University of Manitoba Chapter of SAE International

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Lubrication System and Shifting System Improvements

Final Design Report

Prepared by Team #18 – F4i

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Letter of Transmittal

December 5, 2011

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Please find attached a final submission of our final design report titled **LUBRICATION SYSTEM AND SHIFTING SYSTEM IMPROVEMENTS**. This report was prepared and submitted by Team # 18 – F4i, on December 5 2011. This team consists of Morgan May, Gerald Galvez, Korsh Najar and Bryce Gryba.

This report contains the design and construction methods for a system designed to prevent oil scavenging of the Formula SAE vehicles during the skid-pad events at competition. This report contains discussions on the design and fabrication methods, cost analysis and the modes of operation of the lubrication system.

In addition, recommendations and suggestions are included for shifting system upgrades and methods for extending the lubrication system to work in tandem with the shifting system. The shifting system could not be incorporated with this design project due to the constraints of this project, however, may be a suitable design project for next years project.

Our team would like to express our gratitude for your assistance in all aspects of this project. Please feel free to contact us for any concerns with the content of our final design report

Sincerely,
Team F4i

Morgan May ________________________

Gerald Galvez ________________________

Bryce Gryba ________________________

Korosh Najar ________________________

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Executive Summary

This report presents designs for the lubrication circuit, scavenging prevention system and recommendations for shifting system upgrades to the 2012 University of Manitoba Formula Student Design Team. The scavenging prevention system presented in this report should prevent scavenging assuming that the oil pump is unable to provide oil flow for at least 6 seconds. The proposed design incorporates an accumulator into the hydraulic circuit by means of rigid hydraulic lines and a one-way flow valve. When there is flow available from the oil pump, oil is stored in the accumulator at an elevated pressure. At times when the oil pump is no longer capable of providing sufficient pressure, the accumulator reacts to maintain flow to the lubrication circuit. Supporting calculations, MATLAB code for pressure and flow calculations and excel worksheets for determining pressures and sizes of accumulator are provided with this report.

Design recommendations for the 2011-2012 University of Manitoba formula SAE shifting system are also presented in this report. The leaking pneumatic shifting system should be repaired by replacing the current flex lines and a paint-ball gun air regulator, with rigid lines and an appropriately sized air regulator. Also, recommendations for how to link the high pressure lubrication circuit with the shifting system are outlined. A full design is not included for linking the lubrication and shifting systems as this would fall outside the mass budget for this project.

1. Introduction

Automotive racing is driven by performance and measured in ounces. The simplest way to increase performance in a vehicle is by reducing the mass and maximizing the force the vehicle can generate. The following simplification of Newton's 2nd law:

$$Acceleration = \frac{Force}{Mass}$$

shows the dynamics involved in the design of race vehicles. This relationship is set at the core of the University of Manitoba Formula Design Team (UMFDT) design considerations, and has been adopted by Team F4i in their design of a *Lubrication System and Shifting System Improvement*.

The UMFDT designs, builds and races a prototype formula type vehicle at competitions annually. These competitions consist of events designed to push the prototype vehicles to their maximum operating potential in areas such as speed, acceleration and cornering ability. A photo of a competition vehicle is illustrated in Figure 1.

Figure 1: UMFDT Vehicle - previous year

The skid-pad event consists of a figure-eight track where the vehicles must complete two right hand loops followed immediately by two left hand loops as shown in Figure 2 [1]. In previous competitions the skid-pad event has been of particular trouble for the UMFDT. The skid-pad event is not a typical type of event for the application of these vehicles; however, it is used to put the vehicle through the most arduous of conditions to demonstrate the vehicles performance ability. In addition to the vehicles ability to perform the physical challenges, points are awarded in part to design and marketability of the prototype vehicle.

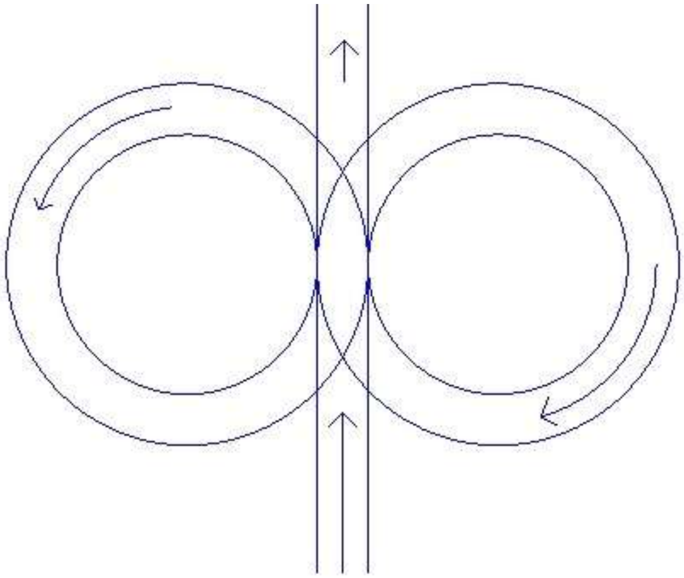

FIGURE 2: SKID-PAD DRIVING SKETCH

During the skid-pad event, oil has a tendency to slosh away from the oil pickup resulting in oil starvation of the engine for a few seconds. This problem, known as oil scavenging, has the potential to increase the metal-on-metal wear of engine components robbing the engine of power which could be directed to generating acceleration. This wear can result in a shortened lifespan of the vehicle or potentially a catastrophic failure of the engine.

The current gear shifting system in use by the UMFDT suffers from air leaks and requires constant maintenance to remain functional at competitions. Specifically, the shifting system leaks air from the pneumatic connections and its reservoir requires replacement after each event.

The lubrication of the engine and the shifting system leaks have not resulted in a failure of the vehicle yet. However the possibility does exist and the risk of failure needs to be minimized. Furthermore, by solving these problems a higher design score may be achieved which would increase the UMFDT's overall score at competition.

1.1 Project Objectives & Scope

This project has one primary objective, and one secondary objective. The primary objective of this project is to design a system that eliminates the negative effects of engine scavenging during prolonged or intense turns expected from the skid-pad events. The secondary objective is to develop a solution to the shifting systems leaks and constant maintenance. The lubrication issue was set as the primary focus as it has the potential to lead to excessive wear of the engine components, and in the worst case, may a catastrophic failure of the engine. The shifting system problems, although a serious issue, are given secondary priority as they are more of a nuisance rather than a critical operations concern.

From discussions with the UMFDT, a general list of requirements was compiled and in Table I.

TABLE I: UMFDT SYSTEM REQUIREMENTS

Identifier	Requirements
R.1	All additions should be relatively quiet.
R.2	Additions to the current vehicle configuration should be as light as possible.
R.3	The engine should run within normal manufacturer pressures and fluids.
R.4	Additions to the current vehicle configuration should be as small as possible.
R.5	Additions should operate without maintenance for at least two skid-pad events.
R.6	Modifications to the current vehicle should not require high pressure lines.
R.7	Modifications must not compromise the safety of the vehicle.

The goal of the final race vehicle is threefold. First, it must pass all the technical and safety tests required for a prototype vehicle at the FSAE competitions. Secondly, the vehicle must perform well during the competition; specifically in the skid-pad event. Finally, the vehicle must be of a quality which would be capable of mass production as this is the ultimate goal of the FSAE competitions.

Adhering to requirements R.1, R.6, and R.7 should result in a system which will not violate the competition technical or safety requirements. The rules of the competition change annually and typically only tighten the safety restrictions. Any system designed should not only be able to function for the UMFDT's vehicle, but should be modifiable to work with any foreseeable future models.

Therefore, these three requirements will be more strictly followed to allow for stricter rule changes for future UMFDTs.

Requirement R.3 is used to limit the design scope of the project to solutions which are manufacturable without significant modification to the existing engine. Significant reverse engineering of the engine would be required to determine what effect different fluids may have on the engine. While solutions using different fluids may be possible, they fall outside the scope of this project.

Requirements R.2, R.4, and R.5 force solutions to follow the UMFDT fundamental concept of minimizing weight while maximizing performance. Deviation from these requirements is acceptable only in the case when it can be shown that doing so will only serve to increase the vehicles performance or enhance the durability of the vehicle.

1.2 Specifications

The following list of specifications was made from analyzing the list of requirements for quantifiable values. Table II includes a list of the specifications and their preferred values.

TABLE II: SYSTEM SPECIFICATIONS [1]

Specification Identifier	Requirement Link	Specifications
S.1	R.1	No system shall increase the sound level of the vehicle to exceed 110 dB.
S.2	R.2, R.3	Additions to the current vehicle configuration shall not exceed 3.75 lbs.
S.3	R.3, R.6	The lubrication system shall operate at a minimum pressure of 70 psi when at idle.
S.4	R.2	The lubrication and shifting systems will not exceed a combined volume of approximately 1 liter.
S.5	R.6	The lubrication system shall not exceed fluid pressures of 300 psi.
S.6	R.3	The lubrication system shall use 0W-40 synthetic oil.
S.7	R.3, R.5	The lubrication system shall operate at engine speeds of 10 000 rpms.
S.8	R.6	The shifting system shall not require fluid pressures in excess of 300 psi.
S.9	R.5	The shifting system should be able to run without requiring maintenance for at least 31.1 miles (two endurance events with a F.O.S.).
S.10	R.5, R.4	The shifting system shall operate for a minimum of 2000 shifting actuations.

2. Details of Lubrication Scavenging Prevention Design

The final design of the lubrication scavenging prevention system is presented in this section; preliminary design concepts are included in Appendix A. Additional concepts considered after the final design phase began which were unable to be fully explored are included in Appendix B. Section 2.1 describes the design of the lubrication system and explains how the system will solve the scavenging problems with the UMFDT vehicle. Section 2.2 shows the components, discusses the assembly method of the design, and the modifications required to the current system. Additional research, supporting calculations are included in Appendices C, D, E, F and G.

The general design of the scavenging prevention system is that an accumulator stores fluid while the oil pump is capable of supplying flow. When scavenging begins the accumulator discharges its supply of oil through the oil lines providing a flow of oil to the engine.

2.1 Design Features

The contribution of scavenging prevention system allows oil to flow to the engine when the current lubrication system scavenges for oil. This system incorporates a gas-charged hydraulic piston accumulator, hydraulic lines of different sizes, a one-way check valve, and fittings. This section explains how these additions to the lubrication circuit solve the scavenging issue.

The stock oil pump on the race vehicle is capable of supplying considerably more oil than what is required to lubricate the engine. The hydraulic accumulator, shown in Figure 3, takes advantage of the additional flow by tapping into the pressurized oil supplied from the current pump and storing the oil for later use.

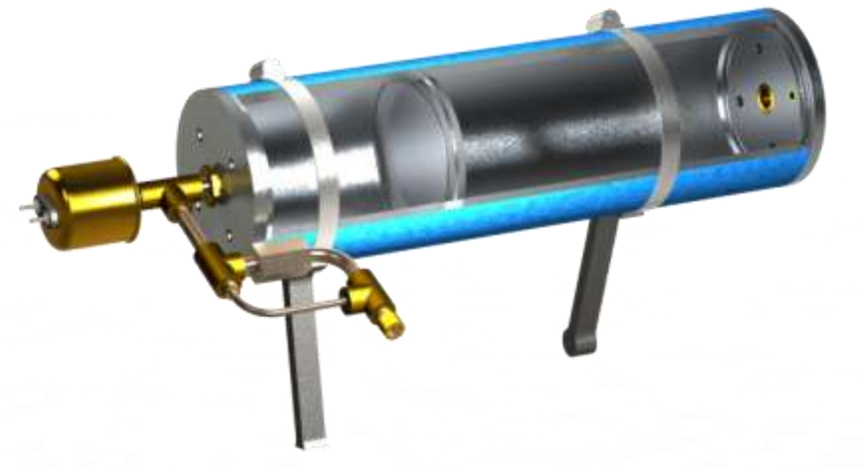

FIGURE 3: LUBRICATION SYSTEM ADDITION

Figure 4 illustrates how the accumulator is positioned in the lubrication circuit such that when there is a supply of oil flowing from the oil pump, the flow is branched and diverted to the accumulator. The flow of oil into the accumulator passes through both of the 1/4 and 1/8 inch hydraulic lines as illustrated in Figure 4Figure 4.

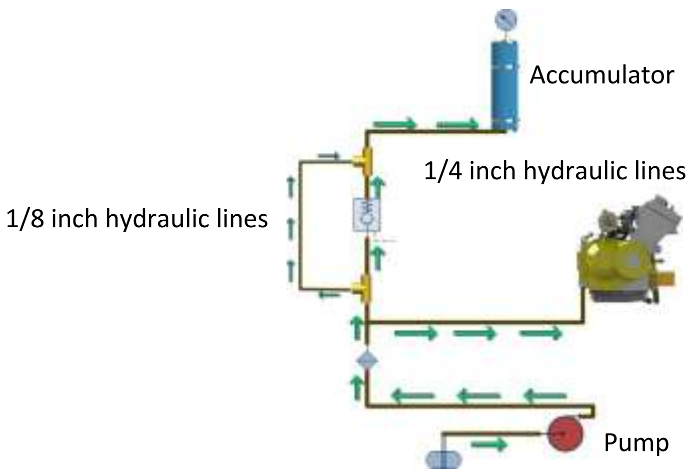

FIGURE 4: LUBRICATION CIRCUIT - CHARGING

Once the accumulator is filled or reaches the same pressure as the engine, the oil is kept in the accumulator until there is a drop in oil pressure in the engine relative to the accumulator. When the engine begins to scavenge for oil, the sudden decrease in oil pressure enables the accumulator to supply the stored oil to the lubrication circuit as shown in Figure 5. When the accumulator discharges, the one-way flow valve forces the stored fluid to flow through the smaller 1/8 inch branch. The pump rotor effectively acts as a plug on one end of the lubrication circuit to prevent the flow to returning back to the oil pan. This allows the oil to flow in the desired direction and toward the engine components.

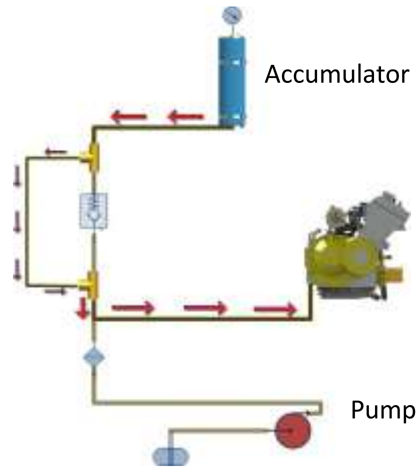

FIGURE 5: LUBRICATION CIRCUIT - DISCHARGING

The accumulator is located as close to the engine as possible to minimize the need for a long flow path. Longer pipes introduce additional resistance to the flow and the additional volume by the larger pipe decreases the accumulator's response time. Coincidentally, the space in front of the engine oil tap provides an adequate amount of space for placing a horizontally mounted hydraulic accumulator. By having the accumulator close to the engine oil tap, the gravitational effects in the difference in elevation of the fluid flow are negligible. The engine oil tap is located after the oil filter so that the oil flowing into the accumulator system is ensured to be clean of any debris extending the lifespan of the piston seals.

The initial pre-charge air pressure in the gas chamber of the accumulator sets the operating oil capacity and affects the performance of the accumulator. The accumulator reaches its full capacity when the pre-charge pressure is set to 13.5 psi according to the calculations in Appendix F. This is the minimum pressure that can be set to allow the flow of oil to overcome resistances in the tubes. A Schrader valve on the end cap of the gas chamber allows the user to set the minimum pre-charge pressure. A higher pre-charge pressure will allow a higher flow rate as the accumulator reaches its minimum operating pressure but at the expense of a reduced oil capacity.

Calculations supplied in Appendix F show that the time the engine scavenges for is approximately 6 seconds. From the flow analysis section in, the accumulator will supply an average of 3.535 L/min per 1000 RPM. This means that a 1 quart accumulator will be capable of supplying oil for more time than the engine will likely scavenge for given the data available.

The system design is made to be adaptable for future generations of UMFDTs vehicles. For optimal adaptability, a gas-charged hydraulic accumulator is used in the design. A gas-charged type accumulator allows the system to be modified to account for different engine operating pressures. In addition, the accumulator can be mounted in both horizontal and vertical orientations without sacrificing its lifespan. These features allow this accumulator type to be modified for different engine configurations without the need to purchase a new accumulator yearly.

For the application of a race vehicle where every ounce detracts from the overall performance, higher weight is undesirable. Since the accumulator will be located just above the engine, temperatures will vary greatly which may severely affect the operation of a spring type accumulator. For these reasons, a compressed gas accumulator is recommended over another type of accumulator.

The accumulator selected only uses about half of the total volume available for fluid as calculations in Appendix F indicates. The slightly oversized accumulator has been selected as the supporting calculations have been based on an assumed maximum scavenging time of 6 seconds from the data supplied by the UMFDT. Depending on the race, driver and possible engine configurations this time may increase and the larger accumulator size will allow for a theoretical 20 seconds of scavenging time. Also, depending on the racing conditions, if the accumulator is not allowed to completely recharge between uses, the larger size would also be beneficial.

A restrictor plate has been added at the engine end of the 1/8 inch tubing to allow for fine tuning of the final product. Since the exact model of the dynamics of the lubrication system could not be made in the time constraints of the project, this system functions as a ‘best guess’ given all available information at the time. This restrictor plate will allow the UMFDT to easily modify the orifice size resulting in a change of the fluid return rate during accumulator discharge. This feature is a low cost method that is easy to implement for the UMFDT allowing for quick modification of the lubrication system addition during testing.

2.2 Models & Assembly

The accumulator will be at the rear of the vehicle between the engine and the body as seen in Figure 6. The accumulator which has a 4 inch diameter indicated in Figure 6, will be placed in the gap between the engine and the body.

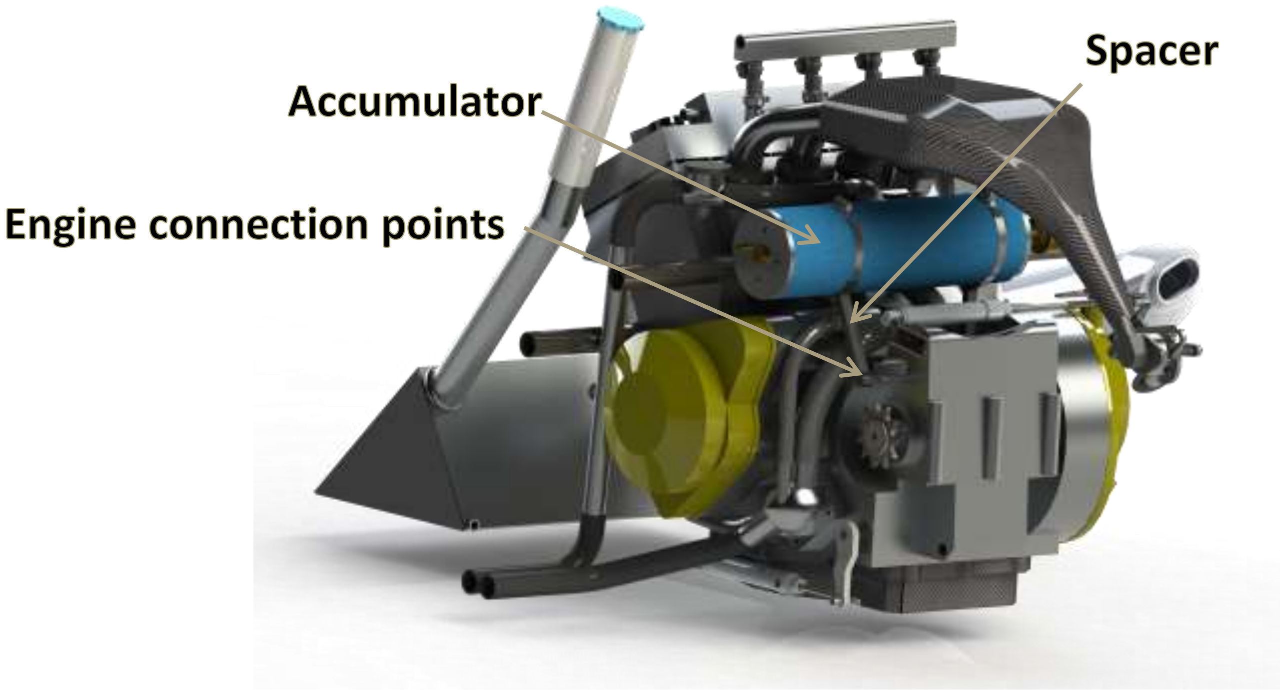

FIGURE 6: LUBRICATION SYSTEM PLACEMENT

Spacers and hose clamps are used to rigidly connect the accumulator to the engine. The spacer-engine connection point is indicated in Figure 6. It is recommended for the accumulator to be mounted in away such that the fluid chamber is slightly elevated compared to gas-charged end. A slight elevation helps prevent air bubbles from entering the accumulator [2].

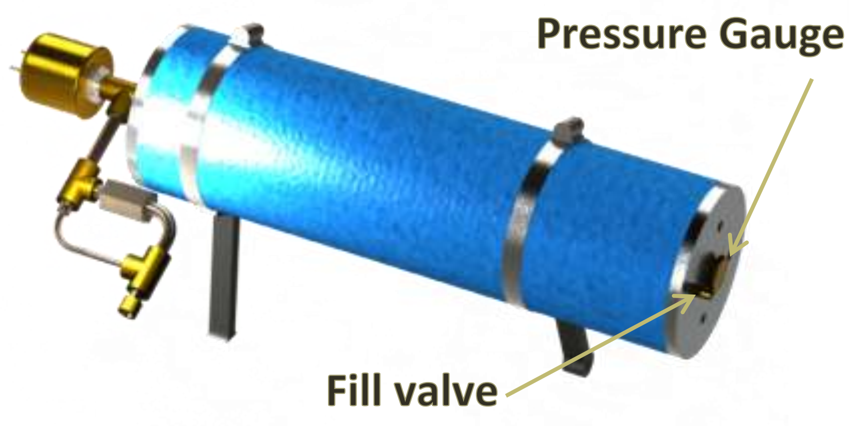

FIGURE 7: ACCUMULATOR WITH STOCK ATTACHMENTS

A standard Accusump accumulator includes safety pressure relief valve, pressure gauge as well as an air fill valve indicated in Figure 7.

Figure 8 illustrates the lubrication system with all critical components:

A. 1/2-AN-4 adaptor will be required to connect the pipe directly to the accumulator

B. One-way flow valve is used to speed charging and minimize discharge rate

C. Tee joints with 1/4 and 1/8 inch pipe fitting

D. Restrictor plate

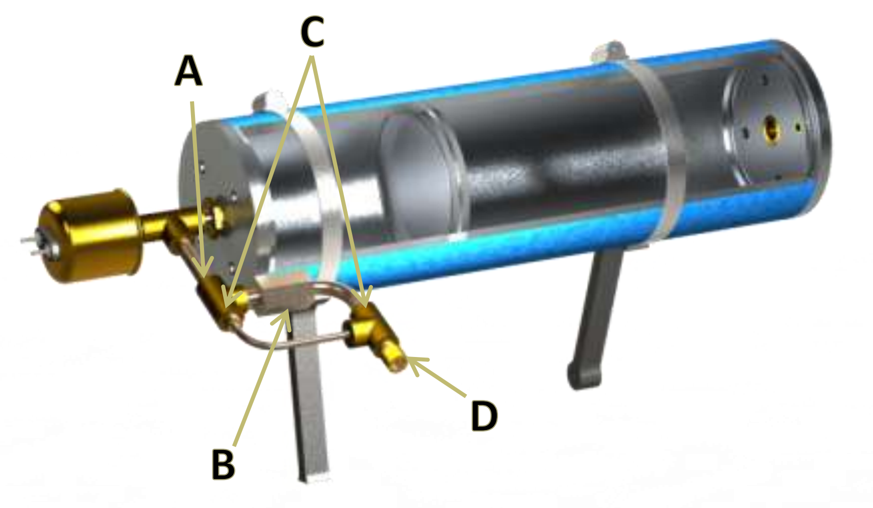

FIGURE 8: LUBRICATION COMPLETE ASSEMBLY

A complete copy of CAD models for all parts is included in Appendix I. Costs for all parts listed here are detailed in Section 4.

In addition to the design outlined in this section, an additional two concepts were considered after the point-of-no-return had been reached in regards to design and analysis. While these two concepts could not be fully evaluated as solutions for this project, they have been included in Appendix B as alternatives to the design presented in this report.

3. Shifting System Design

Details regarding the shifting system recommendations are outlined in this section. The shifting system is composed of an electro-mechanical control system which actuates the external linkages of the gearbox. An electro-mechanical system is not mandated however is desired as it allows for faster shifting and better driver control. The current pneumatic shifting system used by the UMFDT suffers from leaks which result in constant maintenance and high incurred costs. The suggestion of replacing the current pneumatic portion of the shifting system with a hydraulic system powered by pressurized oil from the engine accumulator was considered.

Data obtained from the UMFDT was analyzed in order to obtain an alternate design. Two problems emerged from this data concerning the required oil pressure by the shifting system and the oil flow rate of the engine. The problems with pressure and flow are analyzed in the following sections.

3.1 Considerations

Previous generations of design teams have used linkages in conjunction with a pneumatic piston to activate the clutch and shift gears. The linkages for the clutch and gear shifter require different forces to actuate. Replacing the pneumatic portion of the system with a hydraulic system would require the pressurized oil to be directed to the piston cylinders to actuate the linkages. Due to space constraints and the geometry of the current shifting system, the lubrication system upgrade would be insufficient to actuate the shifting system. Without significant modifications to the geometry and deviation from the mass and size constraints of this project, a hydraulic system cannot be used. Additional details and supporting calculations are discussed in Appendix H.

The oil flow rate of the F4i engine had to be examined to determine if enough oil could be supplied by the engine's lubrication system to supply both the engine and the shifting system. Data obtained from the UMFDT 2011 testing was analyzed. The data shows that the system pressure increases linearly with respect to the engine rpm to a maximum peak at 4500 rpm [3]. This relationship prevents sufficient pressure to activate a shifting system at lower engine rpms. An increase of the lubrication pressure relative to the engine rpm would be needed, otherwise a source of high pressure oil would be required at all times to enable hydraulic shifting at low engine speeds.

Modifying the flow rate from the oil pump would allow for a lubrication system to power the shifting system. However, these modifications would require a significant redesign of the pump. These modifications are possible and would allow the system to have the required flow rate to supply both the lubrication system as well as the shifting system. However, modifying the oil pump may result in changes in the dynamics of the vehicle's other sub-systems. These changes would need to be evaluated before a final decision could be implemented. Given the time constraints with this project evaluating the effect on other sub-systems was infeasible at this.

An alternative method without modifying the oil pump is the addition of a secondary dedicated shifting accumulator. The secondary accumulator would provide the shifting solenoids with high pressure oil charged by the lubrication circuit. At higher engine rpms, high pressure oil from the lubrication circuit charges the shifting accumulator. When the oil pressure drops, the one-way valve retains the high accumulator pressure. The primary accumulator prevents oil scavenging while the secondary accumulator allows for shifting. However, adding a second accumulator is not possible due to space constraints. The benefits of a secondary accumulator would not outweigh the drawbacks of the additional mass and volume.

3.2 Design Recommendations

Due to the limitations of this project, linking of the lubrication system, and shifting system was not possible. Adding two hydraulic accumulators would have increased the weight of the system beyond the specified maximum weight. Also, by linking the shifting system with the lubrication system, the size of the two accumulators would be greater than the maximum size restriction. If linking the lubrication system and shifting system is intended to be pursued, two alterations should be considered; increasing the lubrication circuit pressure or adding a dedicated shifting accumulator.

3.3 Shifting System Recommendations

Linking the lubrication circuit and shifting system is not possible under the project restrictions. Therefore repairing the leaks in the current pneumatic system is recommended. This may be accomplished by using rigid copper lines and an airline pressure regulator. Copper lines are not able to kink or fracture and will slightly increase the weight of the shifting system. The slight increase in weight, however, should be justifiable as the maintenance required would be reduced due to the more durable lines. These recommendations will greatly increase the reliability and endurance of the shifting system.

4. Cost Analysis

The UMFDT is a student run and student funded design team, as such the funds available for this project are limited. An overall budget for this project was placed at $300. This budget motivated the designs to be inexpensive in nature at the expense of potentially better design choices. Design choices were also made such that the system designed would be adaptable for future vehicles without high associated costs.

The UMFDT is sponsored by vendors who supply both in-kind as well as monetary donations to the team. For the purpose of this cost analysis, donations are not accounted for. The cost analysis is performed assuming that all components would need to be purchased at the standard off-the-shelf retail prices. Furthermore, no cost-benefit analysis is included as this system is intended for a prototype vehicle as a proof of concept design. Therefore, since there is no intended commercial sale of the vehicle, a break-even point is irrelevant and a cost-benefit analysis not applicable.

Canton Racing Products was identified as a suitable vendor for the purchase of the hydraulic accumulator. The accumulators manufactured by Canton Racing are specifically designed for racing applications and are supplied with safety pressure relief valves, air fill valves, and air gauges. However, the one-way flow valve is not included in the standard package. Various sizes of accumulators are available including the 1 quart size specified by this design. Table III lists the accumulator's parts and specifications and the total cost of required components is calculated to be $228.96 Canadian [4].

TABLE III: BILL OF MATERIALS [4]

Item #	Description	Quantity	Part Number	Vendor	Specifications	Cost
1	1 Quart Accusump	1	24-046	Canton Racing	12 inch long and 3-1/4 inch D	150 USD
2	Union Tee	2	A-400-3	Swagelok	1/4 inch NPT	18.00 CAD
3	Tube Fitting	2	B-400-7-2	Swagelok	1/4 x 1/8 inch	3.49 CAD
4	SS Hose Clamp	2	8125925	Princess Auto	3-1/2 inch	1.49 CAD
5	Check Valve	1	24-280	Canton Racing	1/2 inch NPT	27.60 USD
6	The manual control ball valve (Optional)	1	24-260	Canton Racing	1/2 inch NPT	15.60 USD
7	Accusump mounting clamps (Optional)	2	24-240	Canton Racing	1-QT-3-1/4 inch	20.40 USD
8	Alloy 360 Brass Tube	1	BRR18	Metals Depot	1/8 inch- 4 Ft.	5.40 USD

The system outlined in this report is intended for the 2012 UMFDTs vehicle and is designed so that it is modifiable if needed. The products listed as optional in Table III are products which may be considered by future generations of UMFDT.

5. Conclusion

This report evaluates two sub-systems on the 2012 UMFDTs race vehicle, the lubrication system as the primary focus and the shifting system as the secondary consideration. The solutions presented for the lubrication system and recommendations for solving the issues with the current shifting system are outlined. Also, recommendations for future work on both systems are suggested.

The lubrication system presented in this report satisfies the UMFDTs requirements for maintaining oil flow during turns. The system design outlined in this report remains within the specified budget set aside for the lubrication system, remains safe, and fits within the mass and volume budget for the 2012 UMFDTs race vehicle. This system incorporates a hydraulic accumulator into the current lubrication system. Fluid is stored in the accumulator when excess pressure is available. When the pump begins to scavenge the accumulator supplies the lubrication circuit with fluid for up to 6 seconds of flow.

The shifting system could not be linked with the lubrication system while satisfying all project specifications. Instead, an alternate solution of fixing the system by replacing the current flex lines with rigid copper lines and replacing the paintball-gun air regulator with an appropriate air regulator is suggested. This solution will solve the air leaks and constant maintenance problems with the current system.

Recommendations for linking the shifting system are included in this report. Specifically, a secondary dedicated accumulator is added for the purpose of actuating the shifting system. This system could not be justified for this project, as the implementation of this system would have resulted in a significant deviation from the mass, volume, and cost budgeted for this project.

In conclusion, the addition of a hydraulic accumulator to the lubrication circuit is recommended University of Manitoba Formula Design Team's 2012 vehicle.

In addition to the CAD models supplied in Appendix I, a digital copy of the MATLAB code with the modified data set used for generating the results presented in this report will be supplied; as well as an Excel worksheet containing programming and instructions for modifying the system outlined in this report. These two additional documents will allow future generations of the UMFDT to easily adapt the work presented in this paper for future iterations of design vehicles. These codes will form the basis for work done in the future to further optimize the scavenging prevention system.

References

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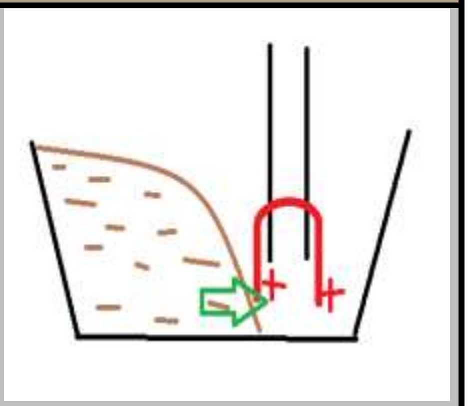
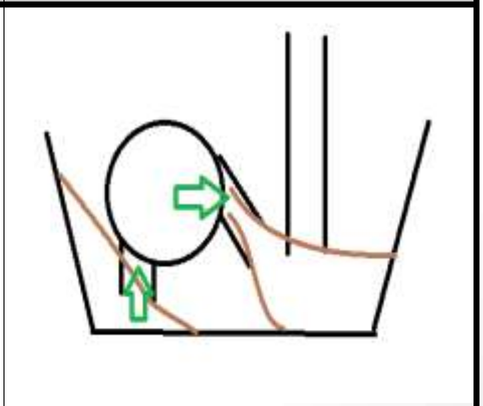
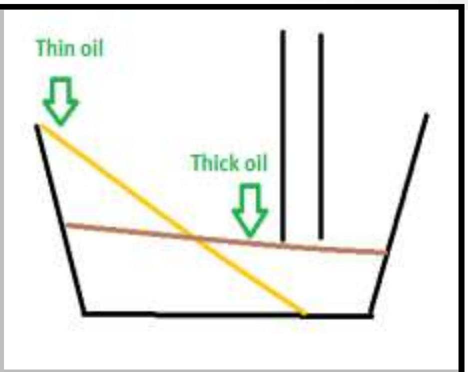
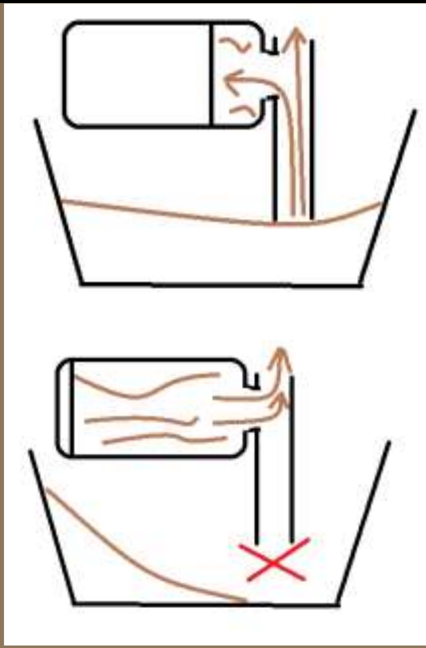
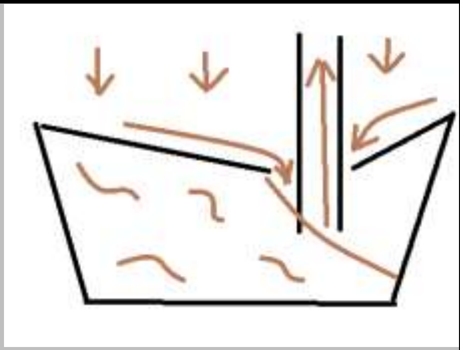
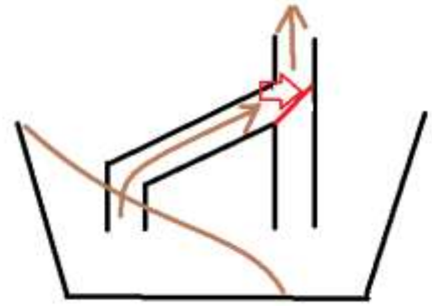
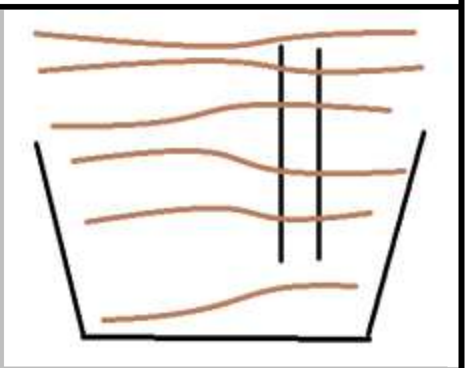
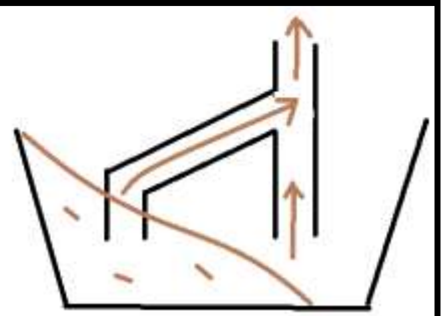
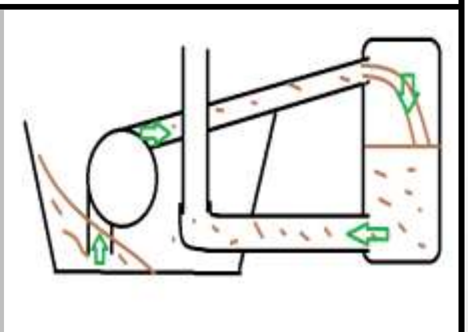
Appendix A – Preliminary Concept Designs

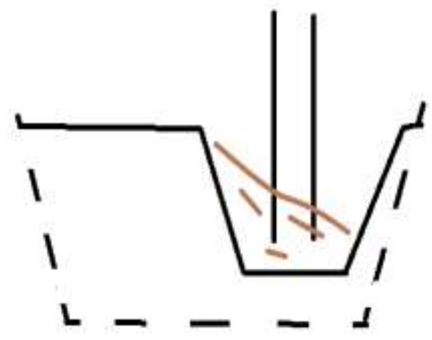
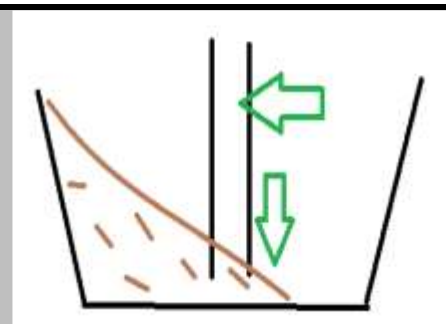
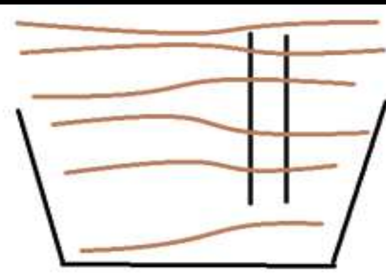
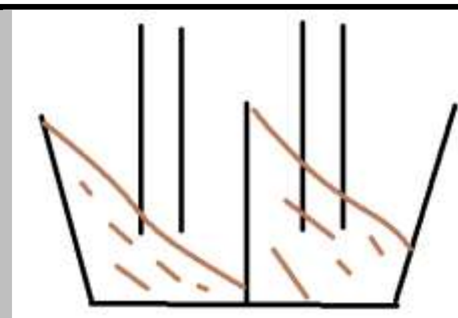
This appendix includes a feature-function-figure listing of initial concepts considered as solutions to the lubrication system. A full analysis of these designs is not included in this report however, all designs listed were analyzed and their various features were evaluated for their effectiveness in solving the scavenging problems with the UMFDT 2011 vehicle. The design presented in the main body of this report was design concept number four, and mas been highlighted for the reader to more easily identify.

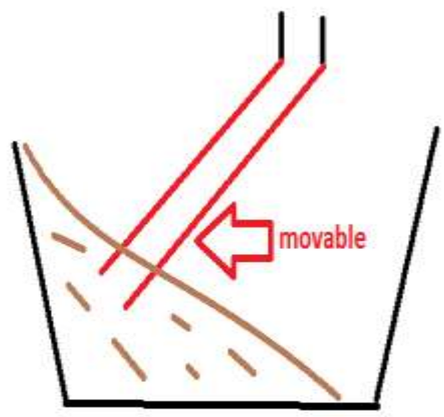
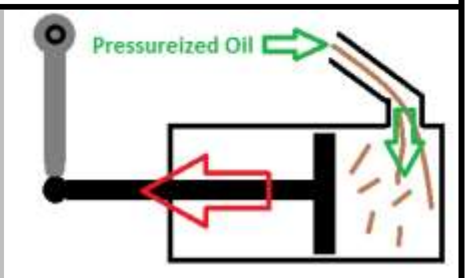
TABLE IV: FEATURE FUNCTION FIGURE OF PRELIMINARY CONCEPT DESIGNS

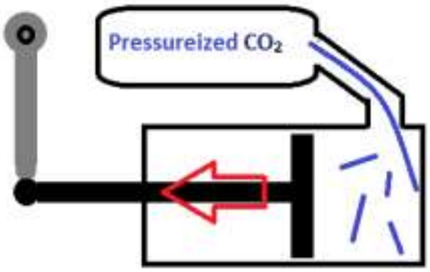
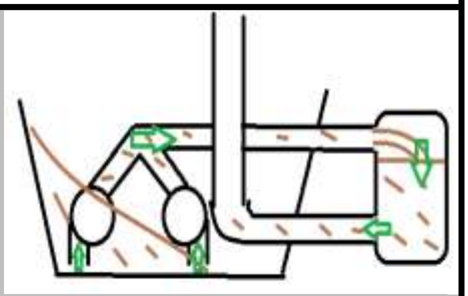
Concept	Feature	Function	Figure
1. Magnetic lube and magnetic oil pump	• No extra "handling" of the oil is required • Charged oil is attracted to magnetic pickup • Oil availability can be controlled by electric current	• Oil is attracted to pick-up by means of magnetism	
2. Pump oil from high spot to pick up	• Power supply for extra pump readily available • 2nd pump ensures oil pressure is not lost	• Pickup is always fed with a supply of oil	

3. Different viscosities of oil	• Reduces the oils readiness to slosh • Increases oil pressure due to thicker oil	• Current oil is replaced with one of a higher viscosity	
4. Oil accumulator in pressurized circuit	• Oil pressure is maintained • Accumulator stores excess oil at high pressure in order to supply the circuit should the pickup scavenge • When pump scavenges, accumulator automatically engages to maintain oil pressure	• Accumulator is attached to high pressure oil circuit • Accumulator after oil pump in circuit	
5. Optimize baffles in oil pan	• Force oil to remain where the pickup can access it • Allow oil to drain back from the engine easier	• Improvement of current system	

6. Hydraulic lines with a mechanical switch	• Contained all within the oil pan • Controlled by momentum of the car • Collects from both sides of the pan • Lateral force opens/closes switch	• Hydraulic lines and a momentum switch attached to pickup	
7. Reduce empty space in engine so oil can't slosh	• Forces oil to be within reach of pickup • Oil is always returned to space where it can be accessed by the pickup	• Fill engine full with oil or remove any open space	
8. 2 oil pick-ups	• Contained within oil pan • Collects from both sides of the pan • Oil is always accessible	• Insert 2nd oil pickup into oil pan	
9. Dry sump oil pickup	• Continuous supply of oil • Oil is gravity fed to engine • External pump	• External tank supplies internal oil pump • External pump sends oil into storage tank	

10. Reduce oil pan volume	• Reduced overall weight • Increased pressure • Increases space around outside of oil pan for external components • Directs flow and improves flow speed towards oil pick-up • Curved oil pan	• Oil volume to empty space ratio increased • Oil pickup is submersed deeper • Angled surface	
11. Optimize oil pan pick-up	• Easy to implement/install • Low cost	• Reduce the size of entry of oil pickup • Place pickup lower towards bottom of oil pan (increase pipe length & shave pegs on oil pan)	
12. Increase volume of oil	• Prevents loss of oil pressure	• Further submerses oil pickup in oil	
13. Two sumps	• When one sump scavenges the other supplies oil • Sumps located where oil will be • Redundant system at non-scavenging times	• Implement a second oil pump and pickup and spilt current sump into two portions	

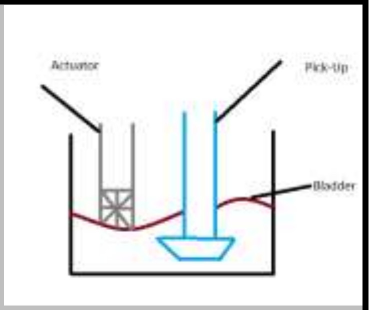
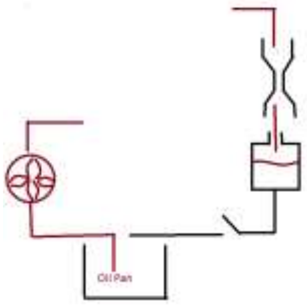
14. Hinged or double-hinged oil pick-up	• Vertical rudder allows360-degree rotation towards direction of oil slosh during lateral & longitudinal acceleration • Better reach towards area of concern (LH side) • Ease of oil strainer install • Rounded support leg improves reliability • Pickup always faces oil occupied area of oil pan	• L-shaped pipe connected via bearings to main (fixed) pipe • Vertical rudder soldered on top of horizontal pipe • Support leg on end of pipe reduces moment acting on bearing from weight of pipe itself and oil sloshing forces.	
15. Run shifting off pressurized oil	• Shift response time improves with engine RPM& its incompressible nature • No refilling required • Improved reliability over current air system • Weight reduction • Low cost	• Current shifting system tapped right after oil pump, and pressure is a function of RPM. • Unlimited supply • No regular maintenance • Replaces current 4 lb. air tank • Uses existing pressure lines & solenoid block reduces cost	

16. CO_2 Shifting System	• Fast response time • Small size • High force • Easy to route pressure lines	• Higher density than air • Solenoid activated • Similar to pneumatic system	
17. Dry Sump 2nd concept	• Can have two pick-ups & two smaller pumps (continuously running) to reduce storage tank volume (See logbook)	• Pick-up installed on each side of oil pan during continuous lateral force events (reduces oil tank volume)	

Appendix B – Additional Considered Concepts

After the final design for the lubrication system had been selected and approved by the UMFDT additional conceptual solutions were still generated. However, due to the time constraints of this project, these designs could not be fully evaluated as to their applicability to solving the scavenging issue. These designs are presented here as concepts for evaluation for the UMFDT as possible solutions in the event that the submitted solution is found lacking in any respect. The additional concepts are presented in Table IV in feature function-figure-format and described in more detail below.

TABLE V: FEATURE FUNCTION FIGURE OF ADDITIONAL CONSIDERED CONCEPTS

Concepts	Feature	Function	Figure
1. Actuator controlled bladder oil Pan	• Oil is forced to the pick-up by the force exerted on the bladder by the actuator	• Fully automatic • Prevents the pump from scavenging • Oil pressure is maintained • In the event of scavenging, the actuator automatically engages to maintain oil flow Relatively inexpensive	
2.Hybrid Wet-Dry Sump	• System either receives oil from the storage tank or from the oil pan	• Pressure is always maintained in the system • Reliable • When pump scavengers, the oil tanks provides oil Controlled automatically by switches	

Actuator controlled bladder:

This design incorporates a bladder type device into the oil pan of the vehicle. Fluid is allowed to fill the bladder through some type of one-way flow valve, and fluid is removed for engine lubrication through the normal oil pump (some modifications of the pump pickup may be required). When the vehicle goes around turns, the actuator is forced down by a sway-bar activated device, or some other non-fluid actuated compression device. The actuator effectively decreases the volume of the bladder, forcing the fluid towards the pump, allowing the pump to continue to draw fluid. The one-way flow valve prevents the fluid from flowing back the engine along the flow path used to drain the oil from the engine.

Hybrid wet-dry sump:

This system incorporates both the current wet sump used on the UMFDT vehicle and a dry sump. In this design the system either supplied with oil either from oil pan or storage tank. During normal operation, the wet sump supplies fluid to the dry sump tank through a very small orifice as well as the engine block. When scavenging occurs or every time pressure is lost, an electric switch is turned on to draw oil from the storage tank instead of the oil pan. This ensures the pressure inside the engine is maintained. The tank is slowly filled with oil under normal conditions through a small hole tapped into the high pressure galley. The oil is passed through a small orifice before dripping into the storage tank due to gravity.

Appendix C – Accumulator Selection

The general function of the accumulator is to store fluid for a later time, when there is a drop in fluid pressure. Many types of accumulators exist each having an advantage over others and vary depending on the system on which they are being implemented. There are six types of accumulators that can be used for hydraulic applications [3], [4].

1. Membrane-free accumulator
2. Gas charged bladder
3. Gas charged piston
4. Spring loaded piston
5. Weight loaded piston
6. Gas charged bellows

Identifying the type of accumulator best suited for the design requires a number of criteria to be taken into account. The main criteria include the response, weight, and reliability [4]. For motorsport applications, the system must react as soon as the engine begins to scavenge for oil while being as light as possible.

Response

The response time required for the accumulator will depend on the frequency of the pressure fluctuations of the system. From the data in Figure 11, the oil pressure responds almost instantaneously with the engine speed. During the race, engine speeds quickly rise and fall in a chaotic manner. As a result, the selected accumulator for this design must be able to quickly respond to these conditions while providing a sufficient flow rate during lubrication scavenging.

The response of an accumulator varies with the type of accumulator selected. Gas-charged systems offer faster response times over spring-loaded and weight-loaded systems. Spring-loaded and weight-loaded systems add additional mass to the piston contributing to inertial effects that must be taken into consideration especially for systems with flow that is continuously varying in pressure [4].

For gas-charged systems, bladder type accumulators offer faster response times over piston type systems as pistons generate a friction between the piston seals and cylinder walls. The slowest reacting gas-charged piston type accumulator is the gas-charged bellows. The additional weight and stiffness of the bellows prevent the accumulator from quickly responding to sudden changes in the fluid pressure.

Weight

The focus of the design is to implement a system that will prevent oil scavenging without adding a considerable amount of weight to the vehicle. For this design, the accumulator should be able to store as much oil relative to its overall size to efficiently occupy the amount of space provided within the confined boundaries of the chassis. Accumulators that operate with high volume ratios from a fully charged to fully discharged position will provide the best use of the limited space. For spring- loaded accumulators, the spring can only compress a distance that is less than its overall length. However, gas in charged accumulators can be compressed to one tenth of its original volume

Reliability

Accumulators endure repeated cycles of charging and discharging of fluid flow at various pressures. As a result, the reliability of an accumulator becomes an important factor in selecting the appropriate accumulator for the design. For piston-type accumulators, the seals can potentially fail leading to leaks between the gas chamber and the pressurized fluid. The life expectancy of the seal is dependent on the cycle rate of the piston movement, the compression ratio, as well as the amount of particulate matter in the fluid. Since the fluid passes through the oil filter before it reaches the accumulator, any particulates reaching the accumulator is minimized reducing the likelihood of premature seal wear. Gas charged in metallic bellows offers the best reliability from all piston-type accumulators because piston seals aren't required as the gas is enclosed within the bellows.

Mounting Position

The mounting position of the accumulator will greatly impact the selection of the accumulator. Some accumulators are specifically designed to be mounted vertically while others are not. For example, bladder type systems should be mounted vertically with the fluid feeding to the bottom. Horizontally-mounted bladder-type accumulators are subject to unequal bladder wear and are prone to trap fluid away from the outlet resulting in poor performance and a decrease in reliability [1]. Similarly, weight-loaded pistons and membrane free accumulators are designed to be vertically mounted.

For this project, the ideal location to mount the accumulator is to place it as close to the oil plug as possible to minimize viscous losses from requiring longer pipes. Directly in front of the oil plug, there is 12 inches of width of available space and 5 inches in height available. Consequently, it is the perfect location to mount the accumulator, but will require the accumulator to be mounted horizontally.

Additional Considerations to Accumulator Selection

To select the ideal accumulator, one must consider the application in which it is intended for. Depending on the orientation of the surrounding components in the system, there will likely be a constraint to the location of the accumulator installation, the mounting position, and the total volumetric size of the accumulator. In addition, the operating conditions of the system must be known to determine the design requirements of the accumulator which include the following.

- Nominal system pressure
- Minimum system pressure
- Pre-charge pressure
- Required flow rate
- Total fluid capacity
- Physical & chemical properties of the working fluid
- Recharge time
- Frequency of pressure changes
- System failure modes
- Range of fluid temperature and ambient temperature
- Pulsation damping
- Energy loss due to viscous effects by the pipes, fittings, valves, bends, and branches.

Accumulator Selection Process

Each accumulator type can be differentiated between one another but due to a limitation in the systems size and weight, only one can be selected for the design. In order to fully evaluate each type and select the best system for the design, a set of criteria were generated and used to rate each accumulator. The systems that are well suited for a specific criteria is labeled with a "+", while systems that are poorly suited are marked with a "-". Systems that are neither well suited nor poorly suited are marked with a "0". The Table V evaluates the six types of accumulators and ranks them in order to show the system that will likely be suited for the design.

TABLE VI: ACCUMULATOR SCREENING MATRIX

	Accumulator Screening Matrix					
	1	2	3	4	5	6
Selection Criteria	**Membrane Free**	**Gas Charged Bladder**	**Gas Charged Piston**	**Spring Loaded Piston**	**Weight Loaded Piston**	**Gas Charged Bellows**
Safety	+	+	+	+	+	+
Mounting Position	-	-	+	+	-	-
Cost	+	-	0	0	+	-
Size Relative to Capacity	+	-	+	-	+	-
Weight	+	+	+	0	-	0
Pressure Ratio	0	0	+	-	-	-
Ease of Tuning	-	+	+	-	+	+
Response	-	+	+	+	-	0
Reliability	+	+	0	+	+	+
Durability	+	0	0	+	+	+
Simplicity	+	+	+	+	+	-
Availability	+	+	+	+	+	-
Sum +'s	8	7	9	7	8	4
Sum -'s	3	3	0	3	4	6
Sum 0's	1	2	3	2	0	2
Net Score	**5**	**4**	**9**	**4**	**4**	**-2**
Rank	**5**	**2**	**6**	**2**	**2**	**1**

The screening matrix demonstrates that the gas-charged piston is well suited for the design. However, the table does not consider the importance of the criteria. As a result, a weighted value is assigned to each criteria based on their level of importance to the design and a scoring process is completed. Based on the values in the screening matrix, "-"values are identified as 0, "0" values are given a 0.5 score and "+" values are assigned a 1 value. These assigned values are multiplied with the weight value from each criterion and are given a numerical score. The scores are tallied for each accumulator and the system with the highest score will be selected for the design. The result of the scoring process is shown Table VI.

TABLE VII: ACCUMULATOR SCORING MATRIX

Accumulator Scoring Matrix													
Selection Criteria	Weight	1		2		3		4		5		6	
		Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score
Safety	17	1	17	1	17	1	17	1	17	1	17	1	17
Mounting Position	15	0	0	0	0	1	15	1	15	0	0	0	0
Cost	13	1	13	0	0	0.5	6.5	0.5	6.5	1	13	0	0
Size Relative to Capacity	12	1	12	0	0	1	12	0	0	1	12	0	0
Weight	11	1	11	1	11	1	11	0.5	5.5	0	0	0.5	5.5
Pressure Ratio	10	0.5	5	0.5	5	1	10	0	0	0	0	0	0
Ease of Tuning	7	0	0	1	7	1	7	0	0	1	7	1	7
Response	5	0	0	1	5	1	5	1	5	0	0	0.5	2.5
Reliability	4	1	4	1	4	0.5	2	1	4	1	4	1	4
Durability	3	1	3	0.5	1.5	0.5	1.5	1	3	1	3	1	3
Simplicity	2	1	2	1	2	1	2	1	2	1	2	0	0
Availability	1	1	1	1	1	1	1	1	1	1	1	0	0
Total Score		68.0		53.5		90.0		59.0		59.0		39.0	
Rank		5		2		6		3		3		1	
Continue?		N		N		Y		N		N		N	

From Table VI the type of accumulator that is best suited for the design is the gas-charged piston accumulator mainly because it can be horizontally mounted, requires less space, is lightweight, and can operate to match the pressure of the engine.

Appendix D – Discussion Assembly & Process

In the installation of the tubing and fittings, the following guides will be useful as reference. The high pressure lines required for the lubrication system will require flared tube fitting connections. This appendix explains the general installation procedure how to flare tubing and correct fitting procedures

Flare Tubing:

Tubes need to be flared before connected to fittings. The tools required for tube flaring are:

- A 45 degree double-flaring tool
- Tube cutters
- An inner/outer reamer
- Files
- Die block

STEP 1: Cut the tube using a tube cutter to the required dimension. Leave extra length for error.

STEP 2: Insert the compression fitting nut over the end of the tube. Insert the tube through the die block and clamp. It is important that tube is placed in the correct size hole.

STEP 3: Place the reamer on to the die block. Tighten the top handle of the reamer until the button die touches the die bar. It is important that the tool is not over tightened; this would result in cracking and therefore the failure of the button.

STEP 4: Smooth the rough edges using a round file.

STEP 5: Install flared tubing

Fittings Installation Steps:

STEP 1: Examine port and fitting to confirm that they are both clean of any contaminants.

STEP 2: Insert a stripe of an anaerobic liquid pipe sealant covering the male threads. Ensure the first two threads are uncovered.

STEP 3: Screw the fitting finger tight into the port.

STEP 4: Using a wrench, tighten the fitting to the correct Turns Past Finger Tight position (TPFT) see Table VII. When installing elbows or tees, consider the final orientation before tightening, as to not exceed the recommended TPFT. Properly installed fittings must engage between 3.5 to 6 threads [5].

TABLE VIII: TUBING TIGHTENING TABLE

Fitting Size Male	Dash Size	Turns Past Finger Tight	Torque ft-lbs
1/8 inch NPT	-02	1.5 - 3.0	12
1/4 inch NPT	-04	1.5 - 3.0	25
3/8 inch NPT	-06	1.5 - 3.0	40
1/2 inch NPT	-08	1.5 - 3.0	54
3/4 inch NPT	-12	1.5 - 3.0	78
1 inch NPT	-16	1.5 - 3.0	112

Maintenance

- It is important to routinely check the pre-charge pressure without any oil in the accumulator to test for any air leakage
- Avoid bleeding air out of the accumulator while the system is in operation as this would make the system ineffective
- Do not disassemble the system without proper care

Appendix E – Oil Pump Performance

In the design for the accumulator, the time it takes for the accumulator to fully deplete from a full charge will depend on the total oil capacity and the flow rate of the accumulator. Under normal engine operating conditions, the engine receives oil flow fed by the pump, but as the engine begins to scavenge for oil, there is a sudden drop in fluid flow and pressure which can ultimately lead to engine failure. While the engine speed, oil pressure, and other data are known, there is no sensor that directly reads the oil flow rate.

An alternative solution to determine the flow rate without completely disassembling the engine involved taking measurements of the pump's inner and outer gear rotors as well as the inlet and outlet port geometries. These measurements were recorded into a special computer application that theoretically calculates the pumps flow rate. For this analysis, a computer application used to determine the flow rate is called Gerotor Design Studio V2.0 [6]. A screenshot of the gear dimensions, port geometries, and fluid properties for 0W-40 synthetic oil are shown Figure 9.

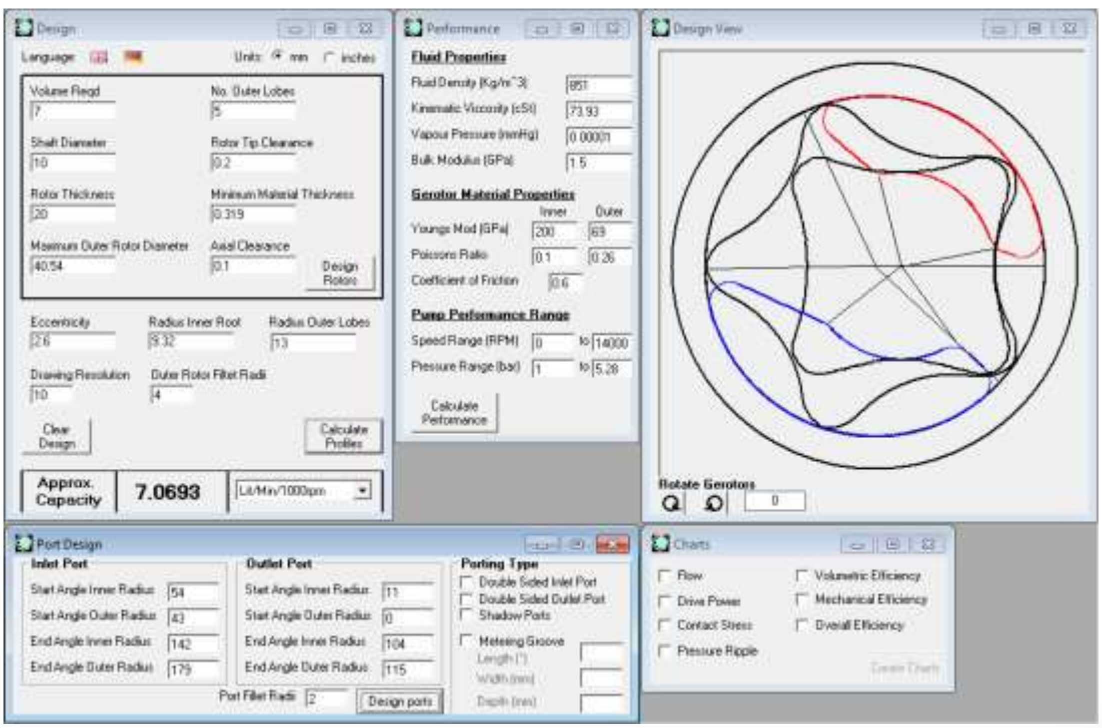


FIGURE 9: OIL PUMP GEAR AND ROTOR ANALYSIS

Based on the approximated results provided by the computer application, the oil pump produces 7.069 L/min per 1000 rpm of oil pump revolutions. However, there is a 1:2 ratio between the speed of the oil pump with the engine speed which results to a flow rate of 3.535 L/min per 1000 rpm of engine revolutions. Therefore, the theoretical pump performance curve will be linear if we neglect the effects of internal leakage and viscous losses at higher flow velocities.

From the data from Figure 11, the oil pressure rises with increasing engine RPM but they system reaches its maximum pressure at approximately 4500 engine rpm. The additional flow pumped at higher engine speeds is forced back out of the bypass valve leading back into the sump. By extrapolation, the oil pump theoretically pumps at 3.535 L/min while at 4500 rpm. From the plot Figure 11, the engine scavenges for a maximum duration of 6 seconds. During this time, the engine would have pumped 0.354 L of oil. In order for the lubrication system to operate as if scavenging had never occurred, the accumulator must pump oil at the same average flow rate of 3.535 L/min which is the target flow rate for the accumulator design.

Appendix F – Flow Rate Analysis

To be able to determine the proper dimensions for the accumulator, it is necessary to understand the effects of the frictional energy losses that link the accumulator to the engine. The design of the accumulator system relies on what will be bridging the flow of oil between the engine and the accumulator. The pipe network system that connects the accumulator to the engine composes of different devices and fittings that all affect the overall performance of the accumulator. A schematic of the pipe network is shown in Figure 10.

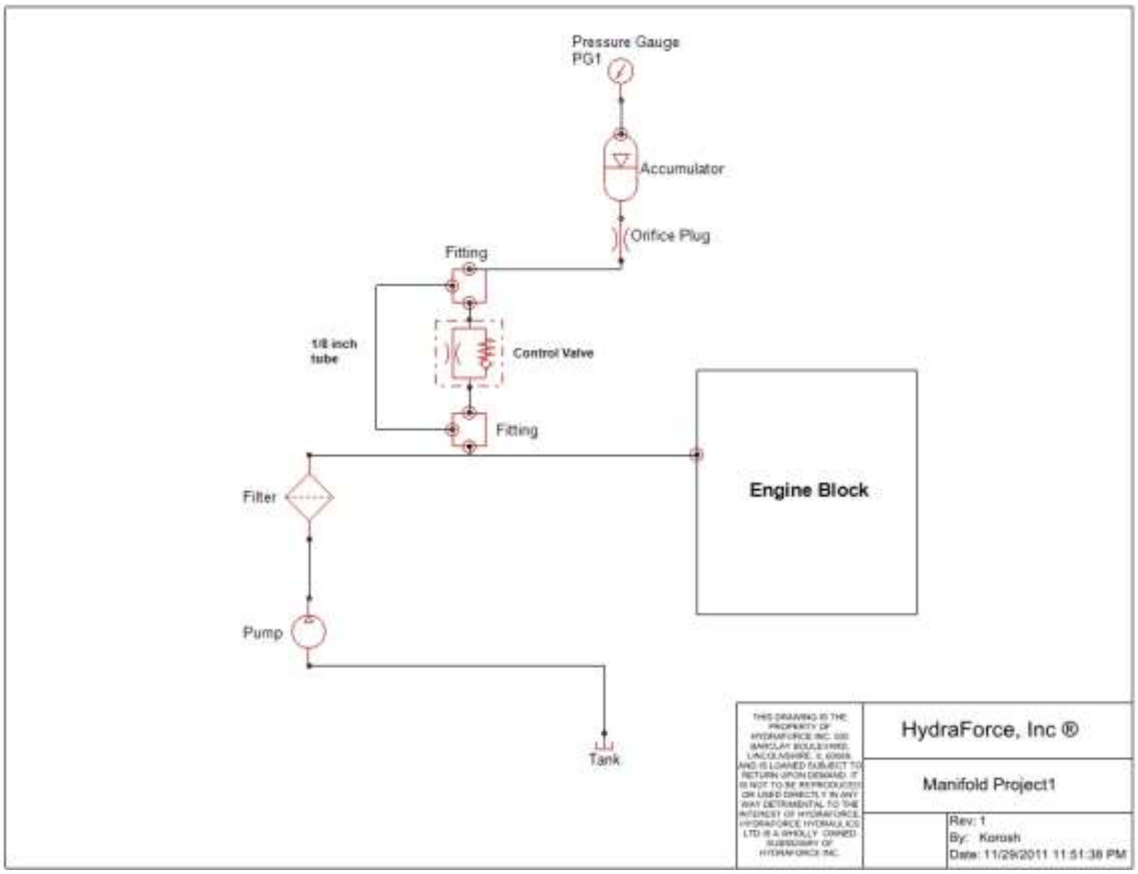


FIGURE 10: LUBRICATION SCHEMATIC

During the charging process of the accumulator, the flow of oil passes through a (1/4 and 1/8)inch NPT fitting on the engine block through 1/4 inch diameter copper pipe that connects two standard T-fittings, a 90° bend, and an L-fitting leading into the accumulator. However, the flow of oil in the reverse direction is prevented by a one-way check valve that allows the flow to move in another pipe that is 1/8 inch diameter with a 90° bend in between connected to the standard T-fittings. There are additional fittings convert the 1/8 inch pipe to connect on the 1/4 inch T-fittings.

The complexity of the pipe network will directly affect pressure losses (or head loss) in the system. Each branch, valve, and bend in the system contributes to a certain degree of head loss. Consequently, the head loss in the system will determine the minimum pre-charge pressure in the accumulator. The minimum pre-charge must overcome the head losses and piston seal friction otherwise there will be no flow. As a result, the team aims to reduce the effects of head loss as much as possible in order to allow a larger pressure ratio to be used. Larger pressure ratios allow more of the total accumulator volume to be used as available oil capacity fully utilizing the limited space available. Therefore, the pipe network design is very important as it directly affects the accumulator's initial pre-charge pressure and overall size and weight.

The head loss equation can be derived based on the principle of energy conservation as the fluid flows from one point to another and can be expressed using Bernoulli's, equation (1) [7].

$$\frac{P_1}{\rho} + \frac{\bar{V}_1^{\,2}}{2} + gZ_1 = \frac{P_2}{\rho} + \frac{\bar{V}_2^{\,2}}{2} + gZ_2 + h \qquad (1)$$

Where P is the fluid pressure, V is the fluid velocity, g is the gravitational constant, Z is the elevation, and h is the total head loss. The subscripts 1 and 2 refer to the accumulator and engine respectively

The total head loss in the system can be considered as two separate categories: major and minor losses. Major head loss consist of the energy lost based on the overall dimensions of the pipes as a whole, while minor losses take account of the obstacles along the way. In either case, the material and geometric properties of the pipe, the velocity of the flow, the kinematic viscosity of the fluid, and the additional devices installed all contribute to the amount of head loss in the system. The list of loss coefficients for each device in the system are listed in Table VIII [7].

TABLE IX: HEADLOSS COEFFICIENTS FOR PROPOSED LUBRICATION SYSTEM

Head Loss Coefficients				
Major	1/4" pipe:	Length,	L =	0.19685 [in]
		Diameter,	D =	0.25 [in]
		Roughness,	*e* =	0.00006 [in]
	1/8" pipe:	Length,	L =	0.19685 [in]
		Diameter,	D =	0.125 [in]
		Roughness,	*e* =	0.00006 [in]
Minor	Valve	Angle Lift Valve,	L_e/D =	55
	Accumulator	Expansion,	K_{ea} =	1
		Contraction,	K_{ca} =	0.5
	Standard Tee	Expansion through Branch,	L_e/D =	0.5
		Contraction through Branch,	L_e/D =	0.25
	Standard Tee	Flow through run,	L_e/D =	20
		Flow through branch,	L_e/D =	60
	Pipe Bends	Curved 90° Bends with 1cm radius	L_e/D =	13
	Flanged Elbows	90° Elbow	L_e/D =	60

L/D and L_e/D are the head losses due to the system lubrication system. These losses have been calculated for the system schematics provided in this report. However the head loss coefficients would need to be recalculated if the system geometry is altered.

TABLE X: HEAD LOSS TOTALS, CHARGING AND DISCHARGING

Sum of Head Loss Coefficients			
Case 1 Charge	Major	L/D =	0.787
	Minor	L_e/D =	209
	Total	209.787	
Case 2 Discharge	Major	L/D =	2.362
	Minor	L_e/D =	154.25
	Total	156.612	

Since the system follows a different path between charging and discharging processes, the head loss for each case will be different, see Table IX. The equations for the major and minor head losses for each case are shown below [7].

Case 1: Accumulator Charging Process

$$h = \sum h_l + \sum h_{lm}$$

$$h = f\frac{L}{D}\frac{\bar{V}_2^{\,2}}{2} + \sum\left(f\frac{L_e}{D}\frac{\bar{V}_2^{\,2}}{2}\right)_{Tees} + \left(f\frac{L_e}{D}\frac{\bar{V}_2^{\,2}}{2}\right)_{Valve} + \sum\left(f\frac{L_e}{D}\frac{\bar{V}_2^{\,2}}{2}\right)_{Bends} + \sum\left(K\frac{\bar{V}_2^{\,2}}{2}\right)_{Fittings}$$

Where f is the friction factor, L is the length of the pipe, D is the diameter, $\bar{V}$ is the fluid velocity, L_e is the equivalent length, and K is the loss coefficient.

$$h_l + \sum h_{lm} = \frac{\bar{V}_2^{\,2}}{2}(208.787f + 1) \qquad (2)$$

The velocity $\bar{V}_2$ can be expressed in terms of $\bar{V}_1$ based on the continuity equation.

$$\dot{m}_1 = \dot{m}_2$$

$$\rho\dot{Q}_1 = \rho\dot{Q}_2$$

$$A_1V_1 = A_2V_2$$

$$\therefore \quad V_2 = \frac{A_1}{A_2}V_1$$

Substituting equation *(3)* into *(2)*, then substituting the result into *(1)*, and solving for $\bar{V}_1$

$$\bar{V}_1 = \sqrt{\frac{2(P_2 - P_1)}{\rho\left[\left(\frac{A_1}{A_2}\right)^2 + 208.787f\right]}} \qquad (3)$$

Case 2: Accumulator Discharging Process

$$h = \left(f\frac{L}{D}\frac{\bar{V}_2^{\,2}}{2}\right)_{1/4"} + \left(f\frac{L}{D}\frac{\bar{V}_2^{\,2}}{2}\right)_{1/8"} + \left(f\frac{L_e}{D}\frac{\bar{V}_2^{\,2}}{2}\right)_{Elbow} + \ldots$$

$$\ldots + \sum\left(f\frac{L_e}{D}\frac{\bar{V}_2^{\,2}}{2}\right)_{Tees} + \left(f\frac{L_e}{D}\frac{\bar{V}_2^{\,2}}{2}\right)_{Bend} + \sum\left(K\frac{\bar{V}_2^{\,2}}{2}\right)_{Fittings} \quad (4)$$

Where f is the friction factor, L is the length of the pipe, D is the diameter, $\bar{V}$ is the fluid velocity, L_e is the equivalent length, and K is the loss coefficient.

$$h_l + \sum h_{lm} = \frac{\bar{V}^2}{2}(156.112f + 0.5) \quad (5)$$

Substituting *(3)* into *(4)*, substituting the result in *(1)*, and solving for $\bar{V}_1$

$$\bar{V}_1 = \sqrt{\frac{2(P_2 - P_1)}{\rho\left[1.5 - \left(\frac{A_1}{A_2}\right)^2 + 156.112f\right]}} \quad (6)$$

As the friction factor f is a function that relies on the velocity $\bar{V}_1$, equations *(4)* and *(6)* requires iterative calculations to obtain an accurate result. Using the final result for the fluid velocity after the iterative process, the flow rate inside the accumulator can be determined.

$$\dot{Q} = A_1\bar{V}_1$$

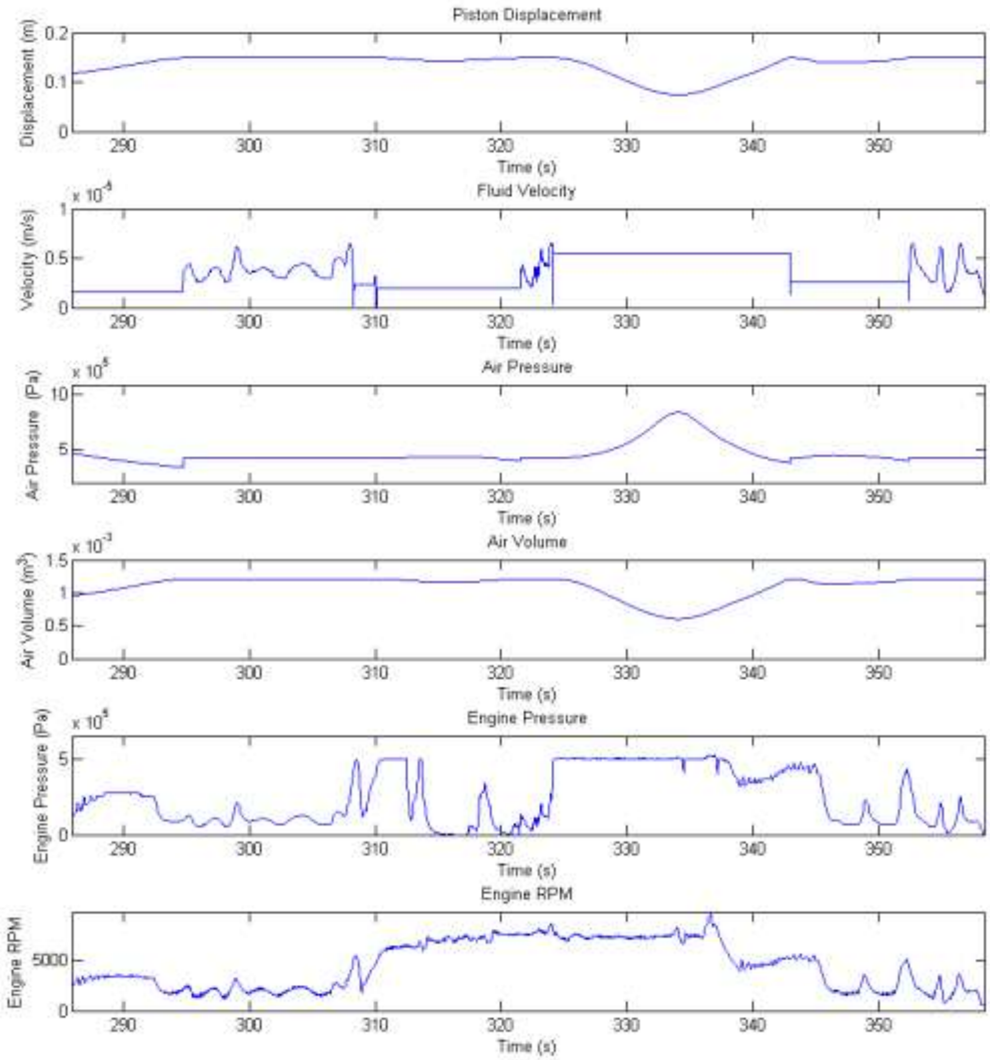


FIGURE 11: SAMPLE MATLAB OUTPUT

Figure 11, shows sample output from the MATLAB written program.

The computational results from the MATLAB software show that the piston displaces from its initial displacement of 0.1504 m to a minimum of 0.0744 m consuming 50.5% of the 1 L oil capacity.

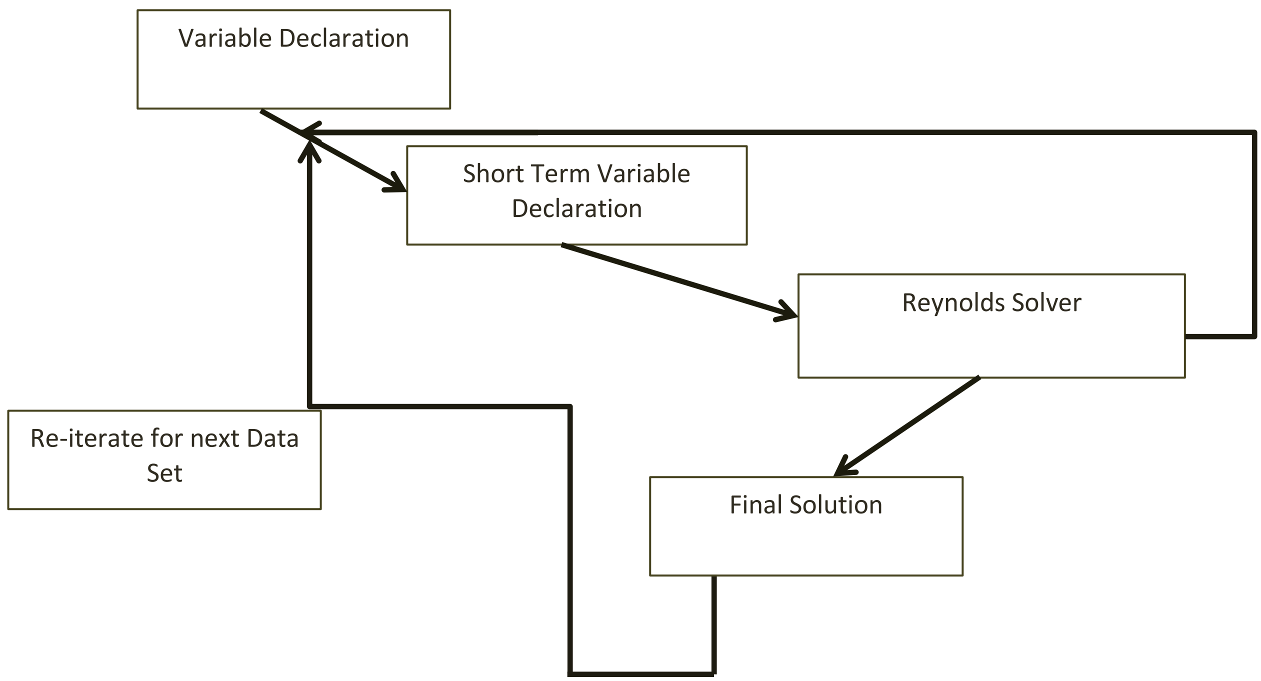
The included MATLAB code incorporates the experimental data provided by the UMFDT and the theoretical flow calculations for head loss, momentum, pressure distribution and continuity. The code provides an intermediary step between the model of the lubrication system, and the experimental data. The program provided is not meant to replace real world experimental verification, but rather to serve as a proof of concept relating experimental data with the theoretical models of the lubrication system additions.

The algorithm used by the MATLAB code is as follows:

1. Data is read in to the program from the experimental data provided.
2. The order of the data is then reversed so that the data set begins at interval 1 and increases rather as time increases rather than decreasing.
3. Assumed and calculated variables are then declared for use in the program. Note if these variables are to be altered, MASS, MaxPressure, MinPressure, MaxDisplacement, MinDispacement, PistonDisplacement and PISTONAREA are all interrelated and their interrelation needs to be considered when changing one of the vaiables. TimeStep is the variable associated with the sample frequency of the data set, if a different data set is used, this variable will need to be set accordingly. TEMPERATURE is assumed to be constant over the course of a single time step in all following calculations. MAXERROR is the variable associated with the acceptable difference between the calculated Reynolds number, more accurate results would be possible with a smaller MAXERROR, however due to the simplifications of this numerical model, more accurately calculated Reynolds numbers would not likely result in more accurate system results. Relax is the relaxation number associated with the numerical calculations performed.
4. The flow direction is determined from comparing the pressure on the engine side of the system with the accumulator side.
5. The ErrorCheck variable is set/reset for the Reynolds number solver
6. The Reynolds number is set/reset for the Reynolds number solver
7. Velocity variable is set/reset
8. Reynolds number Solver begins loop
9. Friction factor is calculated using the Reynolds number set in step 6.
10. Depending on the flow direction determined in step 4. Bernoullis equation is solved accordingly
11. The velocity using the Vprime is calculated using the relaxation variable
12. Vprime is set as the most recently calculated Velocity
13. Using VPrime, the new Reynolds number is calculated
14. The error between the New Reynolds number and the ‘guessed’ Reynolds number is determined
15. If the error is less than MAXERROR the two Reynolds numbers are sufficiently close and the Reynolds number solver ends otherwise go to step 9.

16. Using the velocity determined from the Reynolds solver, the velocity of the accumulator actuator is determined, DeltaX
17. The change in displacement of the accumulator membrane is determined using DeltaX
18. The new volume of the air chamber of the accumulator is determined
19. The pressure of the air chamber is calculated using the new volume
20. Because of continuity, the pressure of the air chamber is the same as the pressure of the oil. The oil pressure for the next iteration is set to be the pressure calculated in this step.
21. Proceed to step 4 until the data set has been completed.

A flow chart for the program is provided below



The MATLAB code using this algorithm is provided here.

```
data= xlsread('skid_pad_kyle_NO_Headers.xls');

TempTime = data(:,3);
TempRPM = data(:,4);
TempP2 = data(:,10);

for i=1800:length(TempTime)
time(i,1) = TempTime(length(TempTime)-i+1,1);
P2(i,1) = TempP2(length(TempTime)-i+1,1);
RPM(i,1) = TempRPM(length(TempTime)-i+1,1);
end
RPM(35851,1)=0;
P2 = P2*14.8535564854*6895;

MASS = 0.004744576; %kg
TimeStep = 0.01;% seconds

MaxPressure = 618431.8;%Pascals
MinPressure = 194404.22;%Pascals

MinDisplacement = .027075892;%m
MaxDisplacement = MinDisplacement+.1233453;%m
PipeDiameter = 0.00635;% m

PistonDiameter = 0.1016;%m
PipeLength = 0.3048;

PISTONAREA = 3.141592653* (PistonDiameter^2)/4;
PIPEAREA = 3.141592653* (PipeDiameter^2)/4;
TEMPERATURE = 373; %Degrees Kelvin
R = 287;

e_val = .0000015; %m
nu = 1.75*10^-5; %m^2 per sec
rho= 851; %kg/m^3

MAXERROR = 10; % this value set how big the difference in the pressure
differential can be considered acceptable
eoverD = e_val/PipeDiameter;

P1(1,1) = MinPressure;% sets the initial pressure of the accumulator at
the minimum possible pressure
Displacement(1,1) = MaxDisplacement;
V1plot(1,1) = 0;
Volume(1,1) = Displacement(1,1) * PISTONAREA;
t(1,1)=TimeStep;

Relax =.8;
```

```
for i= 1:length(time)

    FlowDir = sign(P2(i,1)-P1(i,1)); %determines the direction of the
fluid flow dependant on the pressures
    ErrorCheck = 100;%These values are initiailized at one so that the
solver can begin.
   % VerrorCheck =1;%These values are initiailized at one so that the
solver can begin.

    % these values are used to initiate the Reynolds Interation Loop
    % assumes turbulent flow
    LastReynolds = 20000;
    LastV1 = 0;

        % Reynolds Solving loop
        while ErrorCheck >=MAXERROR %&& VerrorCheck >= MAXERROR

% f = 64 /LastReynolds;
            f =(0.25*(log10((eoverD/3.7) + (5.74/LastReynolds^0.9)))^(-
2));
%              f = (-2*log10((e_val/(3.7*PipeDiameter))-
(5.02/LastReynolds)*log10((e_val/(3.7*PipeDiameter))-
(5.02/LastReynolds)*log10((e_val/(3.7*PipeDiameter))+(13/LastReynolds))))
)^(-2);
Alternative to Colebrook White Equations
%
            % determines velocity flow in pipe
            if FlowDir == -1
            V1 = sqrt((2* (P2(i,1) -
P1(i,1))/rho)/((PISTONAREA^2/PIPEAREA^2)-.25+204*f));
            elseif FlowDir ==1
                %
            V1 = sqrt((2* (P2(i,1) - P1(i,1))/rho)/(2 -
(PISTONAREA^2/PIPEAREA^2) +203*f));
            else

            end
            Vprime =  Relax * LastV1 + (1-Relax)*V1;
            LastV1 = Vprime;
            NewReynolds = (LastV1 * PipeDiameter)/nu;% Re-calculates
Reynolds number based on the velocity determined above
            ErrorCheck = abs(LastReynolds - NewReynolds) % determines the
magnitude of the difference between in the last two iterations
            LastReynolds = NewReynolds;% Used to 'remember' the last
calculated reynolds number

        end% end of Reynolds Solving loop
    V1plot(i+1,1) = LastV1;
    DeltaX = LastV1*TimeStep*FlowDir;% determines the amount of change in
the displacement of the piston due to the pressure difference
```

```
    Displacement(i+1,1) = Displacement(i,1) - DeltaX; %calculates the new
displacement


    %this ladder determines if the piston is within the acceptible ste
            if Displacement(i+1,1) <= MinDisplacement
                Displacement(i+1,1) = MinDisplacement;
            elseif Displacement(i+1,1) >= MaxDisplacement
                Displacement(i+1,1) = MaxDisplacement;
            else
               % Displacement is within the accumulator stops
            end

     Volume(i+1,1) = Displacement(i+1,1) * PISTONAREA;
     P1(i+1,1) = (MASS *R *TEMPERATURE)/Volume(i,1);

     t(i+1,1)=(i+1)*TimeStep;

end
```

Appendix G – Shifting Calculations

Calculations for the system requirements began by obtaining measurements from current system in use. Measurements were made for gear selection and clutch actuation. The clutch cylinder was found to require more force to actuate and is therefore set as the limiting factor for which all calculations are based. The first item measured was the torque required for full activation of the clutch. Using the linkage off set the normal force is calculated.

Measured clutch shaft torque (T): 75 in-lbs

Measured linkage offset (d): 1 in

Normal Force required (F_{NR}):

$$\mathrm{F}_{\mathrm{N}_R} = \frac{T}{d}$$

$$F_{N_I} = \frac{75}{1} = 75\ \mathrm{lb}s$$

The force available from a hydraulic cylinder is calculated by multiplying the oil pressure by the cylinder cross-sectional area.

Oil Pressure Unaltered (P_U): 75 psi

Clutch Cylinder Area (A): 0.875 in^2

Normal Force Unaltered (F_{NU}):

$$F_{N_U} = P_U A$$

$$F_{N_I} = 75 \cdot 0.875 = 65.625\ \mathrm{lbs}$$

Since the current force is less than the force required, the pressure or area will need to be increased, as the space requirements cannot be altered. The previous system used a 100psi pneumatic system; this pressure is used for a comparative analysis.

Oil Pressure Increased (P_I): 100 psi

Normal Force Increased (F_{NI}):

$$F_{N_I} = P_I A$$

$$F_{N_I} = 100 \cdot 0.875 = 87.5 \text{ lbs}$$

From the last calculation, the force input is now enough to fully actuate the clutch. The next step calculates the engine power required to increase the pressure. The oil pump flow rate was calculated from measurements of the pump and the input speed. Calculations for pump flow rate can be found in Appendix E.

Oil Flow Rate: $7\frac{L/min}{1000\ rpm} = 1.56\frac{Gallon/min}{1000\ rpm}$

Hydraulic power conversion factor: 1714

Power Consumption Increase is the power draw due to the change in pressure of the lubrication system. This draw is calculated by finding the product of the system pressure and flow rate, divided by 1714 for the specific imperial units, gallons per minute and psi.

Power Consumption Increase (PCI):

$$PCI = \frac{(P_I - P_U)Q}{1714}$$

$$PCI = \frac{(100 - 75)1.56}{1714} = 0.0227 \left[\frac{HP}{1000\ rpm}\right]$$

The hydraulic efficiency of fluid through a valve is approximately 85%. This efficiency can then be used to find the actual increase in power required.

Hydraulic Horsepower Efficiency (η) – 0.85

Horse Power Increase Required (HP_R):

$$HP_R = \frac{HP}{\eta}$$

$$HP_R = \frac{0.0227}{0.85} = 0.0267 \left[\frac{HP}{1000\ rpm}\right]$$

Once the required power rate is known, the maximum power required can be calculated. Multiplying the power required by the maximum engine speed determines the additional power needed.

Total Power Increase at Maximum RPM (HP_{Tot}):

$$HP_{Tot} = HP_R \cdot 12$$

$$HP_{Tot} = 0.0267 \cdot 12 = 0.32\ HP$$

this calculation shows that 0.32 HP is required at maximum rpm to enable the lubrication powered shifting system.

Appendix H – Cost Analysis

Additional accessories are available for the Accusump manufactured by Canton. A manual valve could be added to the system making possible to transport the accumulator while it is charged. In case design modifications been made and the Accusump would be hard to reach, cable kit can be used. Another type of valve could be considered is the E.P.C electric valve. This allows the accumulator to be mounted almost anywhere in case space is limited in future designs. Another advantage of this system is it will allow the Accusump to discharge at a pre-set engine pressure only and refill when the engine pressure increases above that set level. The electric valve wires to the ignition directly resulting in automatic operation of the Accusump. Table X lists all the possible accessories could be added to the system.

Table xi: Optional Components

Item #	Description	Quantity	Part Number	Vendor	Specifications	Cost
1	Accusump EPC VALVE	1	24-275	Canton Racing	Discharge/Refill of 55-60	242.00 USD
2	Accusump mounting clamps 1QT	2	24-240	Canton Racing	1/4 inch NPT	20.40 USD
3	Accusump Actuator Cable Kit	1	24-506	Canton Racing	6 FT	90 USD
4	Accusump Check Valve	1	24-280	Canton Racing	1/2 inch NPT	27.60 USD
5	Liquid filled SS Gauge	1	24-500	Canton Racing	0-160 PSI	37.20 USD

Detailed Cost Analysis

All the materials and parts considered for this project selected took into account the UMFDT's financial resources available. Vendors distributing the parts were selected considering variables such as shipping times and charges, customs charges and customer support. In the determination of the overall cost, the capabilities of the UMFDT to manufacture basic components were taken into consideration. The detailed final cost breakdown of the lubrication system design is provided in Table XI.

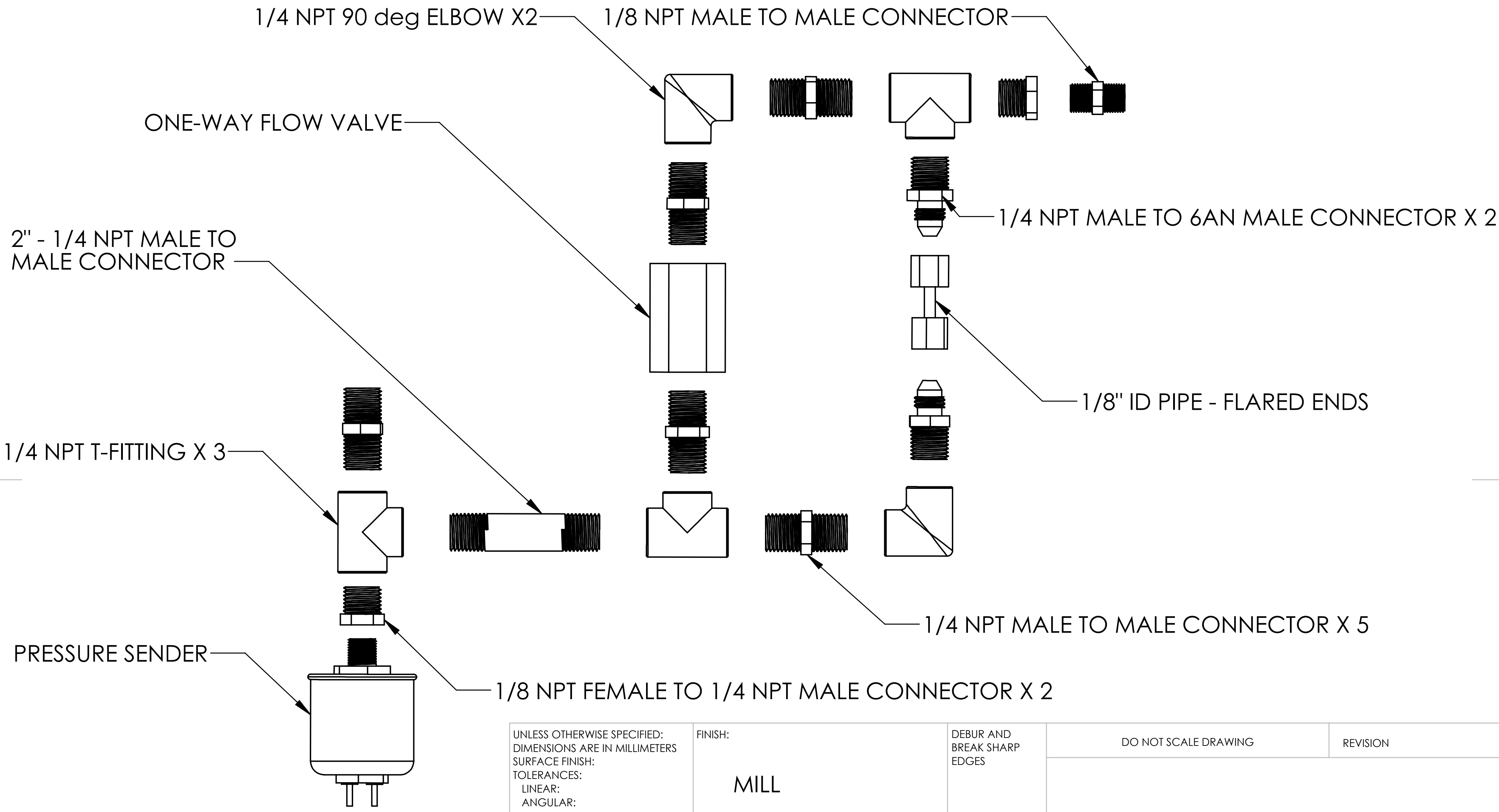
A breakeven analysis is usually performed to determine when a product will pay for itself. However since the intended purpose of the race vehicle is not for production such an analysis is not performed on the design presented. The UMFDT is a non-profit organization and such an analysis is deemed unnecessary since the objective of the team not monetary in nature but rather the production of a reliable race vehicle for SAE competitions.

TABLE XII: COST ANALYSIS

Item #	Description	Quantity	Part Number	Vendor	Specifications	Cost
1	1 Quart Accusump	1	24-046	Canton Racing	12 inch long and 3-1/4 inch D	150 USD
2	Union Tee	2	A-400-3	Swagelok	1/4 inch NPT	18.00 CAD
3	Tube Fitting	2	B-400-7-2	Swagelok	1/4 x 1/8 inch	3.49 CAD
4	SS Hose Clamp	2	8125925	Princess Auto	3-1/2 inch	1.49 CAD
5	Check Valve	1	24-280	Canton Racing	1/2 inch NPT	27.60 USD
6	The manual control ball valve (Optional)	1	24-260	Canton Racing	1/2 inch NPT	15.60 USD
7	Accusump mounting clamps (Optional)	2	24-240	Canton Racing	1-QT-3-1/4 inch	20.40 USD
8	Alloy 360 Brass Tube	1	BRR18	Metals Depot	1/8 inch- 4 Ft.	5.40 USD
9	Alloy 360 Brass Tube	1	BRR14	Metals Depot	1/4 inch-4 Ft.	9.64 USD
10	Accusump EPC VALVE	1	24-275	Canton Racing	Discharge/Refill of 55-60	242.00 USD
11	Accusump mounting clamps 1QT	2	24-240	Canton Racing	1/4 inch NPT	20.40 USD
12	Accusump Actuator Cable Kit	1	24-506	Canton Racing	6 Ft.	90 USD
13	Accusump Check Valve	1	24-280	Canton Racing	1/2 inch NPT	27.60 USD
14	Liquid filled SS Gauge	1	24-500	Canton Racing	0-160 PSI	37.20 USD
15	Oil Pan	1	-	-	-	0 CAD
16	Pump	1	-	-	-	0 CAD

Appendix I – CAD Models

CAD Models presented in this section have been created with SolidWorks 2011 Student edition made available by the University of Manitoba chapter of SAE international of which the UMFDT is affiliated with.



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH: MILL			DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
	NAME	SIGNATURE	DATE		TITLE: PIPE FITTING ASSEMBLY	
DRAWN						
CHK'D						
APPV'D						
MFG						
Q.A				MATERIAL:	DWG NO. FSAEX-XXX-EN-A0002	A4
				WEIGHT:	SCALE:1:2	SHEET 1 OF 1

SolidWorks Student Edition.
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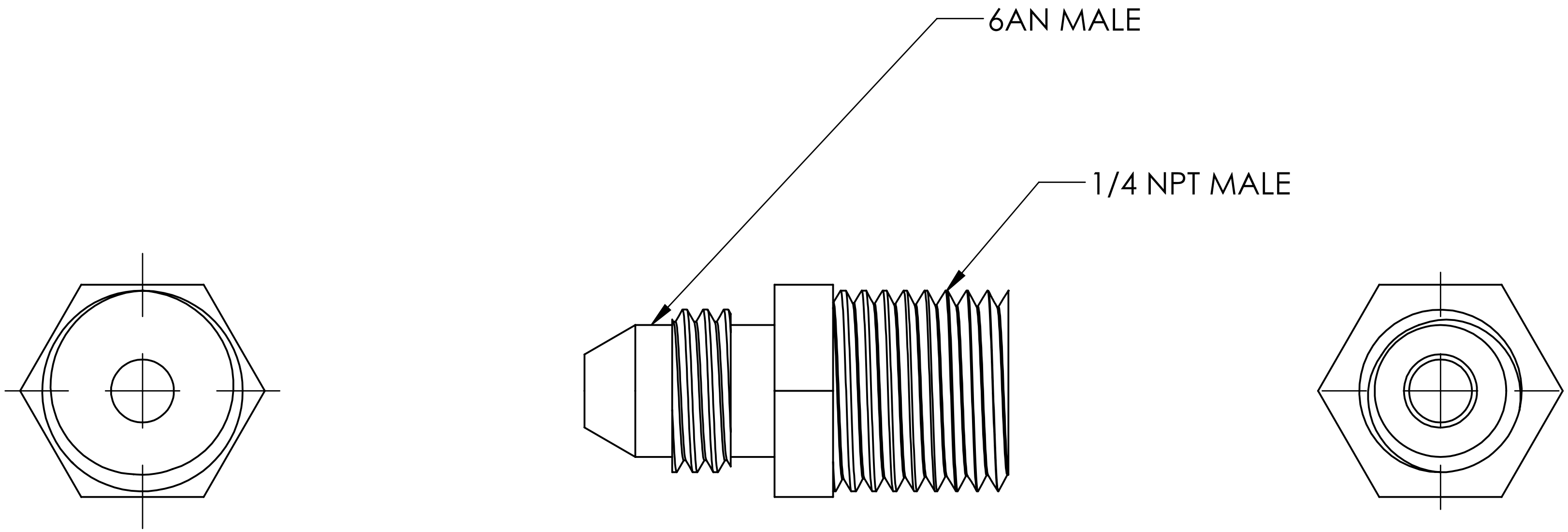

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REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE: 1/4 NPT MALE - 6AN MALE CONNECTOR

MATERIAL: BRASS

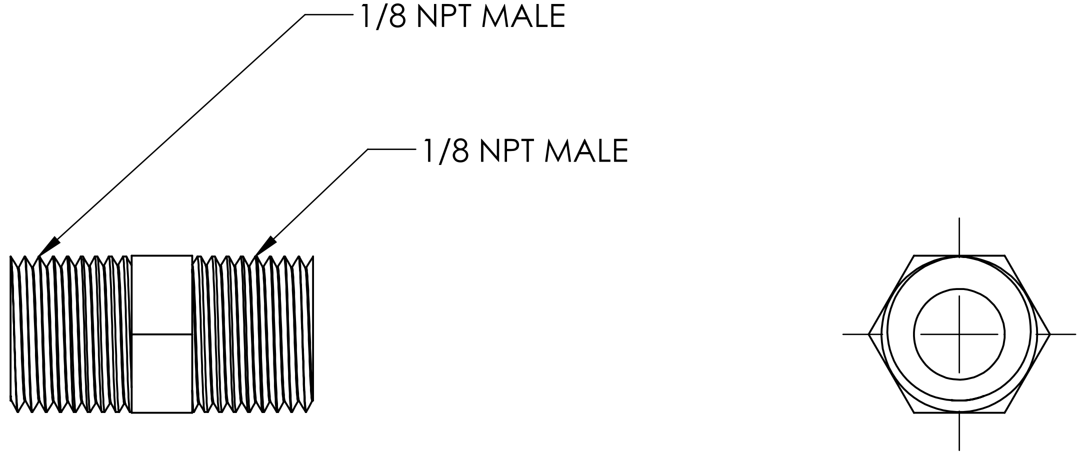
WEIGHT:

DWG NO. FSAEX-XXX-EN-00012

A4

SCALE:2:1

SHEET 1 OF 1



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			REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

1/8 NPT MALE - MALE CONNECTOR

MATERIAL:
BRASS

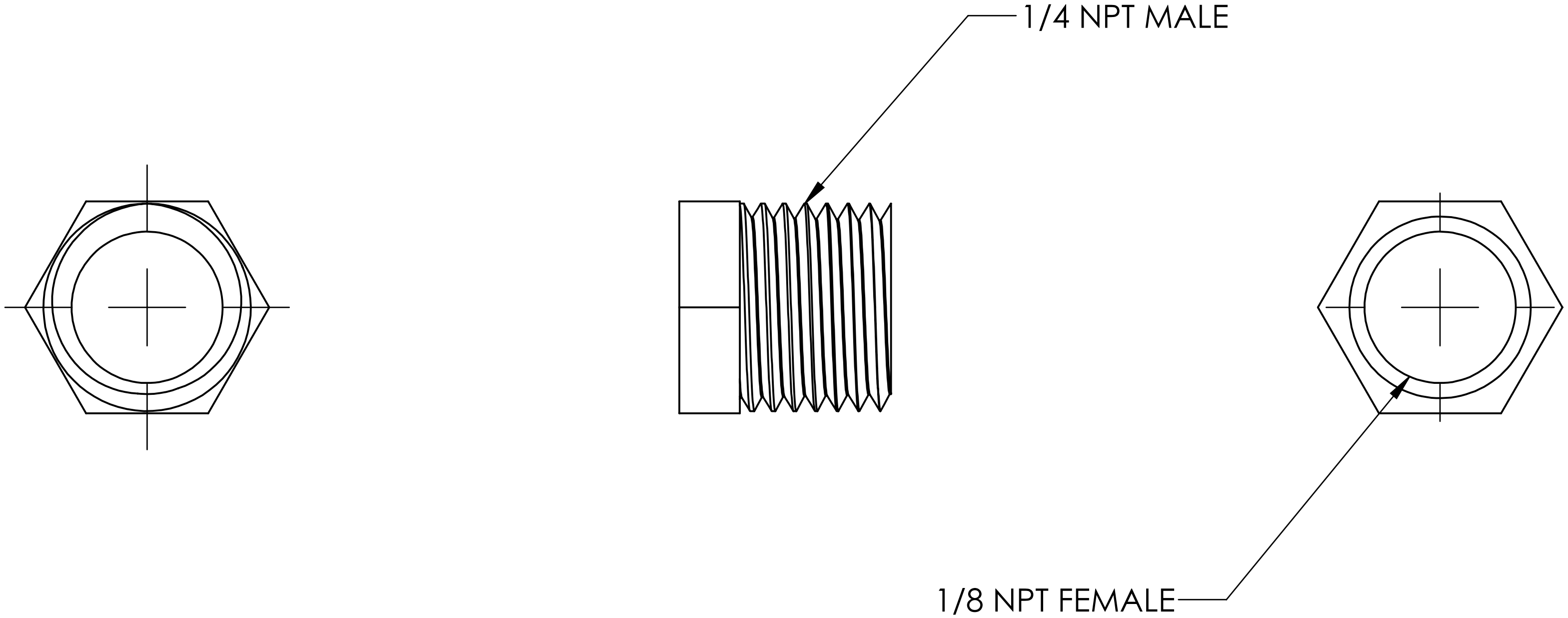
DWG NO. FSAEX-XXX-EN-00013

A4

WEIGHT:

SCALE:2:1

SHEET 1 OF 1



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REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL: BRASS

WEIGHT:

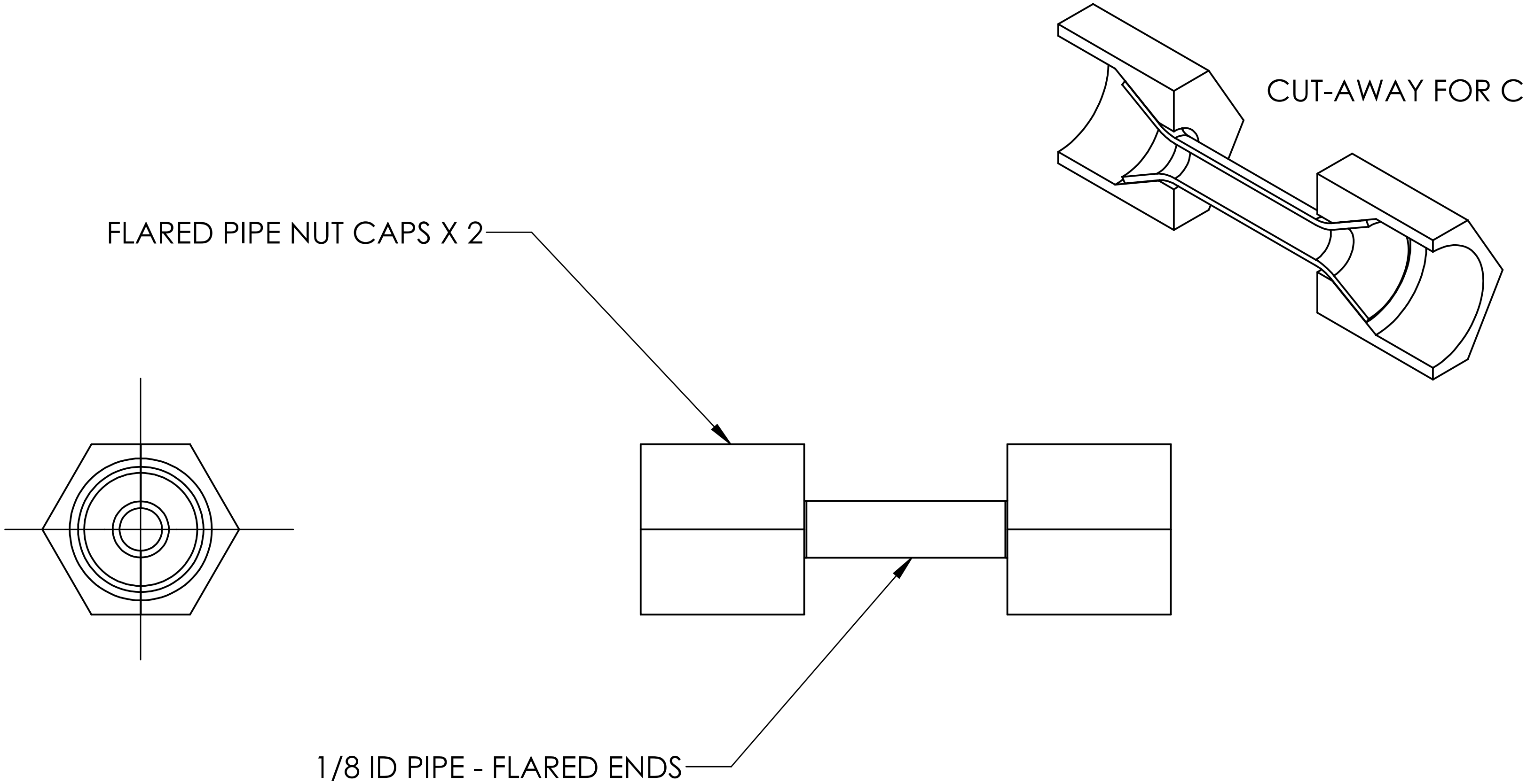
TITLE: 1/8 NPT FEMALE - 1/4 NPT MALE CONNECTOR

DWG NO. FSAEX-XXX-EN-00014

A4

SCALE:2:1

SHEET 1 OF 1



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			REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

1/8 ID PIPE ASM

MATERIAL:
COPPER

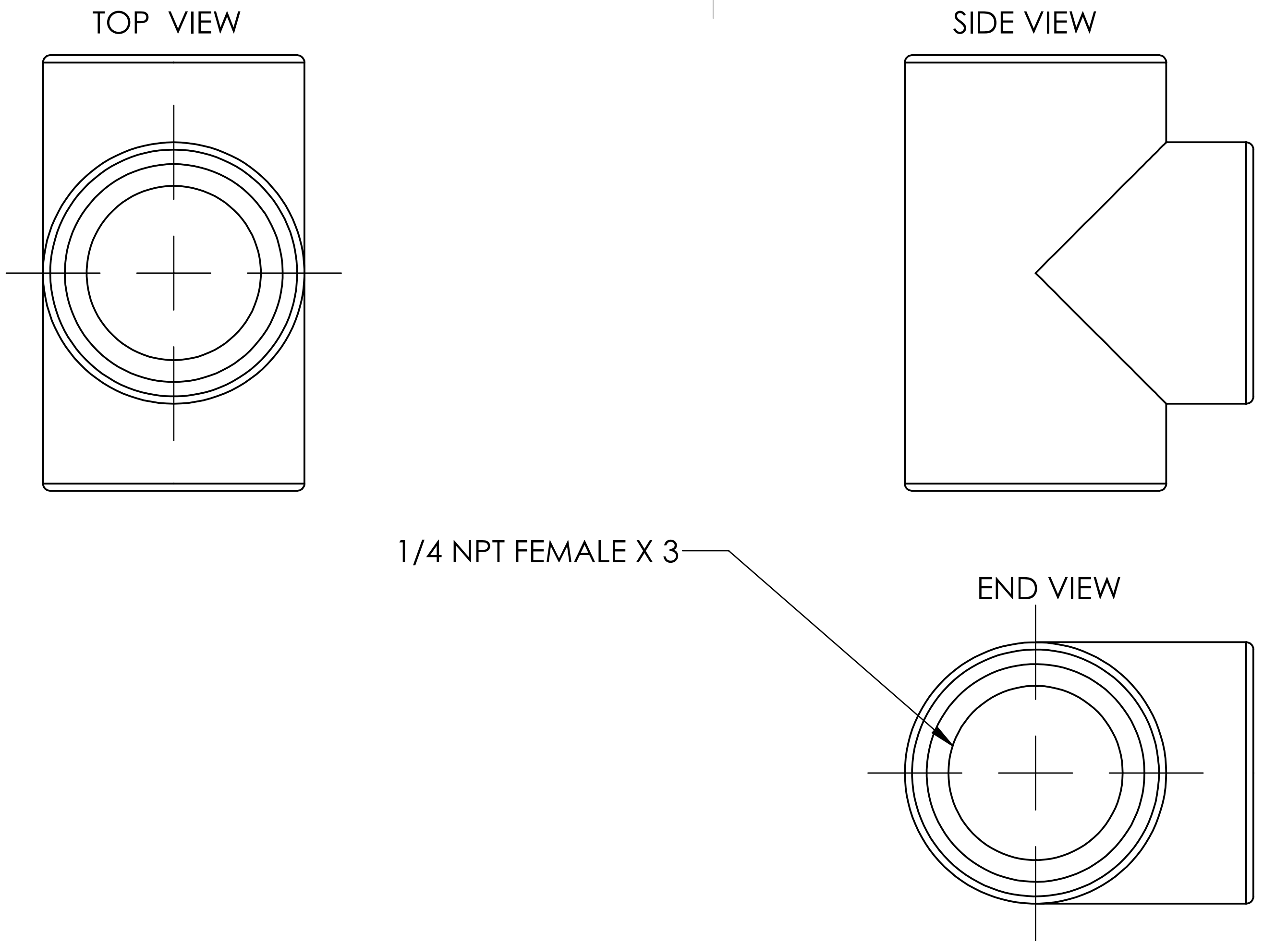
DWG NO. FSAEX-XXX-EN-00015

A4

WEIGHT:

SCALE:2:1

SHEET 1 OF 1



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UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH: MILL		DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
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CHK'D					
APPV'D					
MFG					
Q.A				MATERIAL: ALUMINUM	DWG NO. FSAEX-XXX-EN-00016
					A4
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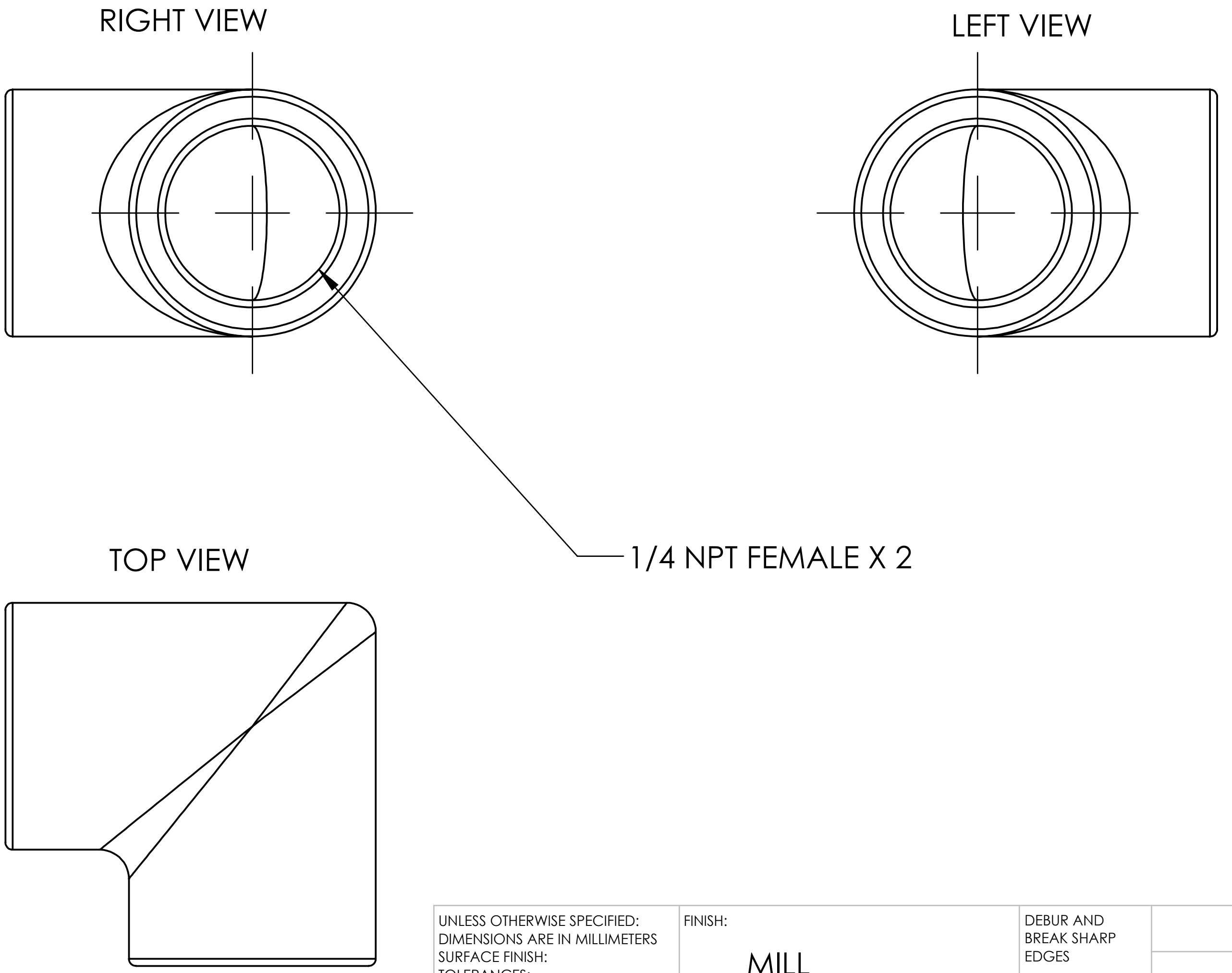
					SHEET 1 OF 1



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DRAWN						
CHK'D						
APPV'D						
MFG						
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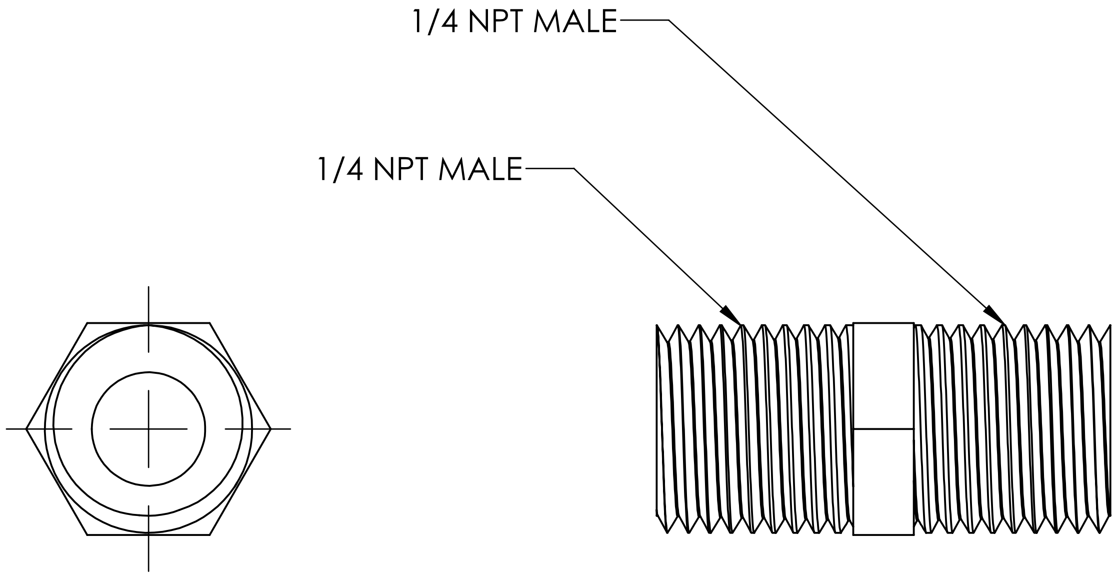


UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH: MILL

DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL: BRASS

WEIGHT:

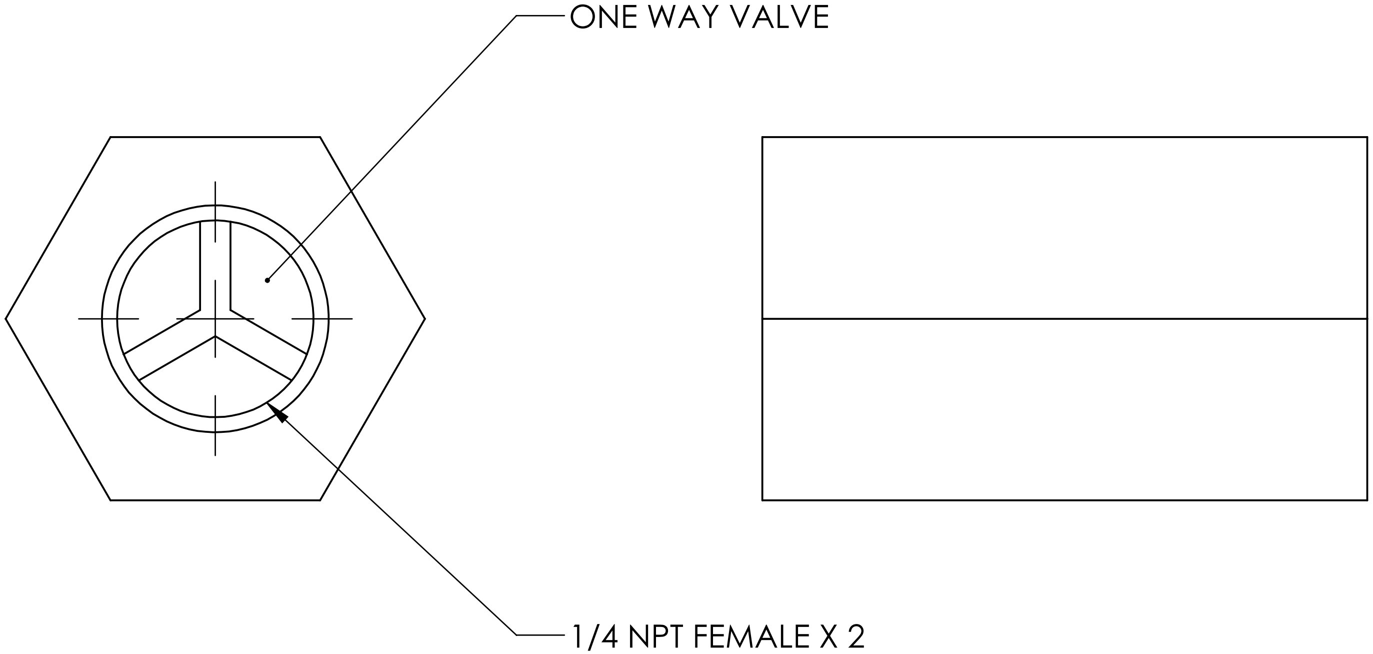
TITLE: 1/4 NPT MALE - MALE CONNECTOR

DWG NO. FSAEX-XXX-EN-00018

A4

SCALE:2:1

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH: MILL	DEBUR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:
ALUMINUM

WEIGHT:

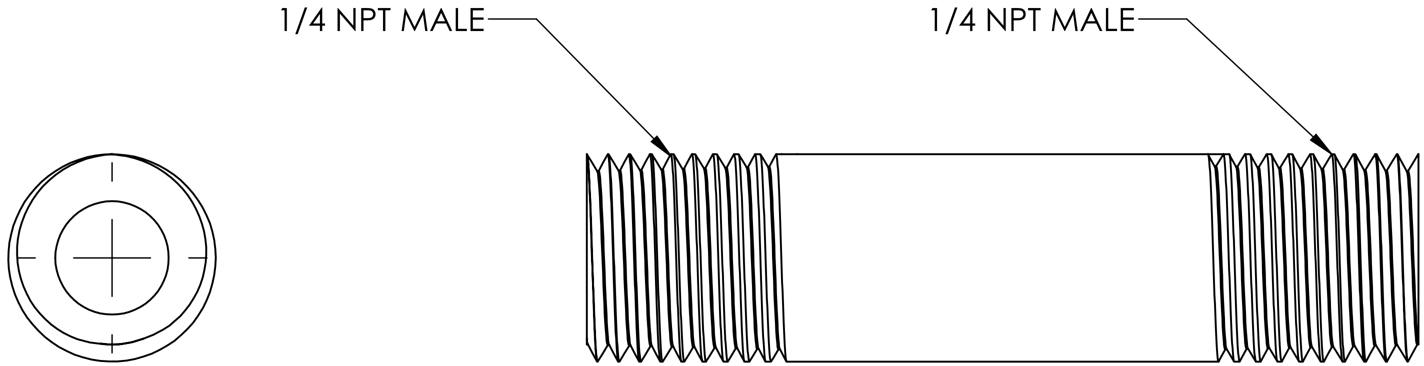
TITLE:
ONE WAY FLOW VALVE

DWG NO. FSAEX-XXX-EN-00019

A4

SCALE:2:1

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH: MILL		DEBUR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION
	NAME	SIGNATURE	DATE	TITLE:	
DRAWN				1/4 NPT MALE - MALE 2'' CONNECTOR	
CHK'D					
APPV'D					
MFG					
Q.A			MATERIAL: BRASS	DWG NO. FSAEX-XXX-EN-00020	A4
			WEIGHT:	SCALE:1:1	SHEET 1 OF 1