4946
			9/16	0.4331

Figure 5-6: Mass flowrates from CFD simulation (optimizing ball size)

Figure 5-7: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 5/16-inch)

Figure 5-8: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 6/16-inch)

Figure 5-9: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 8/16-inch)

Figure 5-10: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 9/16-inch)

Figure 5-11: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 5/16-inch)

Figure 5-12: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 6/16-inch)

Figure 5-13: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 7/16-inch)

Figure 5-14: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 9/16-inch)

Figure 5-15: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 6/16-inch)

Figure 5-16: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-17: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 9/16-inch)

5.1.1 Result Analysis

- (i) For an actual GLV, the stem length from the bottom of the stem to the level where the stem tip (ball) begins is fixed. When a larger ball size is used, the overall length of the stem (from the bottom to the stem tip) increases. As a result, the bellows assembly remains more compressed for larger balls compared to smaller balls when the GLV is closed. Because of this, the stem cannot travel as high for larger balls when the GLV is open and a smaller frustum area is generated which results smaller gas throughput. This phenomenon is evident in all three seats. For the optimized seat (PBD 3/16-inch, PTD 8/16-inch), when the ball size was increased from 6/16-inch to 7/16-inch, it resulted in 8% decrease in frustum area and a 1% decrease in gas throughput. Further increasing the ball size to 9/16-inch resulted in 17% decrease in frustum area and 13% decrease in gas throughput. Similar results were obtained for modified seats. The mass flowrates (Table 5-3 and Figure 5-6) calculated from the numerical model bolster this experimental observation. The difference in change in frustum area and the change in gas throughput can be

attributed to the flow coefficient ((Decker, 1993) and (*Gas-Lift Valve Performance Testing API RP 11V2*, 2008)), which is the other important factor affecting gas throughput.

- (ii) For all three seats, the gas throughput is following a trend that is similar to the frustum area. This means, in these cases, the frustum area is the major factor affecting the gas throughput.
- (iii) The pressure distributions (Figure 5-7 - Figure 5-17) from the numerical simulation show that, when the ball is larger than the PTD, three drastic pressure drops take place for modified seats: one at the juncture of the beveled and the sharp edge parts of the seat, one around the ball-seat contact location, and one at the bottom of the seat. However, for the balls smaller than PTDs, only two such drastic pressure drops are evident for modified seats and one for optimized seat: inside the seat at the juncture of the beveled and the sharp edge parts, and at the bottom of the seat. For optimized seats, such pressure drops occur twice for larger balls as there is no sharp edge part, and only once for smaller balls. This results in a more uniform pressure distribution and less pressure drop inside the GLV for smaller balls. Since pressure distribution is directly related to flow coefficient, the combinations of smaller balls and seats tend to have better flow coefficient.
- (iv) For the larger ball cases, higher-pressure zones are evident at the stem tips (Figure 5-9, Figure 5-13, Figure 5-14 and Figure 5-17). Fluid streams have a tendency to attach to curved surfaces. This phenomenon is called Coanda effect. The lower the curvature the greater the Coanda effect (Li et al., 2014). Larger balls have lower curvatures. Gas streams preferentially attach to this lower curvature and instead of entering the seat, they create a turbulent zone at the tip of the stem.
- (v) For all three seats, decreasing the ball size increased the gas throughput. However, decreasing the ball size from 1/16-inch smaller to 2/16-inch smaller than the PTD resulted in a minimal increase in gas throughput (maximum 1%). Throughout the life of a GLV, the stem tip contacts the seat numerous times with high impact. That is why it is essential for the stem tip to have sufficient strength. Reducing the ball size makes the ball less sturdy. The increase in gas throughput by reducing the ball size comes at the expense of stem tip strength. Since the 2/16-inch smaller ball size

provides an insignificant improvement over 1/16-inch smaller ball, it can be concluded that the 1/16-inch smaller than the PTD is the optimum ball size.

- (vi) Both experimental and simulation studies demonstrated improvements in gas throughput in GLV using optimum ball size (1/16-inch smaller than the PTD) over API recommended ball size (1/16-inch larger than the PTD). Experimental results for modified 3/16-inch port, modified 4/16-inch port and optimized 4/16-inch port seats demonstrated improvements of 7%, 38% and 13% respectively over API recommended ball size. Also, simulation results for these cases demonstrated improvements of 9%, 32% and 12% respectively.

5.2 Optimizing Port Top Diameter

To investigate the effects of PTD, three modified and one optimized seats were used, each with increasing PTD (Table 5-4). For all of these cases, ball sizes 1/16-inch smaller than the PTDs were used since this was found to be the optimum ball size in Section 5.1.

Table 5-4: Effects of port top diameter (ball 1/16-inch smaller)

Seat Type	PBD	PTD	Ball Diameter	Gas Throughput		Mass Flow Rate (From Simulation)	
	(inch)	(inch)	(inch)	(MSCF/D)	% Decrease	(lb _m /sec)	% Decrease
MD	3/16	5/16	4/16	434		0.2624	
		6/16	5/16	428	1	0.2608	1
		7/16	6/16	426	0	0.2557	2
		8/16	7/16	417	2	0.2494	2
	4/16	6/16	5/16	754		0.4654	
		7/16	6/16	747	1	0.4643	0
		8/16	7/16	732	2	0.4533	2
	5/16	7/16	6/16	1242		0.7119	
		8/16	7/16	1169	6	0.6762	5
	OD	8/16	7/16	765		0.5077	
		9/16	8/16	738	3	0.4955	2

Figure 5-18: Effects of port top diameter – gas throughput and mass flowrate (ball 1/16-inch smaller)

As indicated in Table 5-4 and Figure 5-18, increasing PTD did not result in significant change in gas throughput. In fact, increasing the PTD resulted in small decrease in gas throughputs. The maximum decrease in gas throughput was 6% for larger PBD seat (5/16-inch port). Similar trend was observed for mass flowrates calculated using CFD simulation. Maximum decrease of 5% in gas throughput as a result of increasing PTD was detected for 5/16-inch port whereas this percentage varied between 0% – 2% for other seats. To determine the reason(s) behind these observations, frustum areas were calculated and plotted (Figure 5-19), and pressure distribution diagrams (Figure 5-20 - Figure 5-30) were analyzed.

Figure 5-19: Effects of port top diameter - gas throughput and frustum area (ball 1/16-inch smaller)

Figure 5-20: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 5/16-inch, ball 4/16-inch)

Figure 5-21: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 6/16-inch, ball 5/16-inch)

Figure 5-22: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 7/16-inch, ball 6/16-inch)

Figure 5-23: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-24: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 6/16-inch, ball 5/16-inch)

Figure 5-25: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 7/16-inch, ball 6/16-inch)

Figure 5-26: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-27: Pressure distribution (modified seat design, port bottom diameter 5/16-inch, port top diameter 7/16-inch, ball 6/16-inch)

Figure 5-28: Pressure distribution (modified seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-29: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-30: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 9/16-inch, ball 8/16-inch)

5.2.1 Result Analysis

- (i) According to Figure 5-19, although the frustum area increased with increasing PTD, the gas throughput remained almost unchanged with minor decreases for larger port seats (5/16-inch port). The mass flowrates calculated from simulation conforms to the gas throughput trends from experiments. This means flow coefficient is playing the major role in these cases compared to frustum area. With increasing PTD, the ball size needs to be increased as well. This brings into the effects of ball size as explained in Section 5.1. Larger balls create more flow restrictions which decreases flow coefficient. On the other hand, increasing the PTD results in increase in frustum area. The two opposing factors, working at the same time, keeps the overall gas throughput almost unchanged.
- (ii) One way of reducing the effects of larger ball size while increasing the PTD is to increase the PTD without increasing the ball size. In this case, the ball contacts deeper into the seat as PTD is increased. However, it was observed during the experiments that with the current stem design, it is not possible to use ball sizes that are less than 2/16-inch smaller than the PTDs. For ball sizes less than 2/16-inch smaller than the PTDs, the hexagonal part of the stem had contacted the seat top before the ball contacted the seat. This resulted in a leak in the GLV as no seal could be made between the ball and the seat.

Figure 5-31: Hexagonal part of the stem contacting the seat top

- (iii) The pressure distribution diagrams (Figure 5-20 - Figure 5-30) show that for a particular seat, the pressure distributions remained almost the same, albeit increasing the PTD. This matches with the experimental observations and calculated mass flowrates.
- (iv) Another effect of increasing the PTD is the reduction of sturdiness of the seat wall at the upper part of the seat. With increasing PTD, the upper wall thickness of the seat decreases which in turns, decreases the overall seat sturdiness.
- (v) Considering the reduction of seat sturdiness and minor effects on GLV gas throughput as a result of increasing PTD, it was concluded that the PTD 2/16-inch larger than the PBD is the optimum PTD. This also makes the use of ball size 1/16-inch smaller than the PTD possible.

5.3 Effects of Port Bottom Diameter

To investigate the effects of PBD, four modified and four optimized seats were used (Table 5-5). All the seats had the same PTD (8/16-inch) and the same ball size (7/16-inch) to eliminate the effects of PTD and ball sizes from comparison. For all of these cases, ball sizes 1/16-inch smaller than the PTDs were used since this was found to be the optimum ball size in Section 5.1. The pressure distribution diagrams are also presented in Figure 5-34 - Figure 5-41.

Table 5-5: Effects of port bottom diameter

Seat Type	PBD	PTD	Ball	Gas Throughput		Mass Flowrate	
	(inch)	(inch)	(inch)	(MSCF/D)	% Increase	(lb _m /sec)	% Increase
MD	3/16	8/16	7/16	417		0.2494	
	4/16			732	76	0.4643	86
	5/16			1083	48	0.7119	53
	6/16			1351	25	0.9229	30
OD	3/16			423		0.2811	
	4/16			752	78	0.4955	76
	5/16			1150	53	0.7445	50
	6/16			1688	47	0.9611	29

Figure 5-32: Effects of port bottom diameter (modified design seats)

Figure 5-33: Effects of port bottom diameter (optimized design seats)

Figure 5-34: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-35: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-36: Pressure distribution (modified seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-37: Pressure distribution (modified seat design, port bottom diameter 6/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-38: Pressure distribution (optimized seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-39: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-40: Pressure distribution (optimized seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-41: Pressure distribution (optimized seat design, port bottom diameter 6/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

5.3.1 Result Analysis

- (i) As indicated in Figure 5-32 and Figure 5-33, increasing the PBD results in almost linear increase in gas throughput for both types of seat. Mass flowrates calculated from simulations also manifest similar trend. With increasing PBD, the downstream flow area increases significantly. That is why the gas throughput also increases.
- (ii) The pressure distributions presented in Figure 5-34 - Figure 5-41 show that with increasing PBD, the angle between the beveled edge of the seat and horizontal gets closer to 90 degrees. This results in more uniform pressure distribution which helps in achieving better gas throughputs.
- (iii) If 1/16-inch larger ball is used, the gas throughput increases with PBD at a decreasing rate as opposed to almost linear increase in gas throughput when 1/16-inch smaller ball is used. This phenomenon is observed for both modified and optimized seats. The effect of increasing ball size (Section 5.1) adversely affects the effect of increasing PBD. This is why the positive effects of increasing PBD diminishes if larger balls are used.

Figure 5-42: Combined effects of port bottom diameter and ball size (modified seat design, port top diameter 8/16-inch)

Figure 5-43: Combined effects of port bottom diameter and ball size (optimized seat design, port top diameter 8/16-inch)

5.4 Effects of Seat Type

To investigate the effects of seat type, four seats were used each with modified and optimized designs (Table 5-6). All the seats had the same PTD (8/16-inch) and the same ball size (7/16-inch) to eliminate the effects of PTD and ball sizes from comparison. For all of these cases, ball sizes 1/16-inch smaller than the PTDs were used since this was found to be the optimum ball size in Section 5.1. The pressure distribution diagrams are also presented in Figure 5-34 - Figure 5-41.

Table 5-6: Effects of seat type

PTD (inch)	Ball (inch)	PBD (inch)	Seat Type	Gas Throughput		Mass Flowrate	
				(MSCF/D)	% Increase	(lb _m /sec)	% Increase
8/16	7/16	3/16	MD	417	1	0.2494	2
			OD	423		0.2544	
		4/16	MD	732	3	0.4533	9
			OD	752		0.4946	
		5/16	MD	1083	6	0.6762	10
			OD	1150		0.7445	
		6/16	MD	1351	25	0.9229	30
			OD	1688		1.1997

Figure 5-44: Effects of seat type - gas throughput and mass flowrate (port top diameter 8/16-inch, ball 7/16-inch)

Table 5-6 and Figure 5-44 demonstrate that the optimized design performs better than the modified design. To determine the reason(s) behind this observation, frustum areas were

calculated and plotted (Figure 5-45), and pressure distribution diagrams (Figure 5-46 - Figure 5-53) were analyzed.

Figure 5-45: Effects of seat type - gas throughput and frustum area (port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-46: Pressure distribution (modified seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-47: Pressure distribution (optimized seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-48: Pressure distribution (modified seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-49: Pressure distribution (optimized seat design, port bottom diameter 4/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-50: Pressure distribution (modified seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-51: Pressure distribution (optimized seat design, port bottom diameter 5/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-52: Pressure distribution (modified seat design, port bottom diameter 6/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

Figure 5-53: Pressure distribution (optimized seat design, port bottom diameter 6/16-inch, port top diameter 8/16-inch, ball 7/16-inch)

5.4.1 Result Analysis

- (i) For each PBD seat, the optimized design performs better than the modified design which is demonstrated by both experimental gas throughput and simulation mass flowrate (Figure 5-44). This difference is higher for larger port seats.
- (ii) For each PBD seat, the frustum area remains almost unchanged between modified and optimized designs (Figure 5-45). The increased performance for optimized design can be therefore attributed to an improved flow coefficient for optimized design.
- (iii) As indicated in Figure 5-46 - Figure 5-53, an extra pressure drop occurs at the juncture of the beveled and sharp edge part of the modified seats which is absent for optimized seats. This results in smoother pressure distributions inside the ports of the optimized seats which are manifested by the improved gas throughputs.

5.5 Selecting the Optimum Combination of Seat and Ball

For each seat, the optimum performance is observed when the PTD is 2/16-inch larger than the PBD, the ball diameter is 1/16-inch smaller than the PTD, and when optimized seat design is used. This is presented in Table 5-7 and Figure 5-54.

Table 5-7: Optimized combinations of seat and ball

Type	PBD (inch)	PTD (inch)	Ball (inch)	Frustum Area (inch ²)	Gas Throughput (MSCF/D)	Mass Flowrate (lb _m /sec)
MD	3/16	5/16	4/16	0.0253	434	0.2624
	4/16	6/16	5/16	0.0347	754	0.4654
	5/16	7/16	6/16	0.0454	1242	0.7119
	6/16	8/16	7/16	0.0576	1351	0.9229
OD	6/16	8/16	7/16	0.0576	1688	1.1997

Figure 5-54: Optimum combinations of seat and ball (gas throughput and mass flowrate)

With increasing PBD, both the experimental gas throughput and the mass flowrate calculated using simulation increase. For these cases, as the PBD increases from 3/16-inch to 6/16-inch, both the upstream flow area (frustum area) and downstream flow area (port bottom area) increase, which results in increasing gas throughput (Figure 5-55). Between the modified 6/16-inch port seat and optimized 6/16-inch port seat, both the flow areas remain unchanged. However, optimized design achieves better pressure distribution. This is why the optimized 6/16-inch port seat performs better than the modified 6/16-inch port seat. If highest gas throughput is desired, then it is suggested to use the largest PBD optimized seat with 2/16-inch larger PTD and 1/16-inch larger ball diameter.

Figure 5-55: Upstream and downstream flow areas of the optimum combinations of seat and ball

5.6 Improvement over Current Industry Practice

Currently in the industry, sharp edge seats with 1/16-inch larger balls are used. Table 5-8 compares the performances of current industry practice with the proposed design using 5/16-inch port as an example.

Table 5-8: Current industry practice vs. proposed design

	Current Industry Practice	Proposed Design	
Type	SE	MD	OD
PBD (inch)	5/16	5/16	5/16
PTD (inch)	5/16	7/16	7/16
Ball (inch)	6/16	6/16	6/16
Gas Throughput (MSCF/D)	1023	1242	-
Mass Flowrate (lb _m /sec)	0.5633	0.7119	0.7176

Experimental gas throughput shows an improvement of 21% if the proposed modified seat and ball are used instead of current industry practice. The experimental gas throughput of the equivalent optimized seat and ball combination (PBD 5/16-inch, PTD 7/16-inch, ball 6/16-inch) could not be obtained as the seat was unavailable. However, the CFD simulation of the optimized seat and ball combination shows an improvement of 27% over current industry practice. Larger port seats are expected to provide even greater improvement.

CHAPTER 6: CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

- An optimized combination of seat and ball was proposed for actual gas lift valve. This combination showed an improvement of more than 27% over the current industry practice for 5/16-inch port seat and is expected to provide greater improvement for larger port seats.
- Both experimental and numerical schemes were utilized to optimize the seat and ball combination.
- The CFD model was validated using experimental results for actual gas lift valve. The matching was exceptional with maximum error of only 5%.
- New geometric models were deduced to calculate the generated upstream area (frustum area) open to flow for balls that are smaller in diameter than the port top diameter of the beveled seats.
- The applicability of the results obtained from benchmark valve to actual GLV regarding the effects of port top, PBD, and seat type were checked. The effects of port bottom diameter and seat type were found to be similar for actual and benchmark valves.
- Balls with diameters 1/16-inch smaller than the port top diameters of beveled seats were found to perform significantly better than balls with 1/16-inch larger diameters. Frustum area calculated from experimental measurements and pressure distribution obtained from CFD simulation validated this effect of smaller balls.
- The port top diameter of the beveled seat was found to have insignificant effect on gas throughput of actual gas lift valves when 1/16-inch smaller ball was used. Two opposing factors, increasing frustum area and increasing flow restriction, resulted in this effect of port top diameter.
- The port bottom diameter of the beveled seat was found to have a significant positive effect on gas throughput of actual gas lift valves. Increased downstream area (port bottom area) and better pressure distributions were identified as the main reasons for this effect of larger port bottom diameter seats. This effect of port bottom diameter was found to be more prominent with smaller balls than larger balls.

- Better pressure distribution inside the port of the optimized seat was identified as the reason for better performance of this type of seat compared to the modified seat.
- Seats with larger port bottom diameters were found to be more affected by ball size, port bottom diameter and seat type (modified or optimized).

6.2 Recommendations

- Quantitatively study the effects of flow coefficient by implementing the API method for dynamic gas lift valve testing.
- Establish mathematical relationship(s) between flow coefficient and frustum area.
- Compare the gas throughput obtained from blowdown test to that obtained from the API method for dynamic gas lift valve test.
- Update the existing numerical model to include the effects of bellows assembly and dome pressure.
- Verify the optimized combination of seat and ball proposed in this study with 1-inch gas lift valve.
- Study the performance of gas lift valve for seats with convex-edged port (Figure 6-1). This type of seat is expected to perform even better than the current optimized seat as a result of better flow pattern inside the port.

Figure 6-1: Seat with convex-edged port

APPENDIX A: DERIVATIONS

A.1 Larger Ball

Figure 6-2: Frustum area as a function of stem travel for larger balls

In Figure 6-2,

$$LB = AD = r_b = \text{Ball radius}$$

$$AL = y = \text{Stem travel}$$

$$GB = PTD/2$$

$$LG = \sqrt{LB^2 - GB^2}$$

$$= \sqrt{r_b^2 - \left(\frac{PTD}{2}\right)^2}$$

$$AG = AL + LG$$

$$= y + \sqrt{r_b^2 - \left(\frac{PTD}{2}\right)^2}$$

Assume,

$$x = y + \sqrt{r_b^2 - \left(\frac{PTD}{2}\right)^2} \quad (6-1)$$

$$AB = \sqrt{AG^2 + GB^2}$$

$$= \sqrt{x^2 + \left(\frac{PTD}{2}\right)^2}$$

$$DB = AB - AD$$

$$= \sqrt{x^2 + \left(\frac{PTD}{2}\right)^2} - r_b$$

Assume,

$$z = \sqrt{x^2 + \left(\frac{PTD}{2}\right)^2} - r_b \quad (6-2)$$

$$\text{So, } AB = z + r_b$$

ΔAGB and ΔDIB are similar.

So,

$$\frac{IB}{GB} = \frac{DB}{AB}$$

$$\rightarrow IB = GB \times \frac{DB}{AB}$$

$$= \left(\frac{PTD}{2} \right) \times \frac{z}{z + r_b}$$

Frustum area

$$= \pi(FD + GB)DB$$

$$= \pi \left(\frac{DE}{2} + \frac{PTD}{2} \right) DB$$

$$= \frac{\pi}{2} (BC - 2IB + PTD) DB$$

$$= \frac{\pi z}{2} \left(PTD - 2 \times \frac{PTD}{2} \times \frac{z}{z + r_b} + PTD \right)$$

$$= \frac{\pi z}{2} \left[2PTD - PTD \left(\frac{z}{z + r_b} \right) \right]$$

$$Frustum\ area = \frac{\pi z PTD}{2} \left(2 - \frac{z}{z + r_b} \right) \quad (6-3)$$

A.2 Smaller Ball

A.2.1 When GLV is closed

Figure A-3: Ball diameter is smaller than port top diameter, ball is completely below the seat top, and the valve is closed

Figure 6-4: Ball diameter is smaller than port top diameter, ball's top edge is above the seat top, and the valve is closed

A.2.1.1 Perpendicular distance between the center of the ball and the contact circle (AK)

$$EF = \frac{1}{2}(PTD - PBD)$$

$$\alpha = \tan^{-1} \left(\frac{DE}{EF} \right)$$

$$\alpha = \tan^{-1} \left(\frac{2h}{PTD - PBD} \right) \quad (6-4)$$

B and C are the points of tangency. So, AC is perpendicular to DF.

$$\angle DFE = \angle DCG = \alpha$$

$$\angle DCH = 90^\circ$$

$$\angle GCH = \angle ACK = 90^\circ - \alpha$$

$$AK = r_b \cos \alpha \quad (6-5)$$

A.2.1.2 Diameter of the contact circle (CCD)

$$CK^2 = r_b^2 - AK^2$$

$$\rightarrow CK^2 = r_b^2 \{1 - (\cos \alpha)^2\}$$

$$\rightarrow BC = 2r_b \sqrt{1 - (\cos \alpha)^2}$$

$$CCD = 2r_b \sqrt{1 - \left[\cos \left\{ \tan^{-1} \left(\frac{2h}{PTD - PBD} \right) \right\} \right]^2} \quad (6-6)$$

A.2.1.3 Depth of the contact circle measured from the seat top (DCC)

$\triangle DGC$ and $\triangle DEF$ are similar. So,

$$\frac{DG}{CG} = \frac{DE}{EF}$$

$$\rightarrow DG = (GK - CK) \frac{h}{\frac{1}{2}(PTD - PBD)}$$

$$\rightarrow DCC = \frac{1}{2}(PTD - CCD) \frac{2h}{(PTD - PBD)}$$

$$DCC = \frac{h(PTD - CCD)}{PTD - PBD} \quad (6-7)$$

A.2.1.4 Perpendicular distance between the contact circle and the bottom of the beveled part of the seat (EG)

$$EG = DE - DG$$

$$\rightarrow EG = h - \frac{h(PTD - CCD)}{PTD - PBD}$$

$$EG = \frac{h(CCD - PBD)}{PTD - PBD} \quad (6-8)$$

A.2.1.5 Depth of the center of the ball measured from the seat top (DI)

$$DI = |DCC - AK| \quad (6-9)$$

A.2.1.6 Depth of the bottom edge of the ball measured from the seat top

$$\text{Depth of the bottom edge of the ball} = DI + r_b \quad (6-10)$$

A.2.1.7 Perpendicular distance between the top edge of the ball and the seat top (DL)

$$DL = |DI - r_b| \quad (6-11)$$

A.2.1.8 Perpendicular distance between the contact circle and the bottom edge of the ball (JK)

$$JK = r_b - AK \quad (6-12)$$

A.2.1.9 Central angle (θ)

In $\triangle ACK$

$$\cos \frac{\theta}{2} = \frac{AK}{AC}$$

$$\rightarrow \frac{\theta}{2} = \cos^{-1} \left(\frac{r_b \cos \alpha}{r_b} \right)$$

$$\rightarrow \theta = 2\alpha \quad (6-13)$$

A.2.2 When GLV is Open

Figure 6-5: Ball diameter is smaller than port top diameter, initially ball was completely below the seat top, and the valve is open (Case 1)

Figure 6-6: Ball diameter is smaller than port top diameter, initially ball was partially below seat top, and the valve is open (Case 1).

A.2.2.1 Base of the generated frustum (QS)

In ΔMRF

$$\tan \alpha = \frac{RM}{FR}$$

$$\rightarrow RM = \frac{PBD}{2} \left(\frac{2h}{PTD - PBD} \right)$$

$$RM = \frac{h PBD}{PTD - PBD} \quad (6-14)$$

In ΔABM

$$AM = AK + KR + RM$$

$$AM = AK + EG + RM \quad (6-15)$$

ΔABM and ΔNQM are similar. So,

$$\frac{NQ}{NM} = \frac{AB}{AM}$$

NA = Stem travel, y

$$NQ = (y + AM) \left(\frac{r_b}{AM} \right)$$

$$NQ = r_b \left(1 + \frac{y}{AM} \right) \quad (6-16)$$

$$\sin \angle ANQ = \frac{QS/2}{NQ}$$

$$QS = 2 NQ \sin \frac{\theta}{2} \quad (6-17)$$

Case 1 (If $QS \leq PTD$) (Figure 6-5 and Figure 6-6)

A.2.2.2 Slanted height of the generated frustum (PQ)

$$PQ = NQ - NP$$

$$\rightarrow PQ = r_b \left(1 + \frac{y}{AM} \right) - r_b$$

$$PQ = \frac{y r_b}{AM} \quad (6-18)$$

A.2.2.3 Area of the generated frustum

$$Frustum\ Area = \frac{PQ}{2}(CCD + QS) \quad (6-19)$$

Case 2 (If $QS > PTD$) (Figure 6-7 and Figure 6-8)

Figure 6-7: Ball is smaller than port top diameter, initially ball was partially below seat top, and the valve is open (Case 2)

A.2.2.4 Slanted height of the generated frustum (PT)

$$\rightarrow NU = y - DI$$

113

$$PT = NT - r_b \quad (6-20)$$

A.2.2.5 Top of the generated frustum (PV)

ΔNPV and ΔNTD are similar. So,

$$\frac{PV}{NP} = \frac{TD}{NT}$$

$$PV = r_b \frac{PTD}{NT} \quad (6-21)$$

A.2.2.6 Area of the generated frustum

$$Frustum\ Area = \frac{PT}{2} (PTD + PV) \quad (6-22)$$

APPENDIX B: SAMPLE CALCULATIONS

The sample calculations presented in this section are for modified seat design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball 9/16-inch, P_{TRO} 300 psig, P_{inj} 450 psig.

B.1 Probe Tester

Figure 6-9: Probe tester result (modified design, port bottom diameter 3/16-inch, port top diameter 8/16-inch, ball diameter 9/16-inch, test rack opening pressure 300 psig, injection pressure 450 psig)

Figure 6-10: Determination of load rate (increasing pressure)

Figure 6-11: Determination of load rate (decreasing pressure)

Table 6-1: Probe tester data

Increasing Injection Pressure			Decreasing Injection Pressure		
Pressure [psig]	Position [inch]	Stem Travel [inch]	Pressure [psig]	Position [inch]	Stem Travel [inch]
0	0.1815	0	367	0.237	0.0541
100	0.1819	0.0004	340	0.2369	0.054
200	0.1824	0.0009	333	0.2369	0.054
250	0.1825	0.0010	330	0.2368	0.0539
260	0.1825	0.0010	323	0.2362	0.0533
265	0.1826	0.0011	320	0.236	0.0531
270	0.1826	0.0011	315	0.233	0.0501
280	0.1826	0.0011	310	0.226	0.0431
285	0.1826	0.0011	305	0.217	0.0341
290	0.1829	0.0014	300	0.2065	0.0236
295	0.1829	0.0014	294	0.187	0.0041
304	0.1974	0.0159	290	0.1845	0.0016
310	0.2135	0.0320	285	0.1829	0
317	0.2290	0.0475	280	0.1829	0
327	0.2363	0.0548	270	0.1829	0
337	0.2370	0.0555	250	0.1829	0
340	0.2370	0.0555	200	0.1829	0
355	0.2370	0.0555	100	0.1829	0
367	0.2370	0.0555	0	0.1829	0

$$\text{Stem Travel} = \frac{0.0555 + 0.0541}{2} = 0.0548 \text{ inch}$$

Load Rate (Figure 6-10 and Figure 6-11)

$$= \frac{465.34 + 468.93}{2} = 467.14 \text{ psi/inch}$$

B.2 Blowdown Tester

Figure 6-12: Blowdown tester result (modified design, port bottom diameter 3/16"-inch, port top diameter 8/16-inch, ball diameter 9/16-inch, test rack opening pressure 300 psig, injection pressure 450 psig)

Table 6-2: Blowdown tester fixed data

Nitrogen Specific Gravity	0.9672
Ratio of Specific Heat (C_p/C_v)	1.4
Discharge Coefficient, C_d	0.865
Standard Pressure Base, psia	14.696
Standard temperature Base, °F	60
Atmospheric Pressure, psia	14.71
Capacity of Gas Storage Tank, ft ³	0.56
Gas Constant, psi.ft ³ /lb-mole.°R	10.732
Avogadro's Rule, scf/lb-mole	379.6354
Critical Pressure Ratio (P_{up}/P_{down})	0.5282818
Ck1	3.5
Ck2	1.428571
Ck3	1.714286
Ck4	2333.63
Gas Temperature, °R	529.5

Table 6-3: Blowdown tester experimental data

Time [s]	Fitted Pressure [psig]	Average Pressure [psig]	Gas Vented [lb/lb-mole]	Gas Vented [ft ³]	R _p /R _c	q (ft ³ /s)	Q (MSCF/D)
0	433.80096	433.0456914	0.000148858	0.05651185	0.528282	4.709011	406.8585
0.0120008	432.2904228	431.5387481	0.00014815	0.05624295	0.528282	4.689798	405.1985
0.0239934	430.7870733	429.975543	0.000159947	0.06072155	0.528282	4.669866	403.4764
0.0369963	429.1640127	428.418207	0.000146993	0.05580381	0.528282	4.650011	401.761
0.048997	427.6724012	426.9296007	0.000146401	0.05557895	0.528282	4.631032	400.1211
0.0609985	426.1868001	425.3859425	0.000157843	0.05992298	0.528282	4.611349	398.4206
0.0739931	424.585085	423.8486237	0.000145151	0.05510462	0.528282	4.59175	396.7272
0.0859939	423.1121624	422.378707	0.000144559	0.05487971	0.528282	4.573008	395.1079
0.0979947	421.6452515	420.8538936	0.000155971	0.05921218	0.528282	4.553566	393.4281
0.1109982	420.0625356	419.3353037	0.000143332	0.05441405	0.528282	4.534205	391.7553
0.122999	418.6080718	417.8838081	0.000142747	0.05419195	0.528282	4.515699	390.1564
0.1349998	417.1595444	416.3786316	0.000153913	0.05843064	0.528282	4.496507	388.4982
0.1479944	415.5977187	414.8795984	0.000141537	0.05373229	0.528282	4.477396	386.847
0.1599952	414.161478	413.4462513	0.000140966	0.05351577	0.528282	4.459121	385.268
0.1719967	412.7310246	411.9594122	0.000152079	0.05773475	0.528282	4.440163	383.6301
0.1849995	411.1877997	410.4786789	0.000139763	0.05305891	0.528282	4.421285	381.999
0.1970003	409.7695582	409.0638116	0.000139098	0.05280644	0.528282	4.403245	380.4404
0.2089929	408.358065	407.5960835	0.000150181	0.05701414	0.528282	4.384531	378.8235
0.2219964	406.8341019	406.1338659	0.000138012	0.05239412	0.528282	4.365889	377.2128
0.2339972	405.4336299	404.736252	0.000137448	0.05218027	0.528282	4.34807	375.6732
0.245998	404.038874	403.2869498	0.000148199	0.05626161	0.528282	4.32959	374.0766
0.2589926	402.5350255	401.8435629	0.000136283	0.05173766	0.528282	4.311188	372.4867
0.2709934	401.1521003	400.4634598	0.000135726	0.0515265	0.528282	4.293592	370.9664
0.2829942	399.7748194	399.0318143	0.000146441	0.05559425	0.528282	4.275338	369.3892
0.2959977	398.2888093	397.6060121	0.000134575	0.05108928	0.528282	4.25716	367.8186
0.3079985	396.9232149	396.2431691	0.000134032	0.05088342	0.528282	4.239784	366.3173

Time [s]	Fitted Pressure [psig]	Average Pressure [psig]	Gas Vented [lb/lb-mole]	Gas Vented [ft ³]	R _p /R _c	q (ft ³ /s)	Q (MSCF/D)
0.3199999	395.5631232	394.8299605	0.000144501	0.0548578	0.528282	4.221765	364.7605
0.3329939	394.0967979	393.4225203	0.000132895	0.05045182	0.528282	4.203821	363.2101
0.3449954	392.7482428	392.0767525	0.000132346	0.05024326	0.528282	4.186663	361.7277
0.3569962	391.4052622	390.6807962	0.000142787	0.05420709	0.528282	4.168864	360.1898
0.369999	389.9563302	389.2905027	0.00013123	0.04981955	0.528282	4.151139	358.6584
0.3820004	388.6246753	387.9620855	0.000130592	0.0495773	0.528282	4.134201	357.195
0.3939924	387.2994956	386.584072	0.000141005	0.05353051	0.528282	4.116631	355.6769
0.4069959	385.8686484	385.2111977	0.000129579	0.04919277	0.528282	4.099128	354.1647
0.4189967	384.553747	383.8989796	0.00012905	0.04899199	0.528282	4.082397	352.7191
0.4309975	383.2442123	382.5377544	0.000139238	0.05285966	0.528282	4.065041	351.2195
0.4440009	381.8312966	381.1825602	0.000127861	0.04854073	0.528282	4.047763	349.7268
0.4559929	380.5338238	379.8333583	0.000138057	0.05241129	0.528282	4.03056	348.2404
0.4689964	379.1328928	378.489188	0.00012687	0.04816425	0.528282	4.013423	346.7598
0.4809972	377.8454833	377.2043723	0.000126359	0.04797018	0.528282	3.997042	345.3444
0.4929986	376.5632614	375.8720748	0.000136228	0.05171701	0.528282	3.980055	343.8767
0.5059927	375.1808882	374.5452152	0.000125287	0.04756328	0.528282	3.963138	342.4151
0.5179941	373.9095423	373.2764971	0.000124769	0.04736667	0.528282	3.946962	341.0175
0.5299949	372.6434519	371.960464	0.000134612	0.05110355	0.528282	3.930182	339.5677
0.5429977	371.2774761	370.6497695	0.000123717	0.04696721	0.528282	3.913472	338.124
0.5549991	370.0220629	369.3969511	0.000123205	0.04677306	0.528282	3.897499	336.7439
0.5669999	368.7718393	368.0978336	0.000132842	0.05043147	0.528282	3.880934	335.3127
0.5799946	367.4238279	366.8040184	0.00012216	0.04637632	0.528282	3.864439	333.8875
0.5919954	366.1842089	365.5669292	0.000121662	0.04618704	0.528282	3.848666	332.5248
0.6039962	364.9496495	364.2836387	0.000131266	0.04983327	0.528282	3.832304	331.111
0.6169996	363.6176279	363.005586	0.000120629	0.04579513	0.528282	3.81601	329.7032
0.6290004	362.393544	361.7844143	0.000120055	0.04557722	0.528282	3.80044	328.358
0.6409931	361.1752847	360.5176503	0.000129615	0.04920651	0.528282	3.784288	326.9625

Time [s]	Fitted Pressure [psig]	Average Pressure [psig]	Gas Vented [lb/lb-mole]	Gas Vented [ft ³]	R _p /R _c	q (ft ³ /s)	Q (MSCF/D)
0.6539959	359.8600159	359.2556107	0.000119124	0.04522372	0.528282	3.768198	325.5723
0.6659973	358.6512055	358.0492987	0.000118632	0.04503678	0.528282	3.752818	324.2435
0.6779981	357.447392	356.7984064	0.000127911	0.04855938	0.528282	3.736868	322.8654
0.6909928	356.1494207	355.5526195	0.000117625	0.04465476	0.528282	3.720985	321.4931
0.7029936	354.9558183	354.3614529	0.000117145	0.04447251	0.528282	3.705798	320.181
0.7149944	353.7670875	353.1258001	0.000126393	0.04798338	0.528282	3.690043	318.8197
0.7279978	352.4845126	351.8951905	0.000116151	0.04409515	0.528282	3.674354	317.4642
0.7399986	351.3058685	350.7189517	0.000115677	0.04391517	0.528282	3.659357	316.1684
0.7519994	350.132035	349.4992118	0.000124725	0.04735005	0.528282	3.643805	314.8247
0.7649941	348.8663886	348.2844503	0.000114696	0.04354267	0.528282	3.628317	313.4866
0.7769949	347.7025119	347.1229184	0.000114234	0.04336722	0.528282	3.613508	312.2071
0.7889963	346.5433249	345.9180385	0.00012324	0.04678613	0.528282	3.598145	310.8797
0.8019991	345.292752	344.7181065	0.000113259	0.04299699	0.528282	3.582847	309.558
0.8139999	344.1434611	343.5711311	0.000112802	0.04282374	0.528282	3.568223	308.2945
0.8260014	342.9988011	342.381738	0.000121619	0.04617083	0.528282	3.553058	306.9842
0.838996	341.7646749	341.1972294	0.00011184	0.04245827	0.528282	3.537956	305.6794
0.8509968	340.6297838	340.0646543	0.000111383	0.04228498	0.528282	3.523516	304.4318
0.8629976	339.4995247	338.8897812	0.000120176	0.04562316	0.528282	3.508536	303.1375
0.8760011	338.2800376	337.7201134	0.000110357	0.0418955	0.528282	3.493624	301.8491
0.8879931	337.1601892	336.6021402	0.000109988	0.04175518	0.528282	3.47937	300.6175
0.8999939	336.0440912	335.4419872	0.000118671	0.04505154	0.528282	3.464577	299.3395
0.9129973	334.8398832	334.2865694	0.000109054	0.04140088	0.528282	3.449846	298.0667
0.9249981	333.7332556	333.1821713	0.000108615	0.04123406	0.528282	3.435766	296.8501
0.9369995	332.6310871	332.036959	0.000117099	0.04445475	0.528282	3.421163	295.5885
0.9499936	331.4428308	330.896421	0.000107694	0.0408843	0.528282	3.406623	294.3322
0.961995	330.3500112	329.8058601	0.000107248	0.0407153	0.528282	3.392718	293.1308
0.9739958	329.261709	328.6746283	0.00011571	0.04392744	0.528282	3.378294	291.8846

Time [s]	Fitted Pressure [psig]	Average Pressure [psig]	Gas Vented [lb/lb-mole]	Gas Vented [ft ³]	R _p /R _c	q (ft ³ /s)	Q (MSCF/D)
0.9869986	328.0875476	327.5479855	0.000106344	0.04037194	0.528282	3.36393	290.6436
0.9990001	327.0084233	326.4714848	0.000105827	0.04017562	0.528282	3.350205	289.4577
1.0109921	325.9345463	325.3547933	0.000114265	0.04337916	0.528282	3.335967	288.2275
1.0239955	324.7750402	324.2422662	0.000105006	0.03986402	0.528282	3.321783	287.0021
1.0359963	323.7094923	323.1788928	0.000104577	0.03970132	0.528282	3.308225	285.8307
1.0479971	322.6482934	322.0758058	0.000112833	0.04283553	0.528282	3.29416	284.6154
1.0610006	321.5033183	320.9776061	0.000103614	0.03933563	0.528282	3.280159	283.4057
1.0729926	320.451894	319.9279151	0.000103273	0.03920594	0.528282	3.266776	282.2494
1.084994	319.4039362	318.8386488	0.000111414	0.04229679	0.528282	3.252887	281.0494
1.0979968	318.2733614	317.7538286	0.000102396	0.03887327	0.528282	3.239056	279.8545
1.1099983	317.2342957	316.7169105	0.000101973	0.03871258	0.528282	3.225836	278.7122
1.121999	316.1995253	315.6416722	0.000109949	0.04174053	0.528282	3.212126	277.5277
1.1349937	315.0838191	314.5708225	0.000101108	0.03838421	0.528282	3.198474	276.3481
1.1469945	314.0578259	313.5469231	0.000100695	0.03822754	0.528282	3.185419	275.2202
1.1589953	313.0360203	312.4847843	0.000108645	0.04124541	0.528282	3.171876	274.0501
1.1719988	311.9335484	311.4269807	9.9841E-05	0.03790317	0.528282	3.15839	272.8849
1.1839996	310.920413	310.4159128	9.94335E-05	0.03774847	0.528282	3.145499	271.7711
1.1960004	309.9114127	309.3674524	0.000107211	0.04070102	0.528282	3.132131	270.6161
1.208995	308.8234922	308.3232714	9.85901E-05	0.03742828	0.528282	3.118818	269.4659
1.2209958	307.8230505	307.3248453	9.81928E-05	0.03727747	0.528282	3.106088	268.366
1.2329972	306.82664	306.2891582	0.000105934	0.04021628	0.528282	3.092883	267.2251
1.2460001	305.7516764	305.25806	9.72884E-05	0.03693411	0.528282	3.079737	266.0893
1.2579927	304.7644436	304.2725063	9.69575E-05	0.03680848	0.528282	3.067171	265.0036
1.2699935	303.780569	303.2497957	0.000104612	0.03971432	0.528282	3.054131	263.8769
1.282997	302.7190224	302.2312593	9.61347E-05	0.03649615	0.528282	3.041146	262.755
1.2949978	301.7434961	301.2577238	9.57424E-05	0.03634719	0.528282	3.028733	261.6826
1.3069985	300.7719514						

BIBLIOGRAPHY

- Acuna, H.G., 1989. Normalization of One-Inch Nitrogen Charged Pressure Operated Gas-Lift Valves. Tulsa University.
- Aliyeva, F., Novruzaliyev, B., 2015. Gas Lift – Fast and Furious, in: SPE Annual Caspian Technical Conference & Exhibition. Society of Petroleum Engineers, Baku, Azerbaijan. <https://doi.org/10.2118/177359-MS>
- Almeida, A.R., 2011. A Model To Calculate the Theoretical Critical Flow Rate Through Venturi Gas Lift Valves (includes Addendum). SPE J. 16, 134–147. <https://doi.org/10.2118/126184-PA>
- Beggs, H.D., 1984. Gas Production Operations. Gulf Pub Co.
- Bertovic, D., 1995. A Unified Model for Gas-Lift Valve Performance Incorporating Temperature Effects. Tulsa University.
- Biglarbigi, K., 1985. Gas Passage Performance of Gas-Lift Valves. Tulsa University.
- Cook, H.L., Dotterweich, F.H., 1946. Report on Calibration of Positive Flow Beans Manufactured by Thornhill-Craver Company, Inc., Houston, Texas, Aug. 10, 1946. Department of Engineering, Texas College of Arts and Industries, Houston, Texas, USA.
- Decker, A.L., 1976. Analytical Methods For Determining Pressure Response Of Bellows Operated Valves. Society of Petroleum Engineers.
- Decker, K.L., 2008. IPO Gas Lift Design With Valve Performance. SPE Prod. Oper. 23, 464–467. <https://doi.org/10.2118/109694-PA>
- Decker, K.L., 1993. Gas-Lift Valve Performance Testing. SPE Prod. Oper. Symp. <https://doi.org/10.2118/25444-MS>
- Elldakli, F., 2017. Testing, Modeling and Optimizing a Modified Design for Gas Lift Valve Seat. Texas Tech University.
- Elldakli, F., Soliman, M., 2017. Optimum design for new gas lift valve seat. J. Pet. Sci.

- Eng. 149, 456–464. <https://doi.org/https://doi.org/10.1016/j.petrol.2016.10.062>
- Elldakli, F., Soliman, M., Shahri, M., 2014a. Enhanced Gas Lift Valve Performance With Sharp Edged Seat Using Larger Balls, in: Southwestern Petroleum Short Course. Lubbock, Texas, USA.
- Elldakli, F., Soliman, M.Y., Shahri, M., Winkler, H.W., Gamadi, T.D., 2014b. Improved Gas Lift Valve Performance Using a Modified Design for GLV Seat, in: SPE Artificial Lift Conference & Exhibition-North America. Society of Petroleum Engineers, Houston, Texas, USA. <https://doi.org/10.2118/171342-MS>
- Faustinelli, J.G., 1997. Temperature and Flow Performance Modeling of Gas-Lift Valves. Tulsa University.
- Gas-Lift Valve Performance Testing API RP 11V2, 2nd ed, 2008. . American Petroleum Institute (API).
- Gas-Lift Valve Performance Testing API RP11V2, 1st ed, 1995. . American Petroleum Institute (API).
- Heguler, G., 1988. Dynamic Model of Gas-Lift Valve Performance. Tulsa University.
- King, W.R., 1944. Time and Volume Control for Gas Intermitters. 2339487.
- Kulkarni, R., 2005. Modeling of Gas Lift Valve Performance by Regressing Model Parameters to Rapidly Obtain Blowdown Data. Texas Tech University.
- Lea, J.F., Nickens, H.V., Wells, M.R., 2008. Gas well deliquification. Elsevier/Gulf Professional Pub.
- Li, G., Hu, Y., Jin, Y., Setoguchi, T., Kim, H.D., 2014. Influence of Coanda surface curvature on performance of bladeless fan. J. Therm. Sci. 23, 422–431. <https://doi.org/10.1007/s11630-014-0725-3>
- Montgomery, J.W., Neely, A.B., Vogel, J.W., 1974. A Field Test and Analytical Study of Intermittent Gas Lift. Soc. Pet. Eng. J. 257, 502–512. <https://doi.org/10.2118/4538-PA>
- Nieberding, M.A., 1988. Normalization of Nitrogen-loaded Gas-Lift Valve Performance.

University of Tulsa.

Nuenz, S.I.E., 1994. Flow Performance Modeling of Pressure Operated Gas-Lift Valves. Tulsa University.

Rodriguez, M.A., 1992. Normalization of Nitrogen-Charged Gas-Lift Valve Performance. Tulsa University.

Sagar, R., 1991. Improved Dynamic Model of Gas-Lift Valve Performance. University of Tulsa.

Salinas, B.J., Xu, Z., Richard, B.M., 2014. Optimizing Gas Lift Operations Using Disintegrable Gas Lift Valve Plugs, in: SPE Artificial Lift Conference & Exhibition-North America. Society of Petroleum Engineers, Houston, Texas, USA. <https://doi.org/10.2118/171374-MS>

Shahri, M., Winkler, H.W., 2011. Practical Method for Measurement of Injection-gas Throughput of Each Gas-Lift Valve before Well Installation, in: SPE Production and Operations Symposium. Society of Petroleum Engineers, Oklahoma City, Oklahoma, USA. <https://doi.org/10.2118/141055-MS>

Shahri, M.A., 2011. Simplified and Rapid Method for Determining Flow Characteristics of Every Gas-Lift Valve (GLV). Texas Tech University.

Takacs, G., 2005. Gas Lift Valves, in: Gas Lift Manual. PennWell Corporation, p. 182.

Turzo, Z., Takacs, G., 2009. CFD techniques determine gas-lift valve behavior. Oil Gas J. 107, 46–51.

Winkler, H.W., Blann, J.R., 2007. Gas Lift, in: Lake, L.W., Clegg, J.D. (Eds.), Petroleum Engineering Handbook, Volume IV: Production Operations Engineering. Society of Petroleum Engineers, Richardson, TX, p. 521.

Winkler, H.W., Camp, G.F., 1987. Dynamic Performance Testing of Single-Element Unbalanced Gas-Lift Valves. SPE Prod. Eng. 2, 182–190. <https://doi.org/10.2118/14348-PA>

Winkler, H.W., Eads, P.T., 1993. Applying the Basic Performance Concepts of Single-

Element, Unbalanced Gas-Lift Valves for Installation Design 8, 211–217.
<https://doi.org/10.2118/21636-PA>

Xu, Z., Richard, B.M., Kritzler, J.H., 2013. Smart Gas Lift Valves Enhance Operation Efficiency of Offshore Wells, in: SPE Annual Technical Conference and Exhibition. Society of Petroleum Engineers, New Orleans, Louisiana, USA.
<https://doi.org/10.2118/166291-MS>