The University of Manitoba

Team #16
MECH 4860 – Engineering Design
Final Design Report

Load Simulator for Tractor Performance Testing

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MacDon™

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Dec. 5th, 2011

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Dear Dr. Labossiere,

The following is a copy of Team 16's design report for MECH 4860: Engineering Design. This report has been written for MacDon Industries Ltd, with the help of Dr. Meera Singh as an advisor. The report details the design of a Load Simulator for testing the performance of MacDon's Windrower Tractors. The load simulator is designed to dissipate the power put into it, up to 60 hp.

The main processes built into the load simulator design include: power input, heat generation and heat dissipation. The power input consists of multiple shaft and hydraulic couplers. This power is directed into a hydraulic pump, which transforms it into high pressure hydraulic flow. This is throttled by a valve, which decreases the pressure and increases the temperature of the flow. Thermal energy is dissipated from the flow using a radiator, to prevent overheating. Various concepts and components were considered during the development of this design. A concept that allows for quick and convenient switching between mechanical and hydraulic inputs was developed.

It was determined that through careful selection of components, a load simulator that can dissipate up to 60 hp from either shaft or hydraulic inputs could be designed within reasonable cost and size.

Sincerely,

Team 16 - MacAttack

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Abstract

This report outlines a load simulator design for MacDon Industries Ltd. for tractor performance testing. The load simulator is designed to receive hydraulic and mechanical loads induced by an external source and convert the loads into heat. This results in the external source experiencing a resistance, thereby simulating in-field harvesting action. The main objective of the load simulator design is to simulate a resistance of up to 60 horsepower.

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The final dimensions of the design are 5.5 feet wide by 3 feet deep by 3 feet high. The load simulator is self-contained with a mass below 1000 kg, which allows it to be moved by a forklift. A heating element is implemented in the oil reservoir to allow testing in cold temperatures. The budget for this design project is $15 000. The estimated cost of the proposed design is $15 400.

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Glossary

cc:	Cubic Centimetres
E.T.D:	Entering Temperature Difference
ft:	Feet
gpm:	Gallons per Minute
hp:	Horsepower
in:	inches
kg:	Kilogram
kW:	Kilowatt
L/s:	Litres per Second
lbs:	Pounds
psi:	Pounds per Square Inch
rpm:	Revolutions per Minute
V:	Volts

1. Introduction

1.1. Background

MacDon Industries Ltd. is a family owned company specializing in the manufacturing of farm equipment such as windrowers, headers, discs and drapers. Since most of these products are commonly used in harvesting crops and other agricultural products, the need for efficiency and extended performance of components is critical. In order for MacDon to achieve the highest level of performance for its equipment, product testing is a high priority at MacDon [1]. Load simulation is one part of product testing and involves subjecting components to life-like loading conditions.

The load simulator presently used at MacDon is used to test a wide range of tractor components. This load simulator, shown in Fig. 1, is large and heavy. Therefore, a forklift is required to move it [1]. The simulator is used for testing all component sizes, and large components require more testing time. Therefore, the current time efficiency is low, which increases the cost of a test, because operators must be present to observe the test. Another disadvantage for the current load simulator is that it only has hydraulic inputs, which limits the number of ways to input energy and simulate loading conditions on equipment.

Figure 1. The load simulator presently used at MacDon [1].

1.2. Project Objectives

A load simulator is defined as a machine used to simulate the loading on a piece of equipment or component during operation. The primary objective of this report is to design a load simulator capable of testing and evaluating equipment and components under a capacity of 60 hp. The operators of this load simulator design must be able to reasonably estimate the amount of load that components are being subjected to.

The simulator will also be designed to accept mechanical and hydraulic inputs. If the system is designed to only accept one type of input at a time, the load simulator design must be capable of quickly and conveniently switching in between inputs. To achieve this goal, the load simulator must be able to absorb and process power supplied through a shaft or hydraulic system.

The load simulator has three major constraints that help guide the design process:

- Power
- Size
- Environment

The load simulator must be able to operate using shaft inputs as well as different sizes of hydraulic lines. Therefore, an array of couplers should be readily available to attach the system to equipment lines. Multiple hydraulic lines may be in operation at once. However, only one shaft input will be operating at one time. Any electric components selected for the load simulator should be compatible with the electrical system on the tractor being tested.

In order to move the load simulator conveniently, it must be self-contained. The maximum allowable weight of the system should be less than 1000 kg, which may limit the material chosen for construction. The center of gravity must be low enough so that the load simulator will not topple while being transported. For safe operating practice, the simulator will always be stationary while in use .The load simulator will be used outside, so it must be able to operate in the temperature extremes of Manitoba weather [1].

A Gantt chart had been applied in this project to aid in accomplishing all project objectives in a timely manner. The third phase of this project which represents the most crucial time period is illustrated by the Gantt chart in Appendix A.

1.3. Target Specifications

In order to test all of MacDon's equipment, the load simulator is required to be compatible with a 21 tooth spline and a one inch keyed shaft, as well as specific hydraulic couplers. The shaft connections should be able to transmit 60 hp at speeds up to 4000 rpm.

The load simulator is required to be maneuverable by a forklift. Therefore, it should weigh less than 1000 kg and should have dimensions similar to the forklift, which is 5 ft wide by 6 ft high and 3 ft deep.

The working fluid required by MacDon is 15W-40 oil. This limits the working temperature to a maximum of 190 °F so that the oil will not degrade. It is also required by MacDon that there is an automated emergency shutdown when the oil temperature reaches 210 °F.

MacDon specified that the load simulator should achieve 60 hp and be rated for a maximum pressure of 5000 psi. The values of flow rate, pressure, temperature and shaft speed should be measured and displayed on the exterior of the load simulator. Any electric power into the system should be provided through a 12 Volt input from the equipment being tested [1].

Each time MacDon uses the load simulator, it can potentially be run for up to 6 weeks straight. MacDon requested that the load simulator be rated for around 50 000 hours. MacDon specified a total budget of $15 000 [1]. The target specifications for the design are summarized in Table I.

TABLE I
SUMMARY OF TARGET SPECIFICATIONS

Specification	Target
Pump input shafts (mechanical input)	21 tooth spline, 1 inch keyed
Quick couplers (hydraulic input)	To specified dimensions
Maximum weight	1000 kg
Maximum exterior dimensions	5 ft (wide) x 6 ft (high) x 3 ft (deep)
Working fluid	15W-40 oil
Maximum pressure	5000 psi
Temperature	Below 190 °F, automatic shutdown at 210 °F
Input shaft speed	540 - 4000 rpm
Easily switch between hydraulic and mechanical inputs	Within 1 minute
Electrical system	Compatible with 12 Volts
Lifetime	50 000 hours in operation
Maximum individual run time	6 weeks
Maximum Cost	$15 000 CND

2. Overall Load Simulator Design

Energy is supplied to the load simulator through the shaft or hydraulic lines. The mechanical and hydraulic inputs comprise the power input assembly. This is the portion of the design that will directly connect to the components or piece of equipment being tested.

In order to transmit the power input into the load simulator, all power input into the system is transformed into high pressure hydraulic flow. This high pressure hydraulic flow is then transformed into a high temperature hydraulic flow with the application of a hydraulic flow disruption valve. In order to prevent overheating, a radiator removes thermal energy from the high temperature flow. A hydraulic reservoir of 15W-40 oil and a network of pipes facilitate this hydraulic system. In the event of a temperature overload, an electric clutch has been incorporated to function as an emergency shutdown. The overall load simulator design is illustrated in Fig. 2. All major sections discussed above are labelled in the illustration.

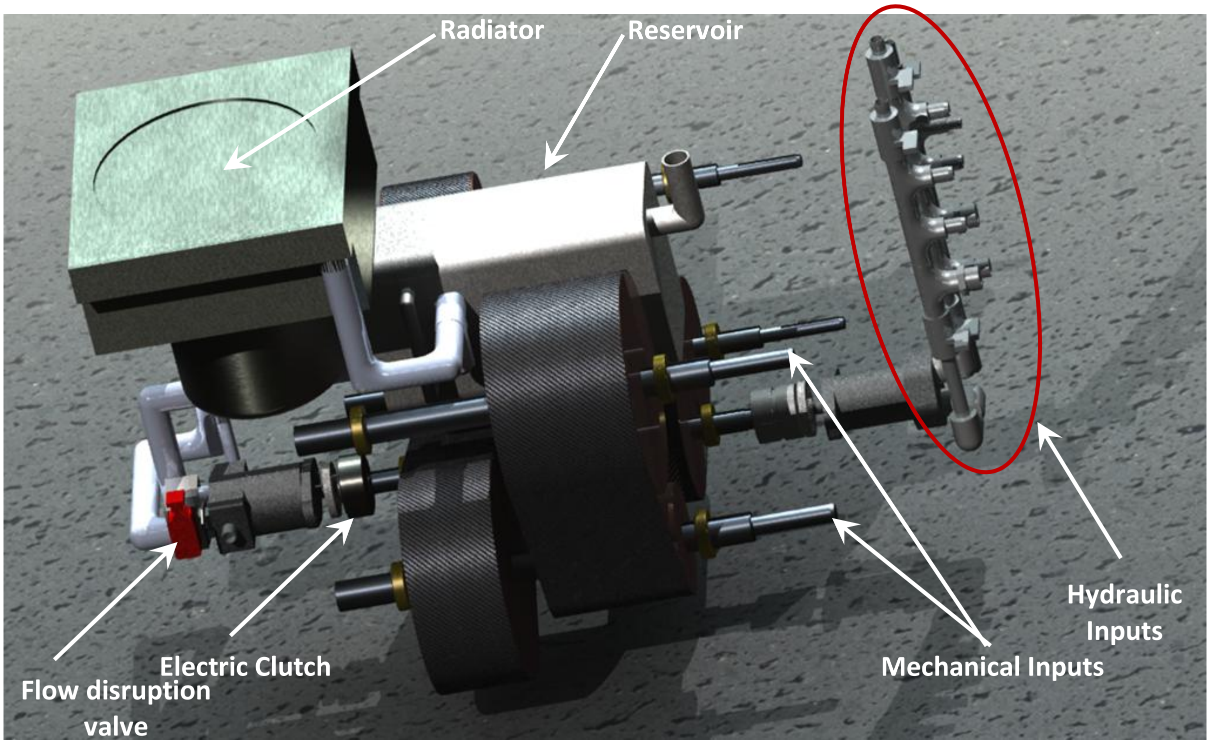


Figure 2. Load simulator design with major sections labelled [2].

3. Power Input Assembly

The power input assembly is the section of the load simulator that accepts all hydraulic and mechanical inputs. It has been designed to meet all the testing requirements outlined by MacDon. Readily available and low maintenance components were chosen to compose the power input assembly. Drawings, estimated costs and mass of components are discussed in detail in the mechanical and hydraulic input assembly sections. The power input assembly is estimated to have a cost of $5780 CDN and an approximate mass of 510kg.

All power inputs are directed into a hydraulic pump, which is the primary mover of fluid in the load simulator hydraulic circuit. The pump is rated up to 126 hp at a maximum speed of 3000 rpm and continuous output pressure of 3600 psi. Fig. 3 below illustrates the general layout of the overall power input assembly.

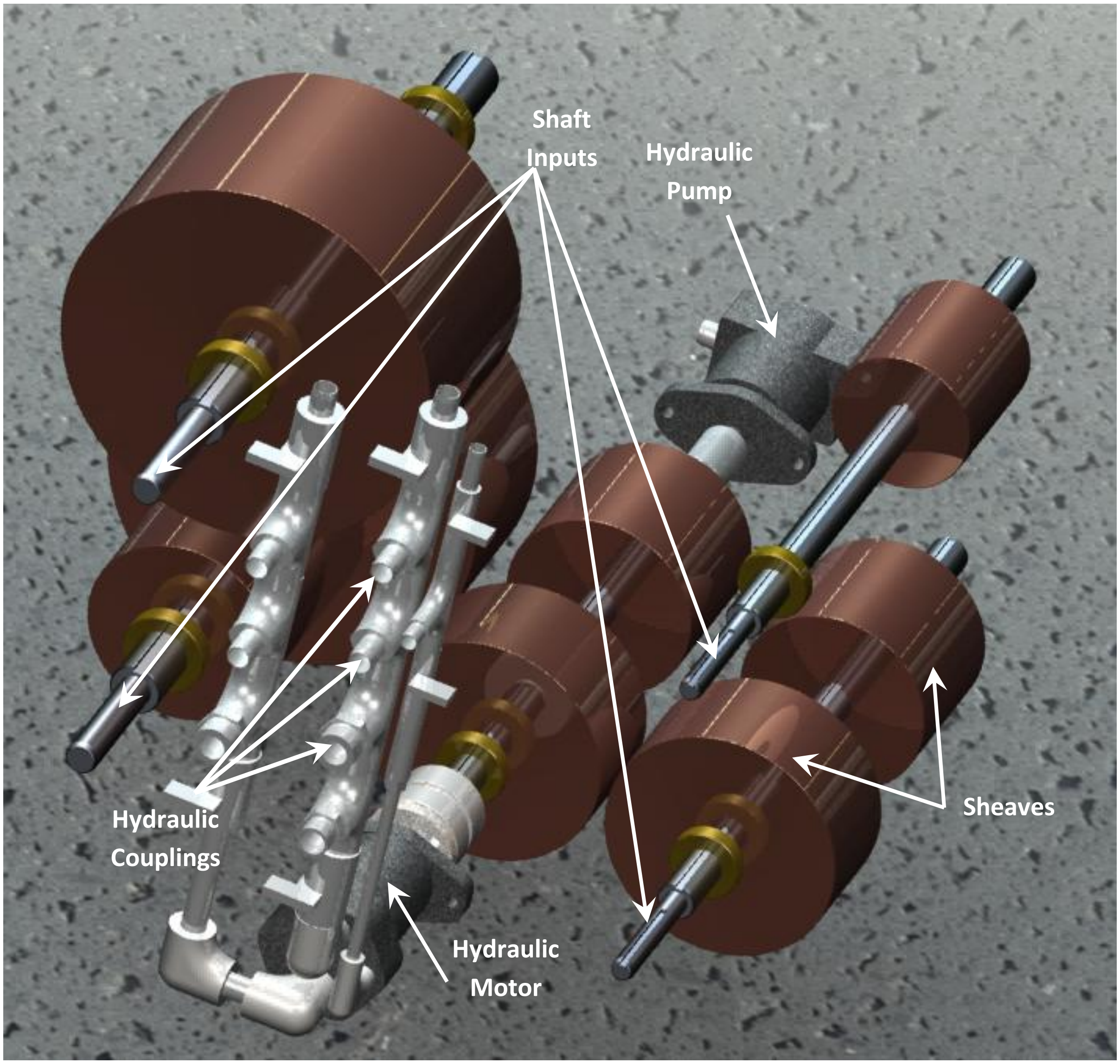


Figure 3. Major components of power input assembly [3].

It can be seen in Fig. 3 that the power input assembly is composed of a hydraulic and mechanical assembly. Note that the belts are not shown on the sheaves for clarity.

3.1. Mechanical Input Assembly

For testing mechanical components, the system is designed to transmit a maximum of 60 hp within the range of 350 to 4000 rpm. Four working shafts host a variety of end-shaft couplings, each appropriate to a specific speed range. These shafts transform torque into an appropriate input for the hydraulic pump, via sheaves and belts. MacDon specified that this design should allow for switching between hydraulic and mechanical inputs within one minute. The mechanical input assembly is estimated to have a cost of $4690 CDN, and a total mass of 450 kg. A more detailed breakdown of cost can be seen in Appendix B, which illustrates a cost breakdown of major components.

3.1.1. Operation of Mechanical Assembly

The four shafts from the input assembly provide a variety of inputs for testing mechanical drivetrains. Each of these shafts is designed to transmit up to 60 hp at distinct speed ranges. Fig. 4 and Table II below illustrate the four end shafts and their individual specifications.

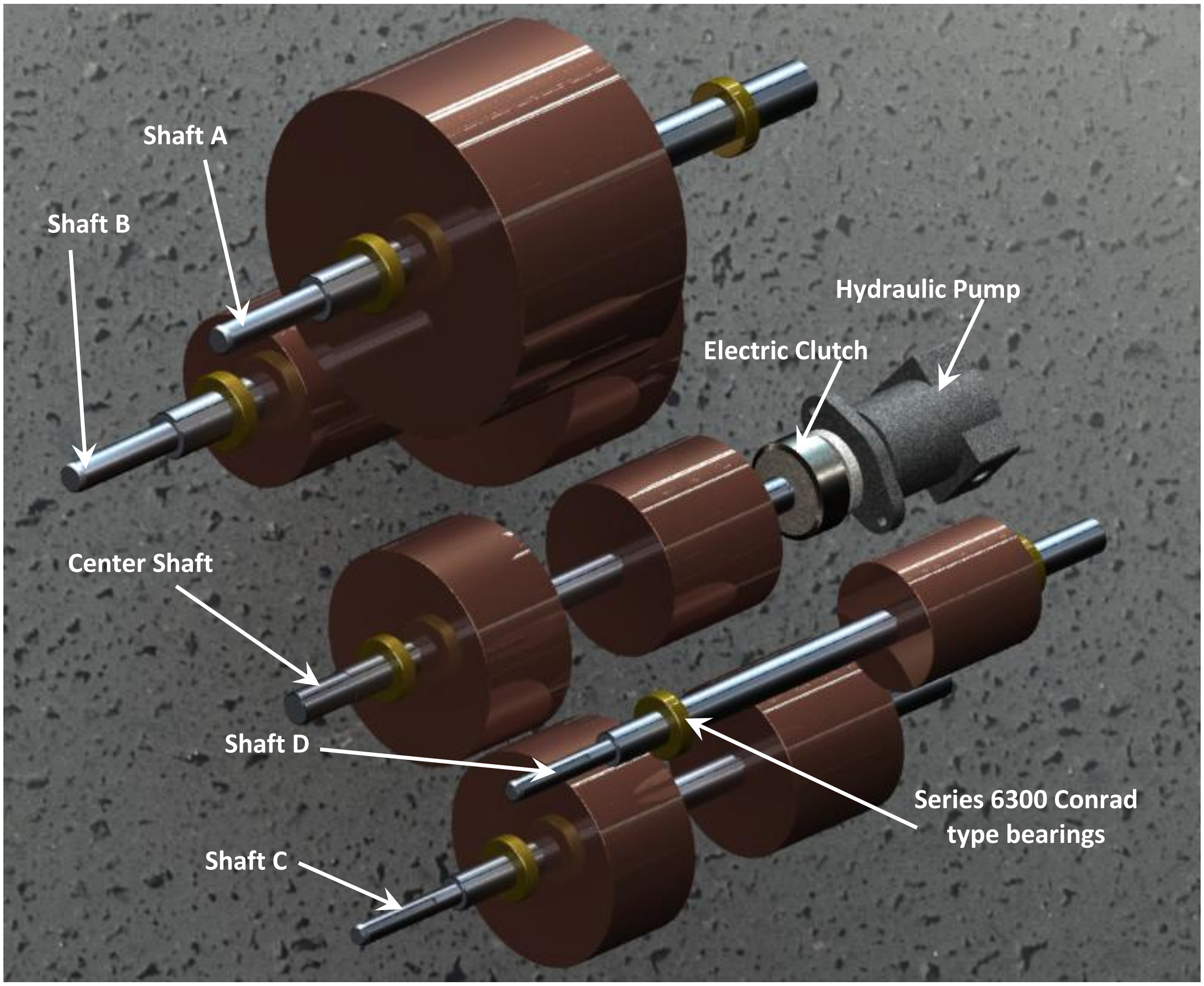


Figure 4. End shaft specifications and major components [4].

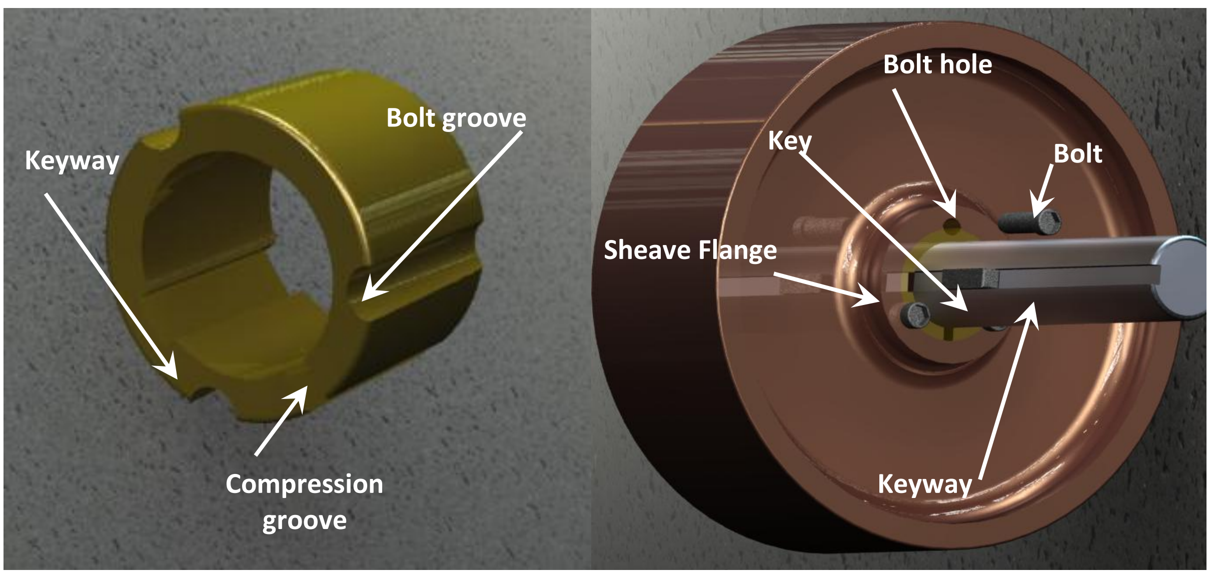
TABLE II
SHAFT SPECIFICATIONS

Shaft Designation	Shaft Speed Range	End Shaft Size and Type
A	350 to 700 rpm	6 spline 1 3/8''
B	700 to 1500 rpm	21 spline 1 3/8''
C	1400 to 3000 rpm	1'' keyed
D	1900 to 4000 rpm	1'' keyed

The shafts are supported by 6300 Conrad type bearings, which are brass coloured in Fig. 4. Shafts A and B have a diameter of 2 3/16'' and use series 6311 bearings. Since shafts C, D, and the center shaft have a smaller diameter of 1 5/8'', series 6309 bearings are used. Given the loads on the shafts, it is estimated that all bearings should last for over 10 000 hours of operation. Therefore, it is recommended that bearings be greased after 100 hours of operation, and visually inspected after 3000 hours. Given this recommended maintenance schedule, warnings of bearing breakdown should be detected before failure.

The hydraulic pump can only transmit 60 hp between 1400 and 3000 rpm. Therefore, a means of transmitting the torque from the input shafts is required. Belt drives were chosen to transmit this torque. L-section Micro-V belts were selected for this task, and they run on the sheaves shown as copper coloured cylinders.

All sheaves are affixed to the shafts via taper lock bushings, which contain a keyway used to grip the key. Fig. 5 illustrates how the taper lock bushing interfaces with the shafts and sheaves.



(a) (b)

Figure 5. a) Taper lock bushing and fastener grooves [5].

b) Interface of shaft with sheave illustrating arrangement of key and bolts [6].

The central component of the mechanical input assembly is the hydraulic gear pump. A 126 hp pump was selected. However, the load simulator must be capable of dissipating only 60 hp, which enables the load simulator to transmit 60 hp over a large speed range. This selection also provides the added feature of overload protection for the pump. In the event of an overload, the belts are designed to fail before the pump can reach its maximum power of 126 hp.

The hydraulic pump is connected to the central shaft via a female couple and keyway. Outputs of the pump are located at the rear and will connect to the variable pressure relief valve. Fig.6 highlights these design features.

MacDon specified that it is not necessary to gauge power dissipation accurately. However, it is required that a power reading is given. For power gauging, a flow meter and pressure gauge are needed. Therefore, it is recommended that an analog pressure gauge be mounted on the hydraulic pump. From flow and pressure readings, the amount of power dissipation can be read from both metric and imperial figures included in Appendix C, which illustrates system operating curves that can be used for power estimation.

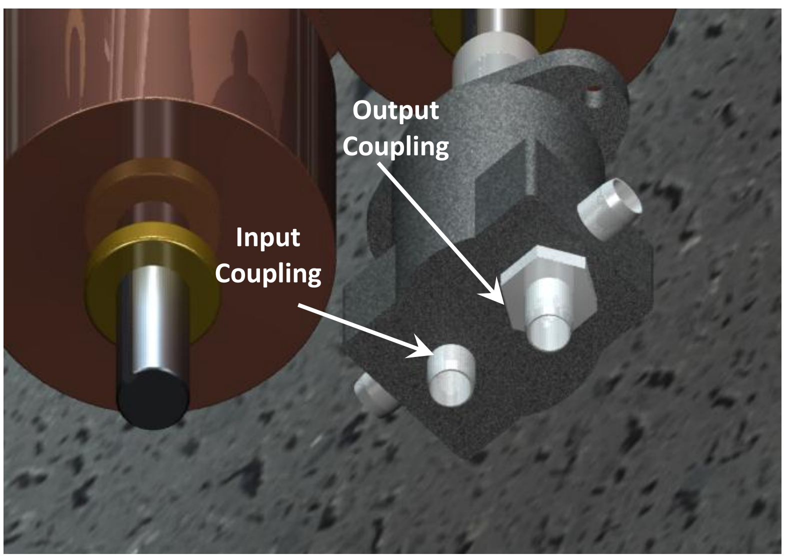


Figure 6. Hydraulic couplings from pump to restrictor valve [7].

Detailed specifications of the hydraulic gear pump are outlined in the section 3.1.2. An electric clutch is interfaced with the hydraulic pump and center shaft. In order to protect the components of the load simulator from becoming damaged due to excessive heat, an electric clutch was chosen as a safety shutdown mechanism. While the radiator illustrated in section 6 should be able to prevent overheating of the hydraulic fluid, the electric clutch is incorporated into the design to serve as a secondary means of protecting the load simulator.

3.1.2. Mechanical Assembly Drawings, Cost and Mass Estimates

The major components of the mechanical input assembly are illustrated in Fig. 7 to Fig. 15. This does not include hydraulic or shaft couplings, bearings, or bolts since they are interchangeable and considered non-important items in comparison to the major components.

The Sauer Danfoss hydraulic pump illustrated in Fig. 7 was selected for this design since MacDon has experience working with the Sauer Danfoss brand.

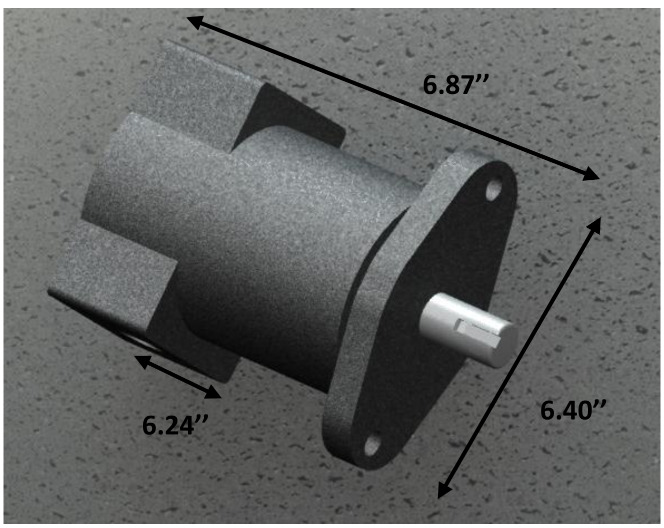


Figure 7. Hydraulic Gear Pump [8].

Specifications [9]:

- Model: Sauer Danfoss CP 180-045
- Number of units required: 1
- Peak Pressure: 4100 psi
- Continuous Pressure: 3600 psi
- Input shaft: 1'' keyed
- Displacement 71.88 cc/rev
- Power rating: 126 hp
- Mass: 11kg
- Estimated cost: $450

The custom designed shafts illustrated in Fig. 8, 9 and 10 were designed to support the belt drive system which is at the core of the mechanical input assembly. These shafts are made of ANSI 1040 cold drawn steel and possess keyways as fixture points for attaching sheaves and couplings. A detailed shaft design procedure is shown in Appendix D.

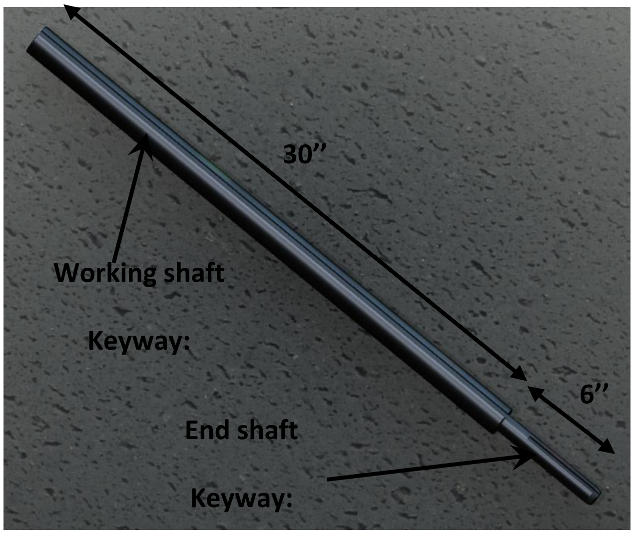


Figure 8. Illustration of shafts A and B [10].

Specifications:
- Custom Designed Shaft
- Number of units required: 2
- Input torque rating: 3000 lbin
- Working shaft diameter: 1 5/8’’
- End shaft diameter: 1’’
- Mass: 40 kg
- Estimated cost: $200

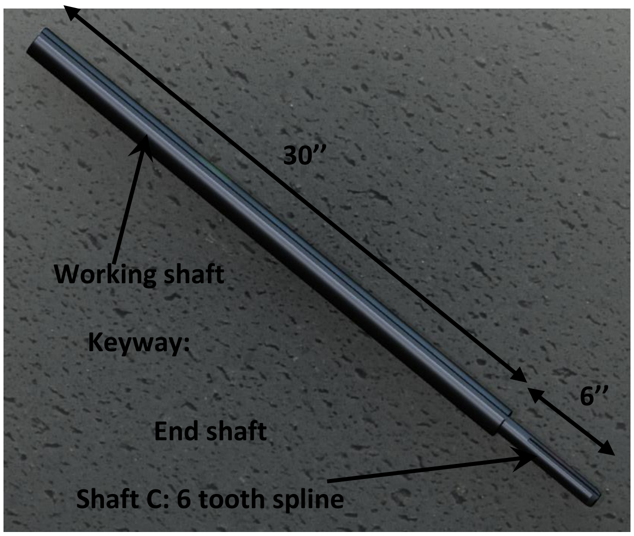
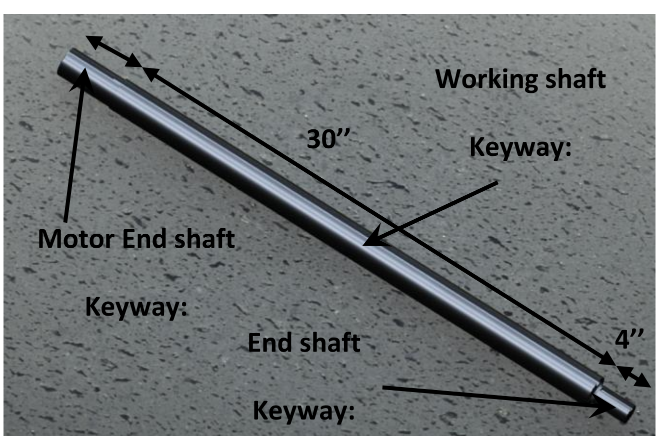


Figure 9. Illustration of shafts C and D [11].

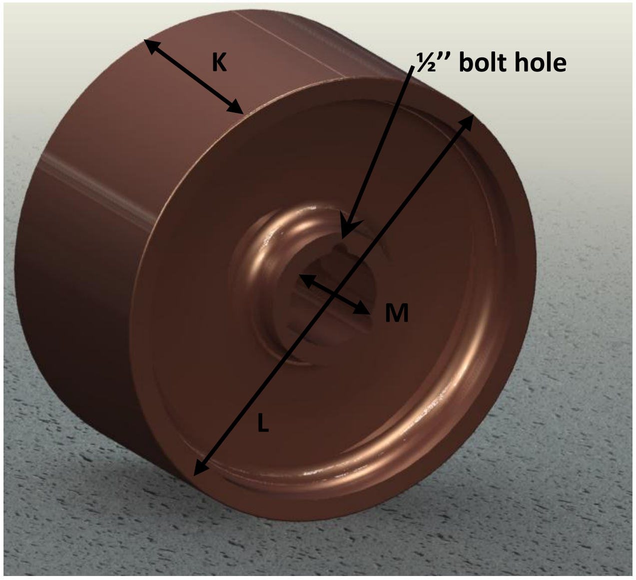
Specifications:
- Custom Designed Shaft
- Number of units required: 2
- Input torque rating: 10 000 lbin
- Working shaft diameter: 2 3/16’’
- End shaft diameter: 1 3/8’’
- Mass: 55 kg
- Estimated cost: $350



Specifications:
- Custom Designed Shaft
- Number of units required: 1
- Motor end shaft diameter 1 1/2 ’’
- Working shaft diameter: 2 3/16’’
- End shaft diameter: 1’’
- Mass: 45 kg
- Estimated cost: $250

Figure 10. Illustration of shafts C and D [12].

Belt and sheave products were chosen from Gates Corporation to meet the design requirements since Gates products are readily available and provide convenient design standards. The stock sheaves selected from Gates are detailed in Fig. 11 and Table III.



Specifications:

- Gates Corporation Stock Sheaves
- Number of units required: 8
- Estimated mass: 45 kg per sheave
- Estimated cost: $160 per sheave

Figure 11. Illustration of belt drive sheaves [6]

TABLE III
SHEAVE DISIGNATION AND SPECIFICATIONS [13]

Sheave Designation	Diameter, K	Width, L	Inside Diameter, M	Quantity
(A)	6''	6.7''	2 1/8''	X 1
(B)	8''	6.7''	2 1/8''	X 2
(C)	8''	10.1''	3 3/8''	X 1
(D)	10''	5.1''	2 1/8''	X 2
(E)	16''	6.7''	3 3/8''	X 1
(F)	16''	10.1''	2 1/8''	X 1

Fig. 12 below illustrates the locations of the sheaves in the mechanical input assembly. This figure shows that the sheaves are mounted on the shafts with space to spare. The bearings are spaced 21'' apart to accommodate the width of the belt drive systems. For the design of the shafts, it is assumed in the technical analysis that the sheaves are placed 1'' from the bearings. Therefore, the bearings could potentially be moved to optimize the design, which would require revisiting the shaft designs. A detailed technical analysis of the shafts is shown in Appendix D.

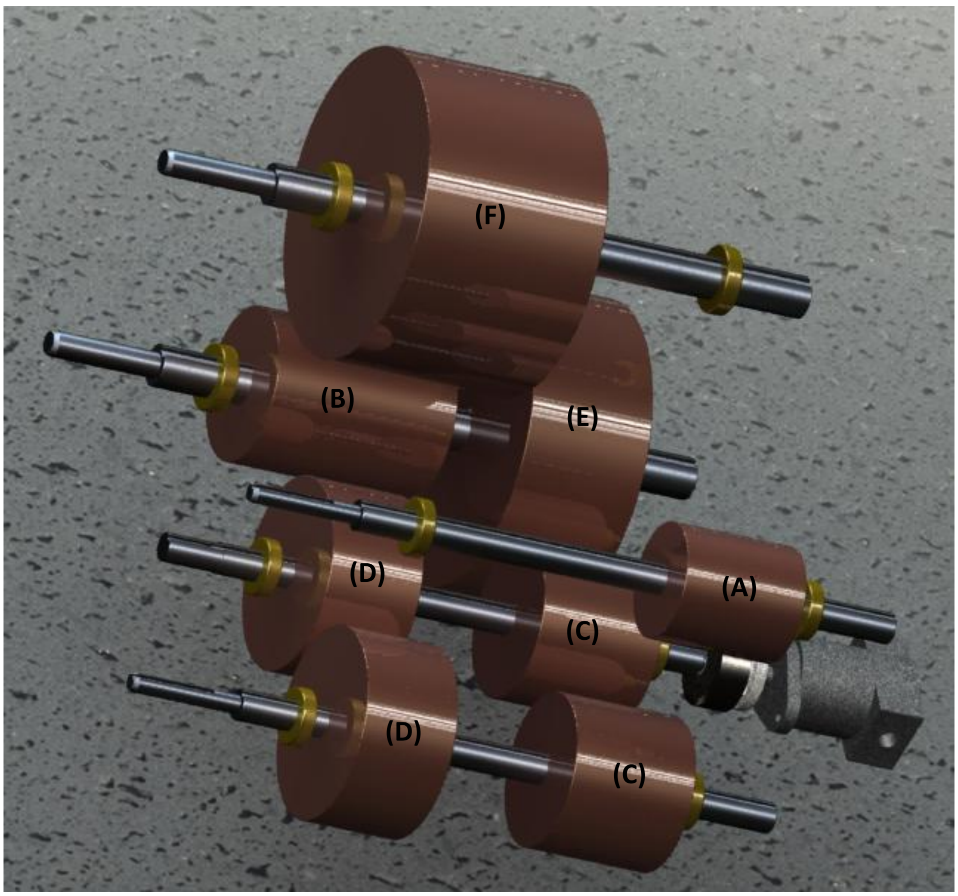


Figure 12. Side view of mechanical input assembly illustrating sheave designation in correspondence to TABLE III [4].

Fig. 13 and Table IV below detail the number, size and quantity of belts that are used in the mechanical input assembly. Micro-V belts were selected, since they are suitable for the speed and range requirements of the load simulator. Table IV specifically illustrates the length of belt for a given sheave-to-sheave span.

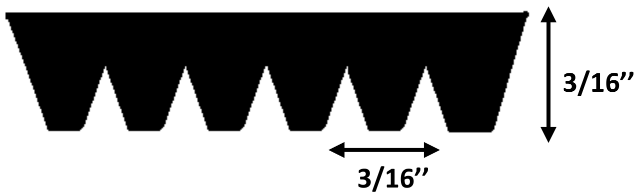


Figure 13. Illustration of Micro-V belts used in mechanical input assembly [14].

Specifications:

- Gates Corporation L-Section Micro-V Belts
- Number of units required: 17
- Estimated mass: 3 to 4 kg per belt
- Estimated cost: $125 per belt

TABLE IV
SHEAVE SPAN SPECIFICATIONS [13].

Sheave Span	Belt Length	Center Distance	Quantity
(C) to (A)	50.0’’	14.0’’	X 4
(D) to (D)	50.0’’	10.4’’	X 3
(E) to (C)	65.5’’	13.4’’	X 4
(F) to (B)	65.5’’	13.4’’	X 6

In comparison to the Micro-V belts used in the design, Kevlar cog belts would lower the cost of the overall design, since they are narrower. These Kevlar belts are used in Harley-Davidson motorcycles and are designed to handle comparable speeds and power as the load simulator [15]. Since Kevlar cog belts have a maximum width of 2’’, as compared to the Micro-V belts that in some cases are 10’’, the length of the overall assembly could be decreased. This shortening of the overall assembly implies that shorter shafts could be used, which would substantially lower the cost of the design. Given the switch to Kevlar cog belts, sheaves comparable in size to motorcycle sheaves could also be used. Since the motorcycle sheaves are much smaller than most of the sheaves employed in the load simulator design, the overall cost could be further reduced. Due to time project time constraints, a Kevlar cog belt design was not developed.

Fig. 14 below illustrates the center distance between shafts of the mechanical input assembly. These center distances represent the maximum shaft-to-shaft distances for the mechanical input assembly. The overall shape of the mechanical input assembly is largely due to the maximum centre distance as a result of the belt drive design.

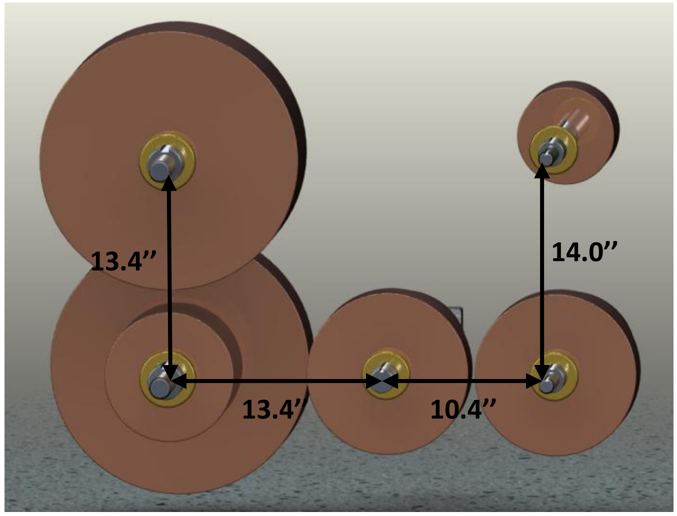


Figure 14. Shaft spacing of mechanical input assembly (rear view) [4].

The clutch serves as a safety shutdown mechanism. This mechanism will be trigger if the temperature sensor reads a value over 210°F, which is the absolute maximum operating temperature of the working fluid. The electric clutch could be actuated by a mechanism similar to that of an on/off control for an electric radiator motor. A keyway serves to attach the clutch to the center shaft and the clutch is bolted to a flange, which meshes with a keyway on the pump shaft.

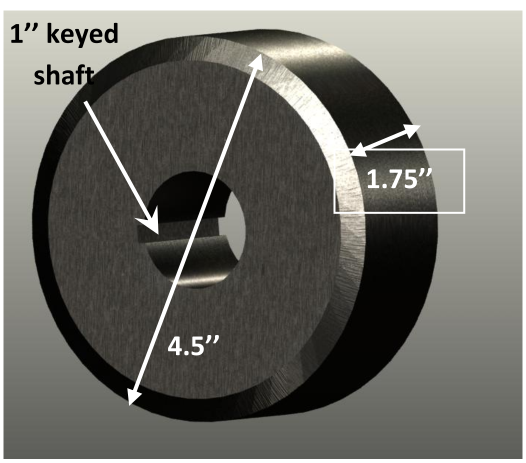


Figure 15. Illustration of electric clutch used in emergency shutdown [17].

Specifications [16]:

- Model: Ortlinghaus 0812-0015
- Number of units required: 1
- Actuation voltage: 12 V
- Estimated mass: 2.6 kg
- Estimated cost: $250

3.2. Hydraulic Input Assembly

For testing hydraulic components, the hydraulic input assembly is designed to transmit a maximum of 65 hp at a maximum pressure of 3500 psi. Hydraulic power is directed into a motor via a bank of hydraulic couplers. These quick coupler attachments present multiple options for connecting hydraulic lines of various calibres. The hydraulic motor then transforms the hydraulic power into mechanical power, which is transmitted into the hydraulic pump. A detailed cost and technical analysis of the hydraulic input assembly components is shown in Appendix B.

Banks of hydraulic couplers provide a variety of input options into the hydraulic assembly. This hydraulic power is directed towards a motor, which converts the hydraulic power into mechanical power. The mechanical power is then transmitted into the hydraulic pump via the center shaft. The input hydraulic assembly is shown in Fig. 16.

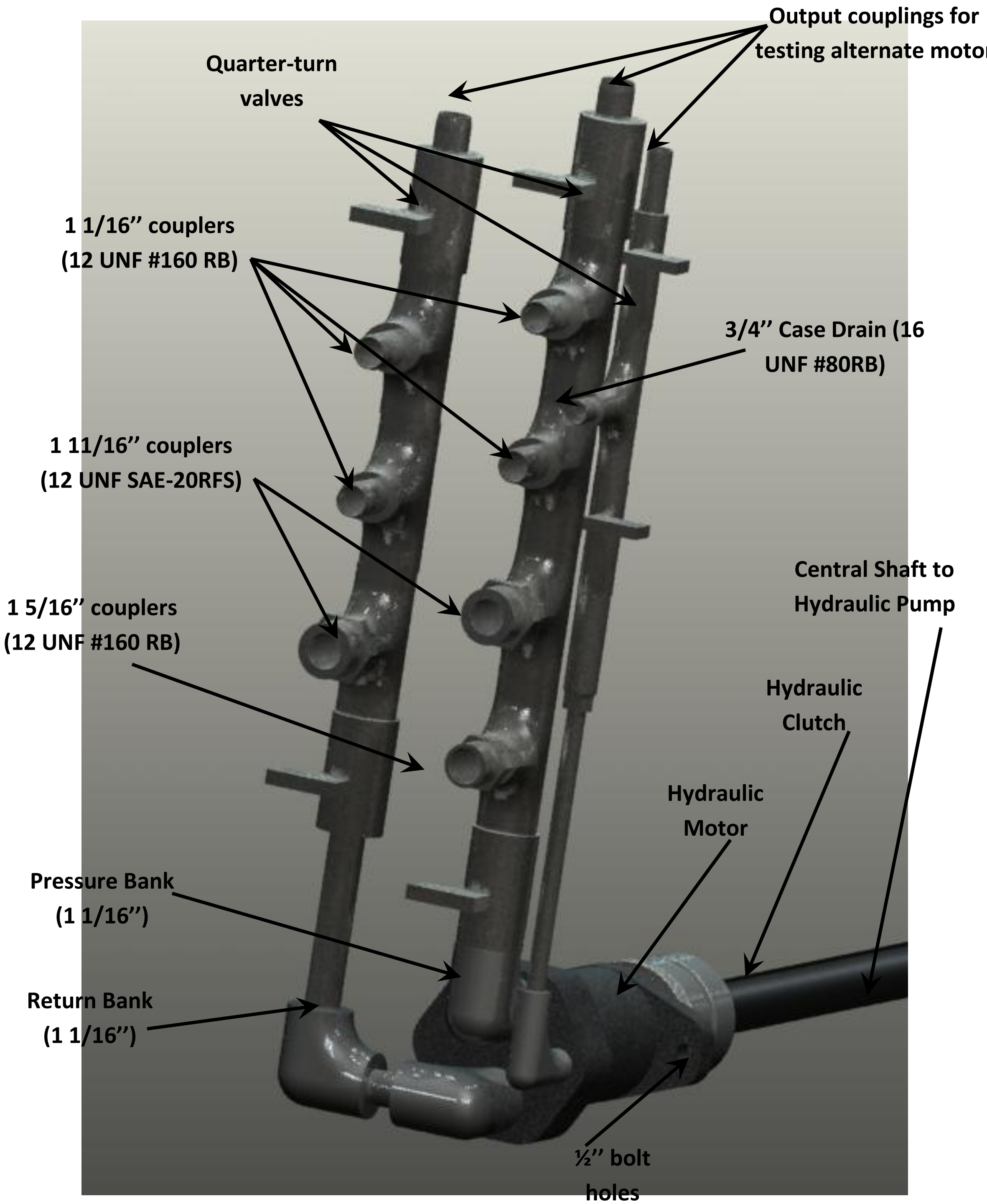


Figure 16. Hydraulic input assembly and major components [18].

Fig.16 illustrates where hydraulic connections can be made to test an external motor. Half inch sized quick couplers could easily be threaded onto the couple at the top of the assembly. These quick couplers would then connect to hydraulic flex lines that would run to an external motor. The working shaft of this external motor would then be connected to one of the four shaft inputs. Note that this external motor is not illustrated in this design report. The remaining couplers on the pressure and return banks are sized to fit a variety of quick coupler connections specified by MacDon.

The quarter-turn valves illustrated in Fig. 16 serve to cut off the flow to either the built-in hydraulic motor or the external motor attachment, as only one set of valves should be closed at a time.

The hydraulic motor used in this assembly is capable of transforming up to 65 hp into mechanical shaft work. The maximum power rating is achieved at a rotational speed of 1500 rpm and a continuous pressure of 3500 psi. More detailed specifications of the motor are shown in section 3.2.1.

Since the hydraulic motor interfaces with the mechanical assembly, a hydraulic clutch is used to disconnect it from the central shaft of the mechanical assembly. This disconnection is necessary because hydraulic motors are generally not designed to have shaft work input into them. Turning the shaft of the motor would prematurely wear out the bearings inside the motor itself. Therefore, whenever the hydraulic assembly is in use, a low-pressure hydraulic flow must be actuated to the clutch to engage it with the central shaft. The clutch is released for mechanical operations by releasing the hydraulic pressure from the clutch. This can be done quickly and easily; therefore, switching from mechanical to hydraulic input could be performed within a minute. The connection of the clutch to the motor is illustrated in Fig. 17 below.

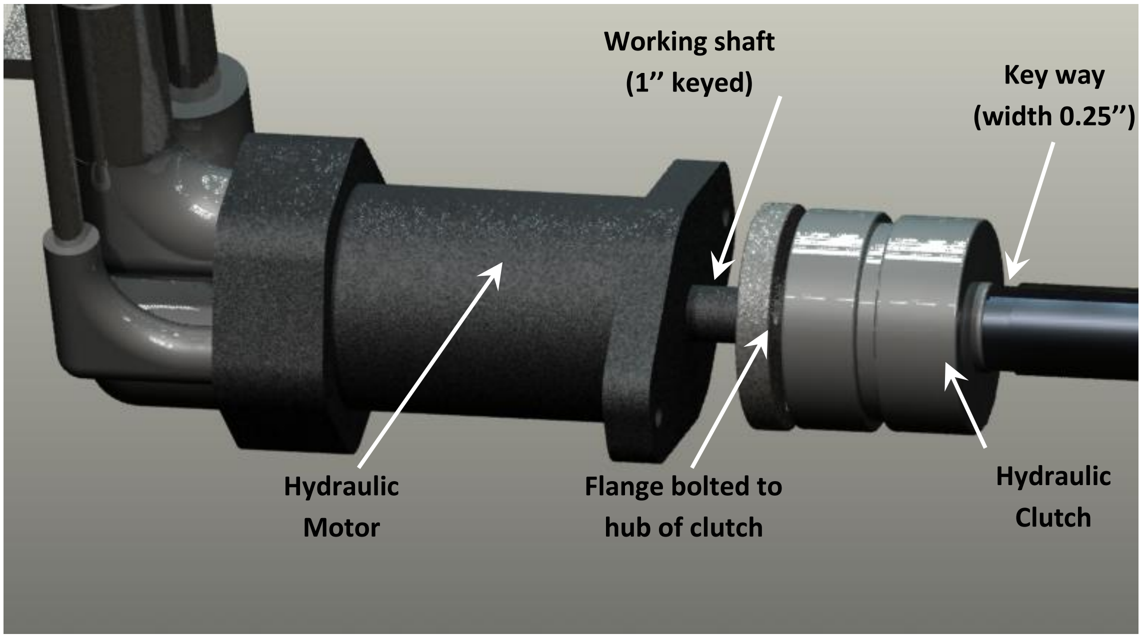


Figure 17. Connection of hydraulic clutch to motor via a bolt flange [18].

It should be noted that the hydraulic clutch is rated for a higher maximum torque and rotational speed than the hydraulic motor, which provides a larger design factor. Given this factor, the hydraulic clutch can be expected to last for thousands of hours of operations.

3.2.1. Hydraulic Assembly Drawings, Cost and Mass Estimates

The major components outlined in Fig. 18 and 19 include only the hydraulic motor and clutch. Other components such as couplers, tees and elbows are interchangeable components and considered non-important in comparison to the specifications of the motor and clutch. The overall cost of the hydraulic input assembly is estimated to be $1040 CDN. This overall cost includes all parts, components and sealants needed for a hydraulic system. Generic components such as tees, quarter-turn valves and elbows are estimated to cost $390 CDN. Cost estimates are outlined in Appendix B, which details the specific cost of each component. The components in Fig. 18 and 19 are estimated to cost $650. These parts represent the major components in the hydraulic input assembly. The total mass of the hydraulic input assembly is estimated to be no more than 60 kg.

The Eaton hydraulic motor illustrated in Fig. 18 was selected for this design since MacDon has experience working with the Eaton brand [1].

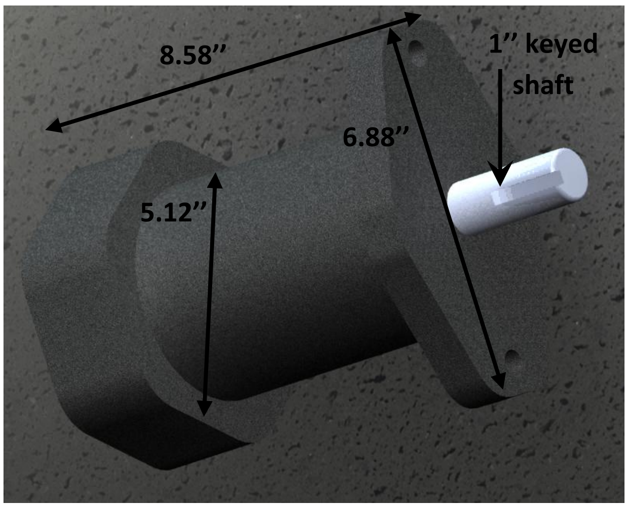


Figure 18. Illustration of hydraulic motor [19].

Specifications [20]:

- Model: Eaton Hydraulic 74624/74644
- Number of units required: 1
- Maximum pressure rating: 3900 psi
- Continuous pressure rating: 3500 psi
- Displacement: 82.6 cc/rev
- Power rating: 65 hp
- Shaft output: 1'' keyed
- Estimated mass: 11 kg
- Estimated cost: $350

The P series clutch from Logan Clutch Corporation was selected since the unit is sealed and pre-lubricated. Since the clutch does not require lubrication operation will be more convenient and maintenance costs will be reduced.

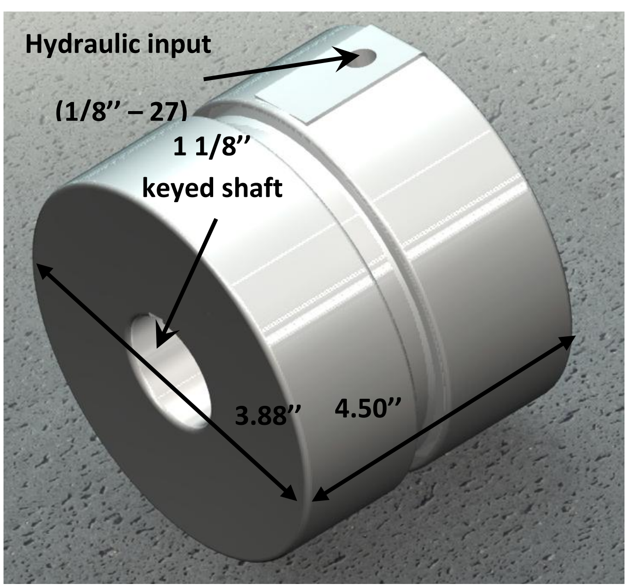


Figure 19. Illustration of electric clutch used in emergency shutdown [21].

Specifications [22]:

- Model: Logan Clutch Corporation 350 P Series
- Number of units required: 1
- Maximum torque rating: 3800 lbin
- Maximum rated speed: 3600 rpm
- Actuating flow: 1.8 gpm
- Estimated mass: 2.7 kg
- Estimated cost: $300

4. Hydraulic Flow Disruption Valve

The hydraulic valve heats oil by disrupting pressure fluid flow. . The type of hydraulic valve used is a variable pressure relief valve. This relief valve allows for accurate pressure drops a between in inlet and outlet.

4.1. Operation

The valve is comprised of a spring and a diaphragm. The spring can be adjusted such that the desired outlet pressure is reached. The spring and diaphragm are connected by a mechanism that opens the diaphragm if the spring relaxes, and closes the diaphragm if the spring compresses. Essentially, as the input pressure increases the cross-sectional area in the valve decreases, causing a restriction to the flow. This mechanism allows for fluctuation of the inlet pressure while keeping the outlet pressure constant.

4.2. Selection Criteria

The criteria for the valve selection include fluid flow rate and operating pressure. The valve must be rated for a maximum flow rate of 57gpm, and a maximum pressure of 5000 psi and must be of a manageable size and mass.

4.3. Selected Valve

The valve used in the load simulator is a R5V model from Parker Hannifin Corporation, illustrated in Fig. 20. It weighs 4.6 kg and its exterior dimensions are 141mm by 141 mm by 60 mm. The selected valve is the 2 port R5V model with the SAE 1” hole and SAE 61 flange. The valve is rated for a maximum flow rate of 79 gpm and a maximum operating pressure of 5075 psi.

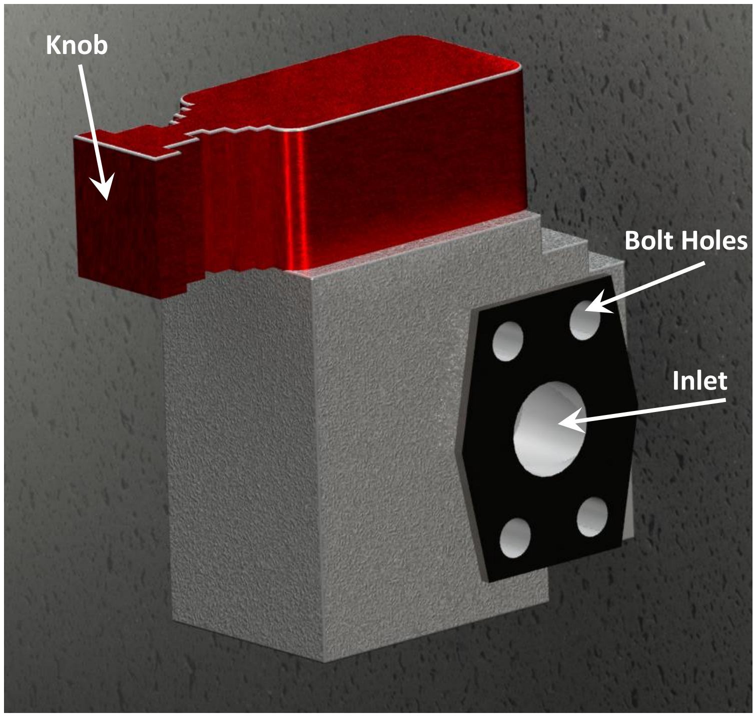


Figure 20. R5V variable pressure relief valve [23].

The approximate cost of the R5V valve is estimated to be $800 CDN.

5. Radiator

The use of a radiator is required for the proposed system because of the rapid heat generated in the system. Using basic thermodynamic principles, displayed in Appendix E, it is shown that without heat dissipation the oil would rapidly overheat. The heat dissipation section of the load simulator will comprise a liquid-to-air radiator run by an electric fan. A liquid-to-liquid would be more efficient, however a liquid-to-air radiator is used for simplicity and cost. The manufacturer selected is Thermal Transfer Products because of the variety of radiators available and the detailed performance curves. The selection criteria and selection process of the radiator will be discussed in the following sections.

5.1. Selection Criteria

The requirements for the radiator include a flow rate of 57 gpm, an entering temperature of 210 °F, and a maximum design power of 90 hp. The maximum input power is 60, however a safety factor of 1.5 is applied to the radiator to ensure sufficient cooling. The flow rate and power input are provided by the operating conditions of the power input assembly. The absolute maximum temperature of the working fluid determines the entering temperature.

5.2. Selection Process

The radiator selection process follows a selection procedure set by the manufacturer. A rule of thumb for radiator selection dictates that the cooling system heat load should be sized for approximately 33% of the input power. In this case the radiator heat load should be 30 hp, as the maximum design input power is about 90 hp. This load is converted to 84 833 BTU/hr, which is in the units from the manufacturer performance curves, shown in Appendix F. The entering temperature difference (E.T.D.) is applied to the load to get the corrected heat rejection value. The E.T.D. is the difference between the maximum entering oil temperature and the maximum ambient air temperature, which are 210 °F and 85 °F respectively. The basic radiator effects are displayed in Fig. 21.

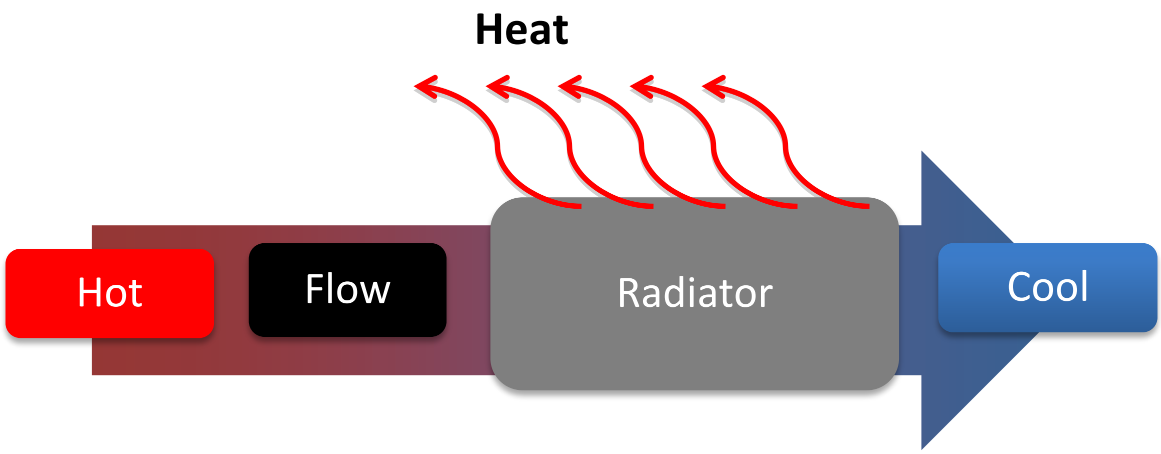


Figure 21. Basic effects of radiator.

The radiator selection process results in a corrected heat rejection value of 67 867 BTU/hr; based on this value and the given flow rate of 57 gpm the ideal radiator is the BOL-400, shown in Fig. 22.

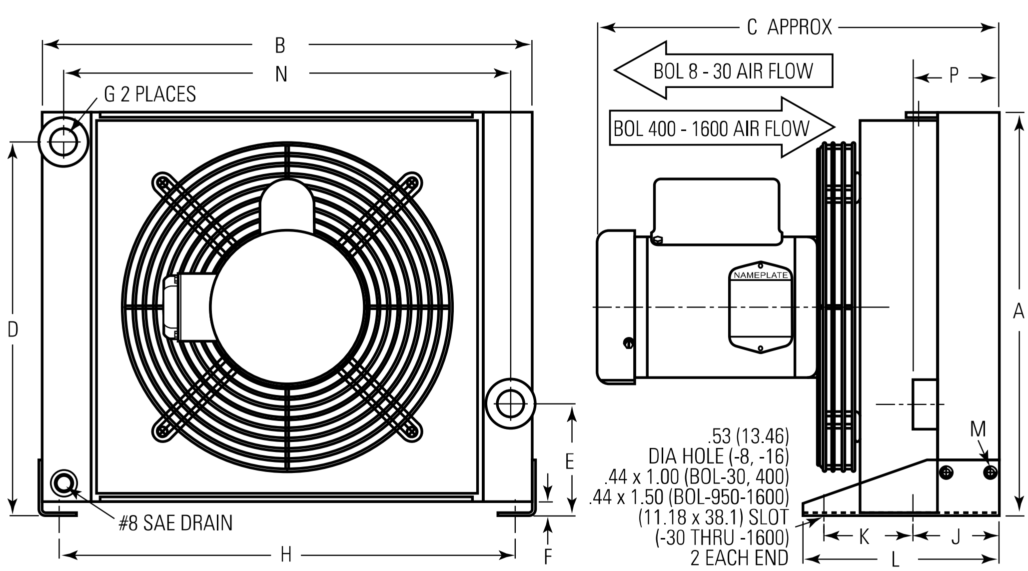


Figure 22. Radiator schematic [24] [used with permission].

The main advantage of the radiator selected is the different configurations and options available for the product. The radiator can be customized by size, connection type, motor type, core, and noise level. The most notable feature is the electrically powered fan motor, which will be used for simplicity and because electrical input is readily available. The retail cost of this product is $2150 CDN as provided by Drive Products [25]. The overall specifications of the chosen radiator are summarized in Table V.

TABLE V
RADIATOR SPECIFICATIONS

Parameter	Value
Weight	67.13 kg
Maximum temperature	300 °F
Maximum pressure	250 psi
Temperature difference	125 °F
Flow rate	57 gpm
Input Power	90 hp
Cost	$2150 CDN

6. Reservoir

The criteria for the oil reservoir design include material selection, shape, orientation, and oil level measurement. Stainless steel and mild steel are both final considerations for the reservoir material. Initially stainless was selected because of its excellent corrosion resistance. Mild steel is the final selection primarily because of the low manufacturing cost and the manufacturability. The size of the reservoir is limited by space available within the power input assembly. Based on the available space the reservoir can be approximately 8” x 10” x 30”, which results in a total volume of 39 litres. The reservoir will be located in the center of the input assembly, which can be seen in Fig. 23.

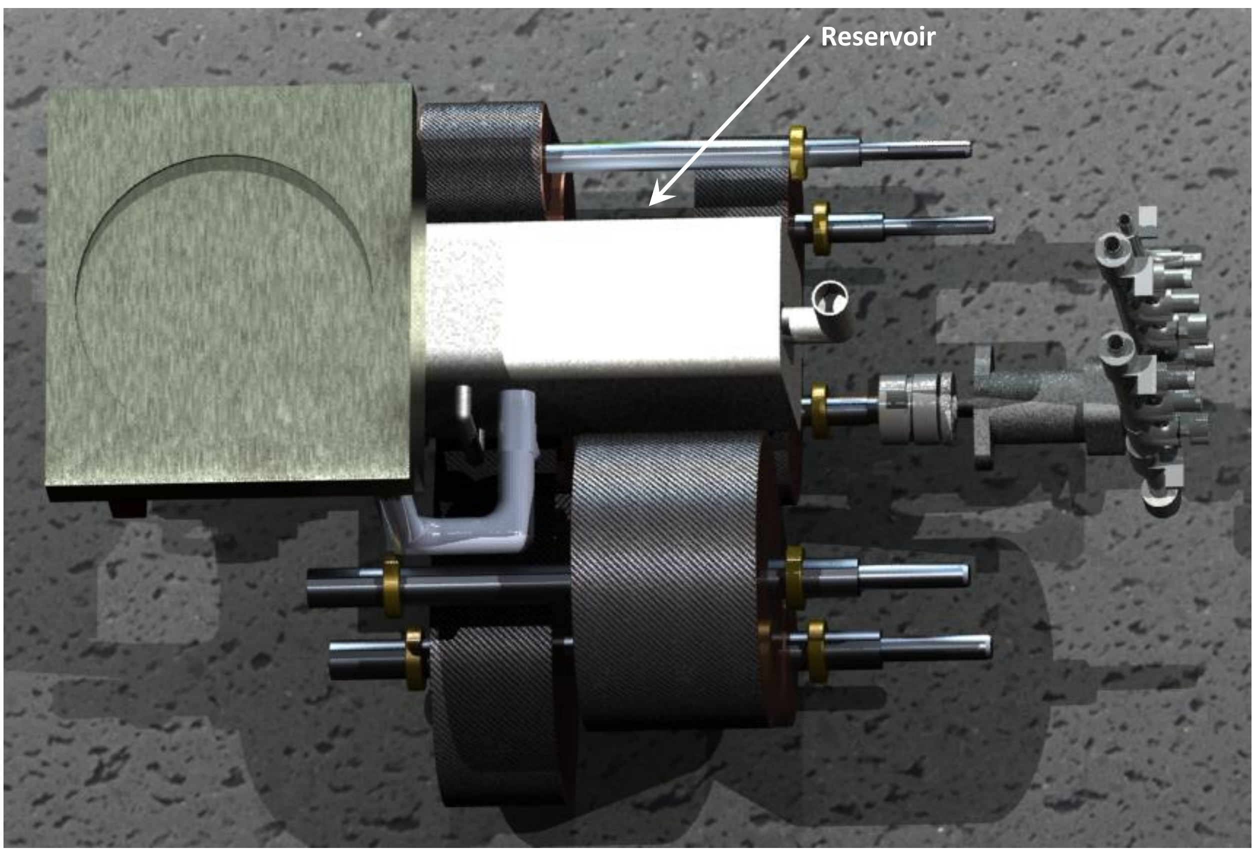


Figure 23. Reservoir orientation [2].

The load simulator will be in operation in all seasons of the year. For safe operation and design life maximization, it is recommended that the load simulator be allowed to warm up indoors to the minimum start up temperature before operation. Another option would be to implement a block heater element inside the oil reservoir. A heating element is necessary for system start-up at colder temperatures because the minimum operating temperature of the oil is 25 °F due to low viscosity as well as minimum pump operating temperatures. The oil level of the reservoir may be observed through an external tube with level measurements on it. The reservoir will be constructed with 3/8” thick mild steel, which will result in a mass of 60 kg. Assuming the reservoir is 70% full, the oil in the reservoir will weigh about 25 kg, which will bring the total reservoir weight to 85 kg.

Corrosion resistance and fabrication methods are considered in the design of the reservoir. The concept analysis stage selected stainless steel because of its excellent corrosion resistance, as described in Appendix E. However, mild steel is considered a better option because of the high maintenance costs and complex welding techniques for stainless steel. Mild steel is readily available to MacDon and in house protective coating techniques can be used to increase corrosion resistance. Welding methods should be carefully chosen to avoid localized corrosion. The manufacturing cost for the reservoir is estimated to be approximately $200 CDN. Table VI summarizes the specifications of the selected reservoir.

TABLE VI
RESERVOIR SPECIFICATIONS

Parameter	Value
Operating weight	85 kg
Width	8 inches
Height	10 inches
Length	30 inches
Thickness	3/8 inch
Cost	$200 CDN

7. Emergency Shutdown

An emergency shutdown system is required to avoid degradation of the working fluid at extreme temperatures. The fluid used is 15W-40 oil, which degrades at temperatures above 210 °F. To avoid operation above 210 °F, a temperature probe must be installed between the valve and the radiator component, as that is where the highest temperature is experienced. The location of the temperature probe in the hydraulic circuit is highlighted in Fig. 24.

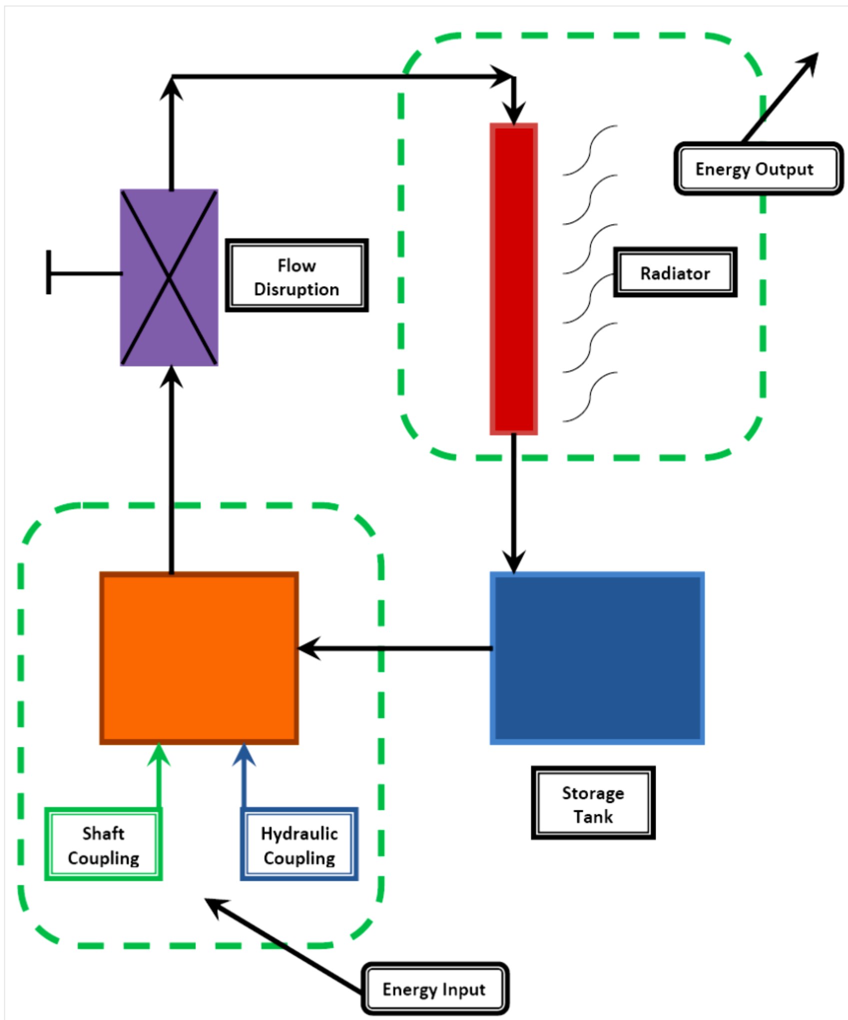


Figure 24. Load simulator hydraulic circuit [26].

When the working fluid reaches 210 °F the temperature probe will trigger an electrical system that will activate an electrical clutch disconnecting the input assembly from the pump. Selection of an electrical system is beyond the expertise of the design team. An electrical control system should be designed that can be programmed to register data from the temperature probe and signal the electrical clutch. This system could also serve to interpret the signals from the flow meter and incorporate an active switch. This active switch could be used by operators to shut down the test when required. The temperature probe shown in Fig. 25 is the 215-01-120 probe from Paine Electronics and was chosen because of its high temperature capabilities [27]. The retail price of this temperature probe is $480 CDN.

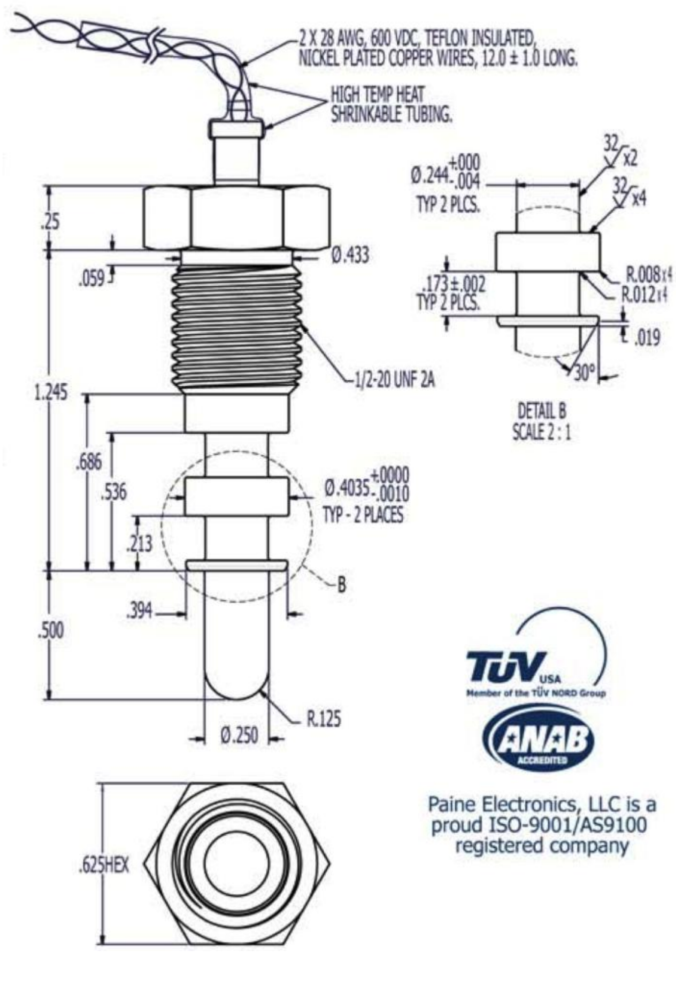


Figure 25. Temperature probe used in emergency shutdown with electric clutch [27] [permission pending].

Table VII summarizes the specifications of the selected temperature probe.

TABLE VII
TEMPERATURE PROBE SPECIFICATIONS

Parameter	Value
Operating temperature	-25°F to 500 °F
Maximum pressure	30 000 psi
Cost	$480 CDN

8. Summary of Load Simulator Design

The layout of the system was determined after all of the components were selected. The main goal of the final load simulator layout is to keep everything organized, for both maintenance and ease of use. The load simulator was requested by MacDon to be within the dimension 5’ wide by 6’ tall by 3’ deep, so it can be easily maneuvered by a forklift.

Fig. 26 below is an illustration of the final design, highlighting different orientations of the assembly.

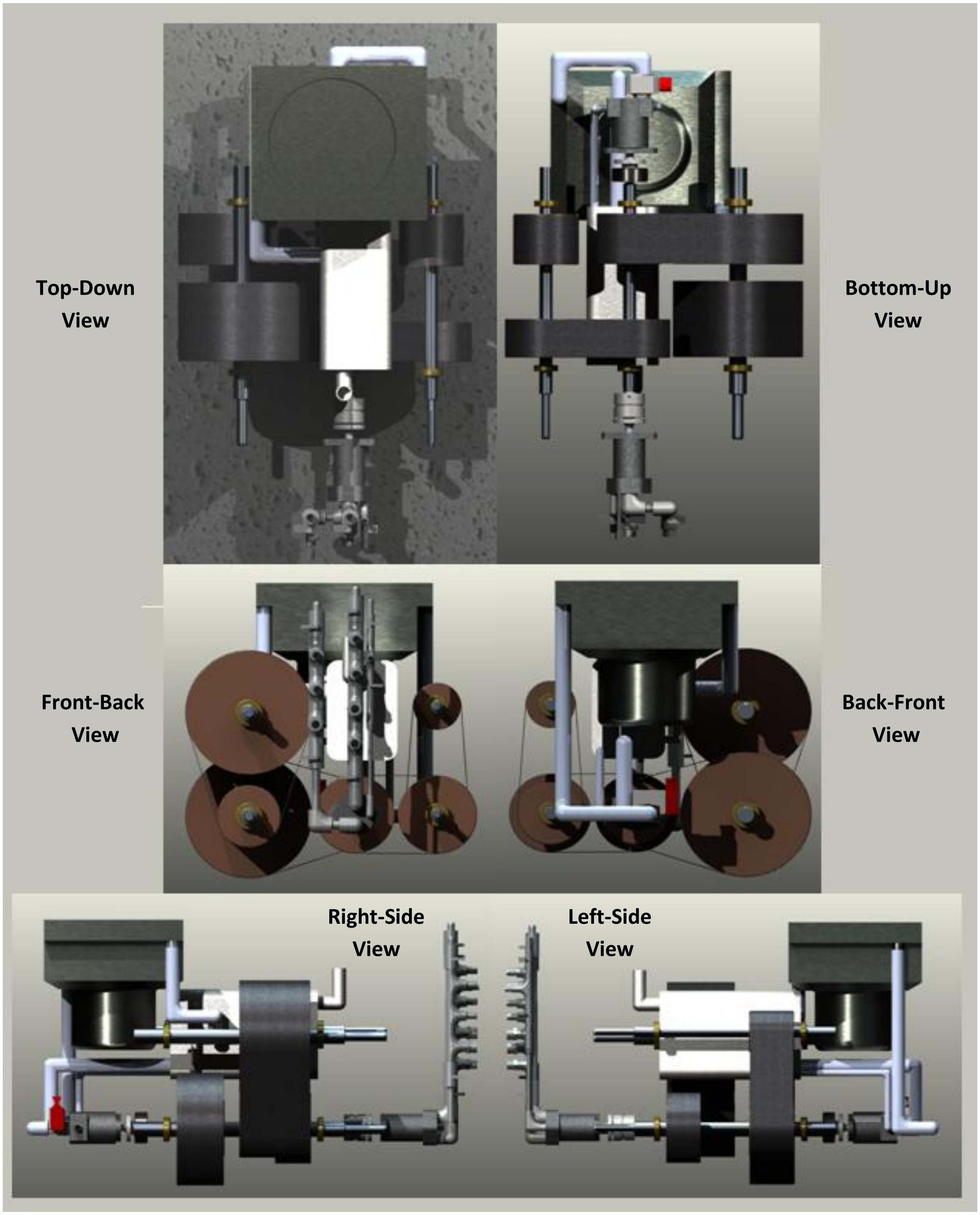


Figure 26. Design views [28].

The main component to build around in the assembly was the belt drive system. This is because it takes up the most space and has to be positioned such that it is easy to access the shafts attached. Fig. 27 and Table VIII illustrate the belt drive system.

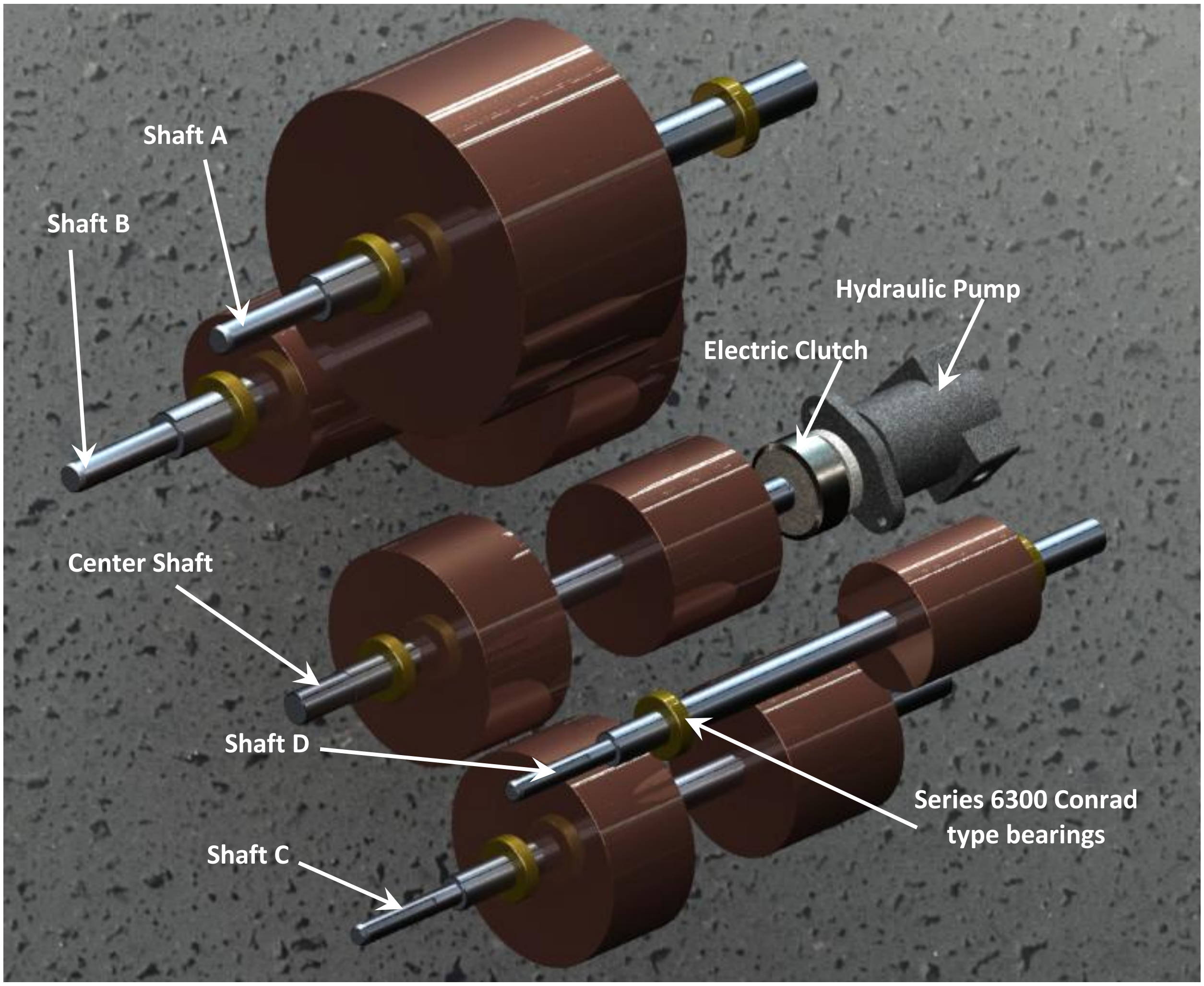


Figure 27. Labelled belt drive [29].

TABLE VIII
SHAFT SPECIFICATIONS

Shaft Designation	Shaft Speed Range	End Shaft Specifications
A	350 to 700 rpm	6 spline 1 3/8’’
B	700 to 1500 rpm	21 spline 1 3/8’’
C	1400 to 3000 rpm	1’’ keyed
D	1900 to 4000 rpm	1’’ keyed

It is recommended that the frame contain pillow blocks in order to secure the bearing attached to the shafts. The blocks provide support while maintaining the mobility needed by the shafts in order to rotate. Fig. 28 shows a pillow block.

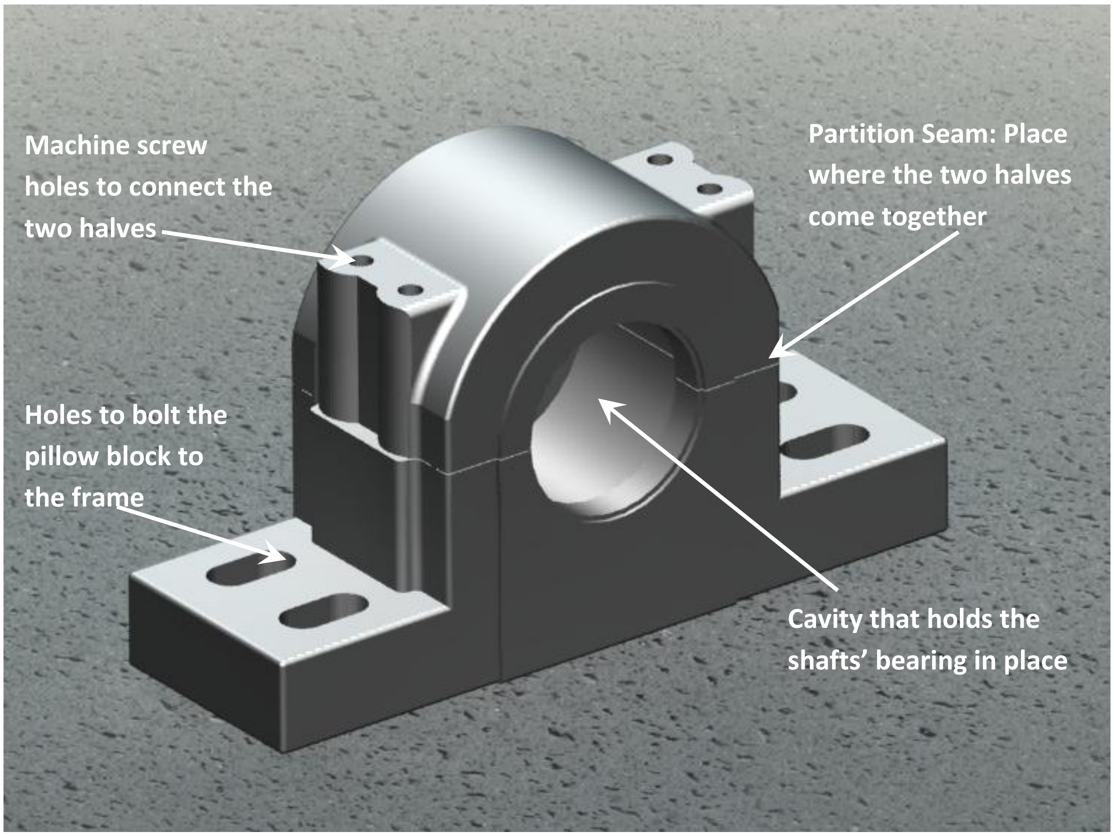


Figure 28. Pillow block [30].

The maximum dynamic load exerted by belt tension is 1700 lbs. Therefore, in order to support the bearings, the pillow blocks need to be rated for a maximum applied load of 1700 lbs. The valve was positioned next to the pump in order to reduce space and to provide convenient operating control at the rear of the assembly. Fig. 29 below shows the position of the valve.

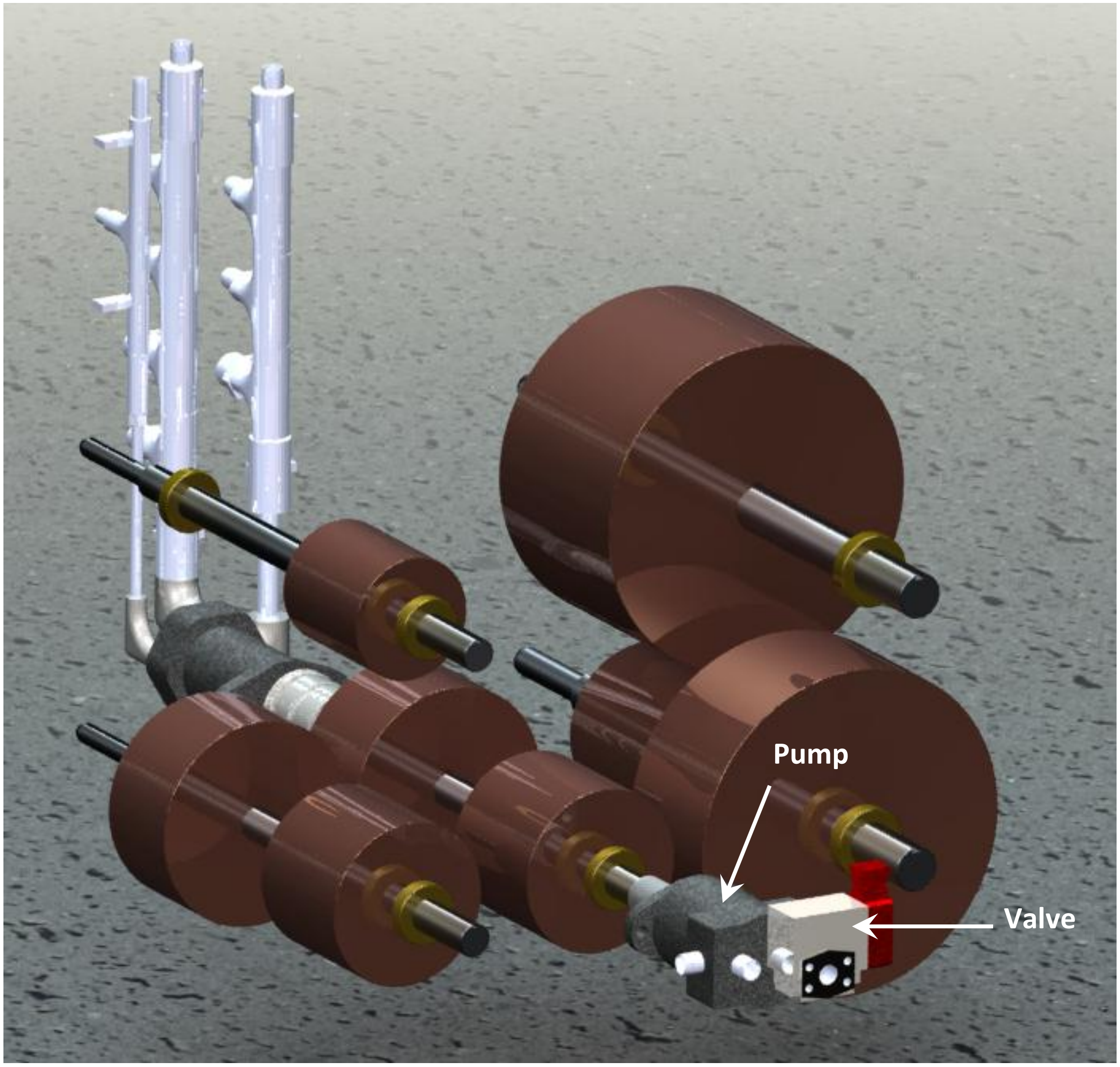


Figure 29. Valve position [31].

The reservoir is positioned between two of the shafts in order to utilize the space available. The radiator sits above the pump and valve, just behind the reservoir, and vents heat upward into the atmosphere. This location is the only position available for the radiator that maintains the overall dimensions requested by MacDon. An illustration of the radiator and the reservoir is located in Fig. 30 below.

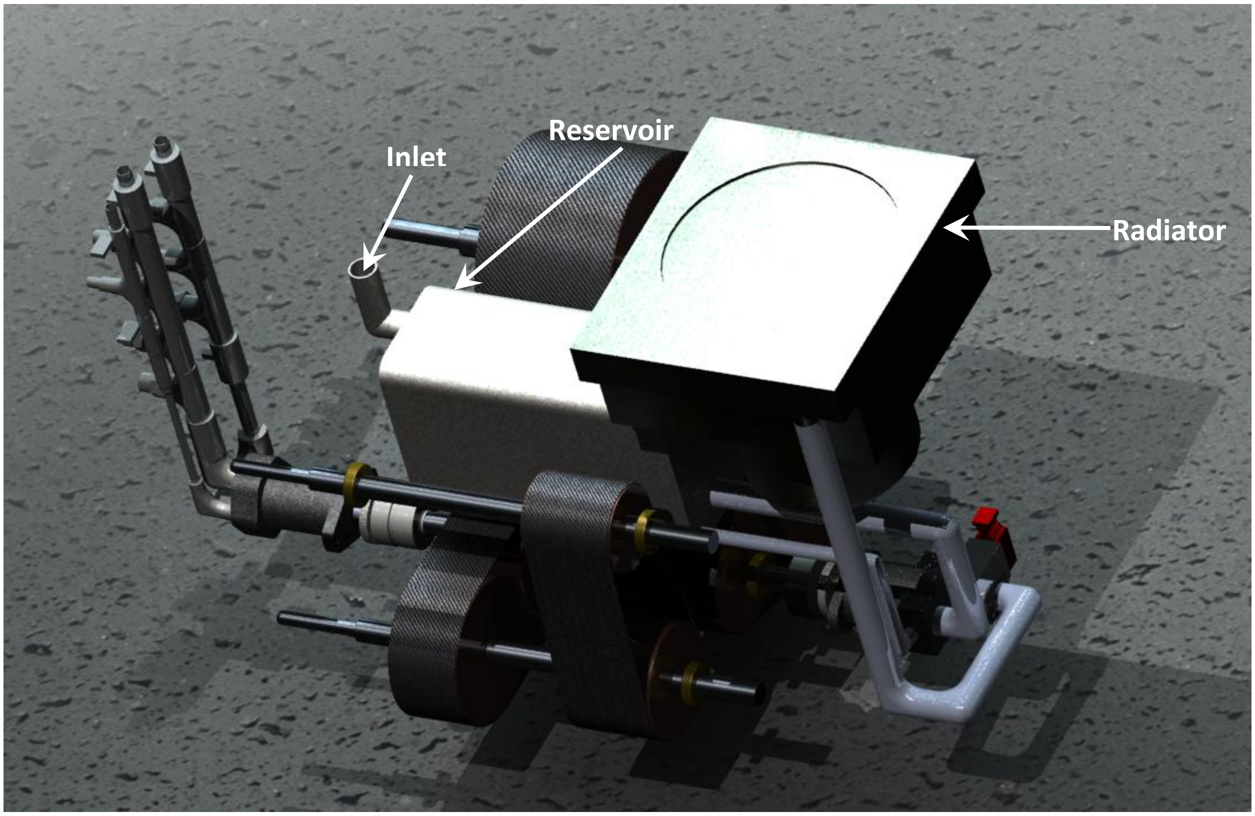


Figure 30. Reservoir and radiator orientation [32].

Overall, the approximate exterior dimensions of the load simulator are 5.5’ wide by 3’ tall by 3’ deep. The width is beyond MacDon’s dimensional limit; the other dimensions are in agreement with MacDon’s request. Fig. 31 shows the final assembly with both design dimensions and MacDon requested dimensions.

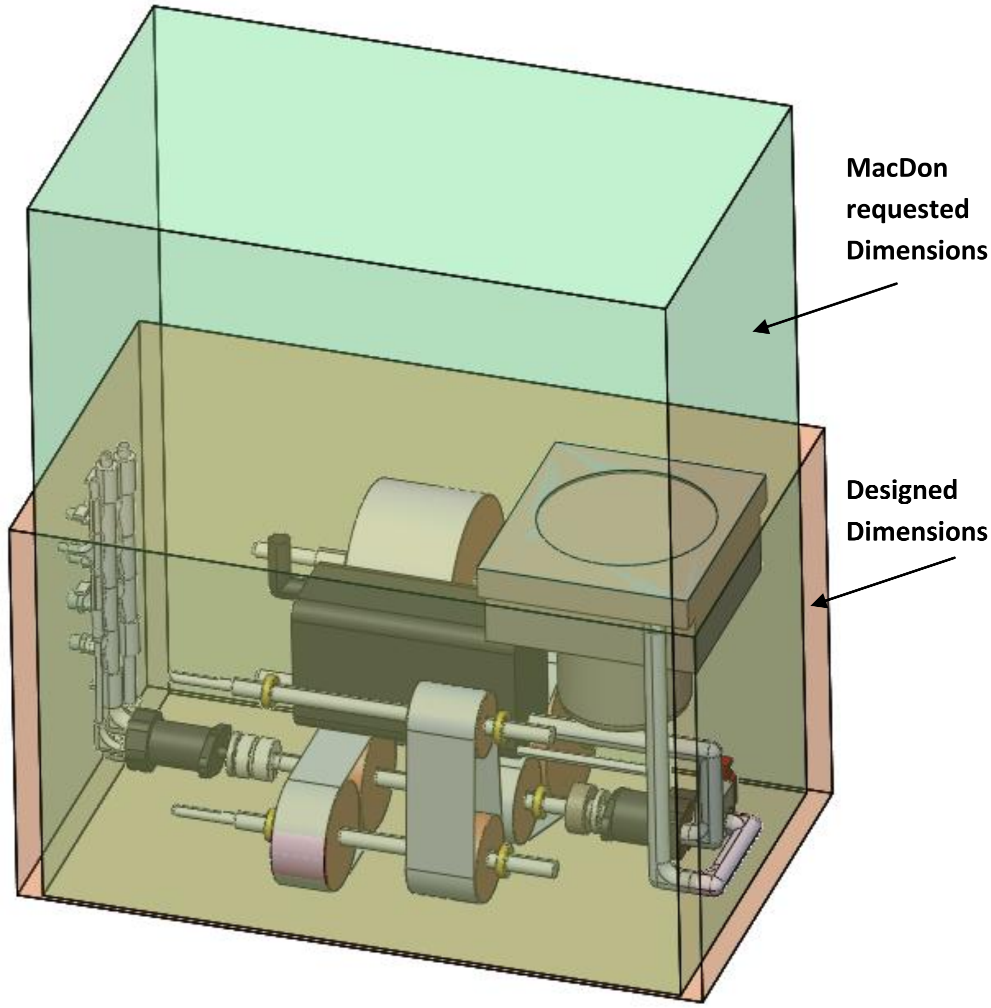


Figure 31. Dimension comparison: designed vs. MacDon requested [33].

While the green box illustrates the maximum exterior dimensions that MacDon specified, the red box shows the maximum exterior dimension of the load simulator design.

9. Discussion and Recommendations

The load simulator design incorporates many of the design features specified by MacDon Industries Ltd. These features provide for a versatile platform for testing windrower tractors and high speed hydraulic motors. The load simulator is designed to absorb up to 60 hp, as specified, through mechanical and hydraulic inputs. Energy absorption is accomplished by driving a hydraulic pump and utilizing a valve to throttle the flow. A radiator then dissipates the heat generated in the hydraulic flow. The overall specifications requested by MacDon are compared to the load simulator design features detailed in this discussion.

9.1. Cost of Load Simulator Design

MacDon specified that the cost of the load simulator should be less than $15 000. This cost should include all components, miscellaneous parts and manufacturing. In addition to the cost of the components and parts of the design, manufacturing costs must also be considered. Assuming that two welders would be manufacturing the frame at $30/hr for approximately 2 weeks, the total cost of the load simulator would be approximately $15 400. A cost summary of the major components of the design is located in Table VII below.

TABLE IX
COST SUMMARY

Section	Cost
Power input assembly	$5780 CDN
Hydraulic input assembly	$1040 CDN
Valve	$ 800 CDN
Pipes	$150 CDN
Radiator	$2150 CDN
Reservoir	$200 CDN
Temperature probe	$480 CDN
Total parts cost	$10 600 CDN
Estimated manufacturing cost	$4800 CDN
Total estimated cost	**$15 400 CDN**
Target cost	**Below $15 000 CDN**

It is recommended that MacDon carry out a detailed manufacturing cost assessment for more accurate results.

The cost of the mechanical input assembly could be greatly reduced with the application of motorcycle belts. Kevlar cog belts are commonly used in Harley-Davidson motorcycles and are designed to transmit 70 to 90 hp at speeds of up to 3000 rpm. Given that these operating parameters are very similar to those in the load simulator design, Kevlar cog belts would be suitable for use in the load simulator design. The application of these Kevlar cog belts would reduce the overall size and cost of the load simulator. Due to the project time constraints, a Kevlar cog belt design was not developed.

9.2. Pipes

The hardline pipes connecting components of the load simulator are a crucial aspect of the design. The pipes facilitate the transmission of the working fluid, which is 15W-40 oil. They must be rated for both the maximum flow rate of 57 gpm and maximum pressure of 5000 psi. Pipes are generic components and are considered readily available to MacDon. Therefore, pipes are not analyzed in detail for this design.

9.3. Power Gauging

MacDon specified that it is not necessary to gauge power dissipation accurately. However, it is required that a power reading is given. For power gauging, a flow meter and pressure gauge are needed. Therefore, it is recommended that an analog pressure gauge be mounted between the hydraulic pump and the valve. From flow and pressure readings, the amount of power dissipation can be read from both metric and imperial figures included in Appendix C. This illustrates the system operating curves used for power estimation.

9.4. Mechanical and Hydraulic Inputs

MacDon specified that that two shaft inputs should be available to input mechanical power into the load simulator. The two shaft inputs specified include a 21 tooth spline and a 1’’ keyed shaft. MacDon also specified that that shaft inputs for speeds up to 4000 rpm should be available. To accomplish these requirements, the load simulator design incorporates four shaft inputs with the following specifications:

1. 1’’ keyed shaft with a speed range of 1900 to 4000 rpm.
2. 1’’ keyed shaft with a speed range of 1400 to 3000 rpm.
3. 1 3/8’’ 21 spline shaft with a speed range of 700 to 1500 rpm.
4. 1 3/8’’ 6 spline shaft with a speed range of 350 to 700 rpm.

The mechanical input connections are designed to transmit up to 60 hp. MacDon requires the load simulator to last up to 50 000 of operation. It has been estimated that the series 6300 Conrad type bearings would last at least 10 000 hours. Therefore, it is recommended that bearings be greased after 100 hours of operation, and visually inspected after 3000 hours. Given this recommended maintenance schedule, warnings of bearing breakdown should be detected before failure. For tests that could last up to 6 weeks, the load simulator will need to be shutdown accordingly for minor maintenance in order to avoid risk of failure.

For hydraulic circuit testing, MacDon specified a series of hydraulic couplers that would be required in order to make the necessary connections. The load simulator design possesses the following couplers:

1. three 1 1/16'' 12 UNF couplers for hydraulic pressure.
2. three 1 1/16'' 12 UNF couplers for hydraulic return.
3. one 1 11/16'' 12 UNF coupler for hydraulic pressure.
4. one 1 11/16'' 12 UNF coupler for hydraulic return.
5. one 1 5/16'' 12 UNF coupler for hydraulic pressure.
6. two ¾'' 16 UNF couplers for case drain.

These hydraulic couplers allow for the attachment of quick coupler connections that are designed for standard hydraulic flex-lines, which MacDon intends to use. Therefore, the above list of couplers meets MacDon specification. Exact components for these couplers were not chosen, since they are interchangeable and non-crucial components that are known to be readily available.

Finally, MacDon had specified that it must be possible to switch between mechanical and hydraulic input within one minute. The load simulator design incorporates few components that need to be manipulated to accomplish a rapid switch between input types. This switching process includes turning three ¼-turn valves and actuating a hydraulic clutch, which can be done in less than a minute.

9.5. Size and Mass

The load simulator design is a movable self-contained testing platform, and is estimated to weigh less than 1000 kg. Approximate exterior dimensions were given to be 5 ft long by 3 ft wide by 6 ft high. The load simulator design in this report is 5.5 ft long by 3 ft wide by 3 ft high. Given these design features, the load simulator is suitable for transport via forklift, as specified.

Given the shaft and belt drive systems in the design, safety panels would be required on all sides. Since the load simulator is a portable system that does not need to fit into a specific area of a building, it is expected that increasing the each exterior dimension will carry little consequence. MacDon gave the initial specification of 5 ft long by 3 ft wide by 6 ft as a preliminary guideline. A load simulator 6 ft in length, instead of 5ft, is expected to be a small inconvenience with regards to moving the design through shop doors.

9.6. Operating Temperature

MacDon specified that the load simulator must be able to operate in the temperature extremes of Manitoba with 15W-40 oil. Given the viscous properties of 15W-40 and the requirements outlined by the hydraulic pump, the minimum start up temperature for the load simulator was determined to be 25 °F, with an optimum operating oil temperature of 140 °F. For safe operation and design life maximization, it is recommended that the load simulator be allowed to warm up indoors to the minimum start up temperature before operation. Another option discussed is implementing a block heater element inside the oil reservoir. This element could be used during the colder months in a similar manor as an automobile.

9.7. Temperature Control and Electrical Systems

MacDon requested that the design include a system that would prevent sustained operation above 190 °F and would automatically shutdown safely if the oil temperature reached 210 °F. The load simulator design incorporates a temperature probe, which is capable of fully regulating the oil temperature via the radiator. The temperature probe is used to signal an electric clutch to disconnect the shaft from the hydraulic pump.

An electrical control system to regulate the temperature and actuate the safety shutdown was not designed. Therefore, it is required that MacDon construct a design internally for this crucial electrical system. MacDon requested that all electrical components be compatible with 12 Volt systems from the tractors they would be testing. The specified temperature probes and radiator motor are 12 Volt systems. However, the electric clutch is a 24 Volt system. It is recommended that an electrical control system be designed that can be programmed to register data from the temperature probe and signal the electrical clutch. This electrical control system would also be able to account for the required 24 Volt output. Incorporating this feature is expected to be a minor inconvenience. This system could also serve to interpret the signals from the flow meter and incorporate an active switch. This active switch could be used by operators to shut down the test when required.

10. Design Conclusion

The load simulator design is estimated to be slightly over the specified budget of $15 000 CDN. The total estimated cost of $15 400 CDN is within a 5% range of the budget and can therefore be assumed negligible. While, the length of the design is 1 foot longer than specified, this problem can be resolved with the application of Kevlar cog belts. Kevlar cog belts are narrower than Micro-V belts and would result in a shorter overall design. This would also decrease the expected mass well below 1000 kg.

The application of Kevlar cog belts would not interfere with the design of the mechanical and hydraulic inputs that closely meet MacDon's specifications. The design is capable of dissipating 60 hp and switching between hydraulic and mechanical inputs within one minute.

It is recommended that the load simulator be allowed to warm up to 25 °F after exposure to extreme winter temperatures. Installing a block heater element in the reservoir would serve to speed up this heating process.

Finally, it is recommended that an electronic control system be designed to regulate operating temperature and actuate the safety mechanism. This system will be required to output 24 Volts for the electric clutch that serves to shutdown the load simulator at oil temperatures in excess of 210^{o}F.

References

[1] MacDon Industries LTD. (Private Communication), Sept 20, 2011.

[2] S. Peters. (2011, November 24). AssemblyV5.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[3] K. Morgenstern. (2011, November 12). Powerinput.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[4] K. Morgenstern. (2011, November 12). Mechanicalinput.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[5] K. Morgenstern. (2011, November 12). Taperlockbushing.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[6] K. Morgenstern. (2011, November 12). Demonstrationsheave.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[7] K. Morgenstern. (2011, November 12). Hydraulicpumpview.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[8] K. Morgenstern. (2011, November 12). Pump.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[9] Sauer Danfoss. (2010, Aug.). “Gear Pumps and Motors” [Online].

[10] K. Morgenstern. (2011, November 12). Smallshaft.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[11] K. Morgenstern. (2011, November 12). Largeshaft.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[12] K. Morgenstern. (2011, November 12). Centreshaft.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[13] Gates Heavy Duty V-Belt Drive Design Manual, Gates Standard 14995-A, 1999.

[14] K. Morgenstern. (2011, November 14). L-sectionbelt.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[15] Harley-Davidson Winnipeg (Private Communication), Nov. 25, 2011.

[16] K. Morgenstern. (2011, November 14). Electricclutch.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[17] Ortlinghaus. ("no date"). Electromagnetically actuated clutches and brakes, clutch/brake combined units and tooth clutches [Online].
Available: http://www.ortlinghaus.co.uk/p_electroclutches.html [November 18, 2011].

[18] K. Morgenstern. (2011, November 14). Hydraulicinputassembly.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[19] K. Morgenstern. (2011, November 14). Motor.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[20] Eaton. (2011, Oct.). "Medium Duty Piston Motors" [Online]. Available: http://hydraulics.eaton.com/products/motors_main.htm [November 11, 2010].

[21] K. Morgenstern. (2011, November 14). Clutch.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[22] Logan Clutch Corporation. ("no date"). P Series [Online]. Available: http://www.loganclutch.com/PSeries [November 14, 2011].

[23] Parker Hannifin Corp 2011. ("no date"). "3-EN2900-C.pdf" [Online]. Available: http://www.denisonparker.com/Catalogos/VALVE_Sales_Publications/Pressure%20Controls/3-EN2900-C.pdf

[24] M. Schaefer. (2011, Nov. 28). "RE: Hydraulic Oil Cooling Inquiry." Personal email.

[25] M. Taylor. (2011, Nov. 16). "RE: Hydraulic Oil Cooling Inquiry." Personal email.

[26] K. Morgenstern. (2011, October 16). PowerCycle.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[27] Paine Electronics, LLC. (2011, Nov). “215-01-120 Temperature Probe” [Online]. Available: www.paineelectronics.com/temperatureprobes.htm. [Nov. 19].

[28] S. Peters. (2011, November 16). AssemblyWithBelts.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[29] S. Peters. (2011, November 16). BeltDrive.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[30] S. Peters. (2011, November 16). PillowBlock.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

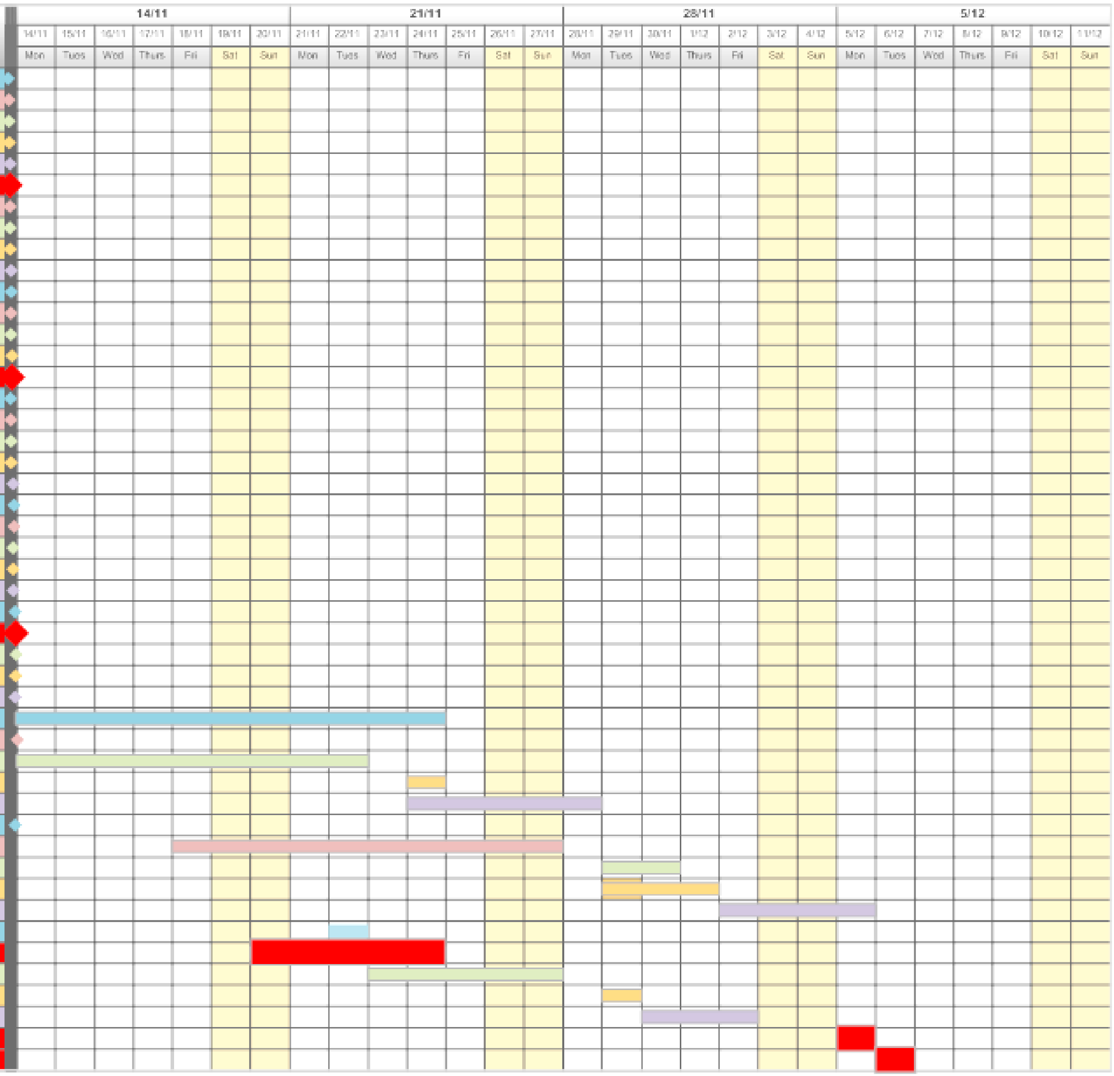
[31] S. Peters. (2011, November 16). AssemblyV5.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[32] S. Peters. (2011, November 18). FinalAssembly.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[33] S. Peters. (2011, November 18). DimensionComparison.sldprt. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

Appendix A: Full Gantt Chart Phase 3

Number	Task	Start	End	Duration
1	Introduction, Project Selection	8/9/2011	12/9/2011	5
2	Team forming, Establishing Expectations	13/9/2011	13/9/2011	1
3	Individual Research on Load Cells	13/9/2011	15/9/2011	3
4	Contact MacDon, Generate Questions	15/9/2011	20/9/2011	6
5	Establish Project Management Practices	20/9/2011	21/9/2011	2
6	Milestone: Meet With MacDon	20/9/2011	20/9/2011	1
7	Redefine Project and Research	20/9/2011	22/9/2011	3
8	Discuss Project Definition, Distribute Work	20/9/2011	20/9/2011	1
9	Writing Project Definition Report	21/9/2011	27/9/2011	7
10	Compile Project Definition and Review	28/9/2011	29/9/2011	2
11	Follow Up With MacDon	29/9/2011	30/9/2011	2
12	Discuss Oral Presentation	26/9/2011	26/9/2011	1
13	Work on Oral Presentation	26/9/2011	1/10/2011	6
14	Present Oral Presentation	4/10/2011	7/10/2011	4
15	Due: Project Definition, Peer Evaluation	3/10/2011	3/10/2011	1
16	Meet With Professor Soliman	23/9/2011	23/9/2011	1
17	Team Meeting: MacDon Questions	26/9/2011	26/9/2011	1
18	Submit Log Book, Minutes & Gantt	27/9/2011	27/9/2011	1
19	Brain Storm Technology Concepts	27/9/2011	17/10/2011	21
20	Submit Log Book, Minutes & Gantt	11/10/2011	11/10/2011	1
21	Follow Up With MacDon	17/10/2011	17/10/2011	1
22	Submit Log Book, Minutes & Gantt	18/10/2011	18/10/2011	1
23	Discuss Concept Design Report	11/10/2011	11/10/2011	1
24	Finalize Design	11/10/2011	14/10/2011	4
25	Writing Concept Design Report	12/10/2011	22/10/2011	11
26	Compile Concept Design Report	24/10/2011	27/10/2011	4
27	Due: Submit Concept Design Report	28/10/2011	28/10/2011	1
28	Follow Up With MacDon	31/10/2011	31/10/2011	1
29	Submit Log Book, MM, Gantt, P. Evaluation	28/10/2011	28/10/2011	1
30	Subsystem Level Design	25/10/2011	7/11/2011	14
31	Writing Final Report	7/11/2011	24/11/2011	18
32	Submit Log Book, Minutes & Gantt	8/11/2011	8/11/2011	1
33	Team Revision of Final Report	14/11/2011	22/11/2011	9
34	Discuss Final Oral Presentation	24/11/2011	24/11/2011	1
35	Write Final Oral Presentation	24/11/2011	28/11/2011	5
36	Brainstorm Poster	25/10/2011	25/10/2011	1
37	Generate and Print Poster	18/11/2011	27/11/2011	10
38	Compile Final Oral Presentation	29/11/2011	30/11/2011	2
39	1st Rehearsal of Final Oral Presentation	29/11/2011	1/12/2011	3
40	2nd Rehearsal of Final Oral Presentation	2/12/2011	5/12/2011	4
41	Submit Log Book, Minutes & Gantt	22/11/2011	22/11/2011	1
42	Compile Final Report	20/11/2011	24/11/2011	5
43	Compiled Report Revision	23/11/2011	27/11/2011	5
44	Submit Log Book, Minutes & Gantt	29/11/2011	29/11/2011	1
45	Redistributed Final Report to Teams	30/11/2011	2/12/2011	3
46	Submit Final Report	5/12/2011	5/12/2011	1
47	Final Oral Presentation	6/12/2011	6/12/2011	1



Appendix B: Power Input Assembly Design Details

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Introduction

A number of designs were considered before the final power input assembly was chosen. Numerous criteria, largely pertaining to initial cost, maintenance and ease of use contributed to the final selection. This technical analysis details how the final design of the power input assembly was derived. This section also includes suggestions on how the power input assembly may be manufactured and gives a detailed cost breakdown of individual components.

Technical Analysis of Components

The rated horsepower of the components increases with each system from the input. This is shown by a service factor for each component. The design horsepower represents what each component will actually experience. The equation below illustrates this service factor.

$$Service\ factor = \frac{Rated\ horsepower}{Design\ horsepower}$$

For example, the 126 hp hydraulic pump, which is at the end of the system, has a service factor of 2.1. Alternatively, the belt drive system, which is earlier on in the design, has at a service factor of 1.5. Actual power dissipation by the pump can be estimated by the figures in Appendix C. The following are two main reasons why this is the general case.

1. Components near the output end of the load simulator are more expensive than at the input end. Therefore, in the event of a malfunction, an inexpensive belt would simply break, rather than rupturing the hydraulic pump.

2. In the event of an accidental power overload into the load simulator, fewer components would be strained by the overload since the failure would occur near the input end of the power input assembly.

Components such as the hydraulic motor and pump come with detailed specification on how much torque they can transmit. Other components such as the belts, sheaves and shafts are selected from standards or designed in house.

The belt drive design manual by Gates is the primary source for selecting a specific belt drive system [1]. For a simple and standard belt design, the following parameters were followed:

1. The maximum sheave speed must be less than 6500 ft/min.
2. The sheave diameters must be between 3 and 18’’.
3. For layout practicality, the maximum width of a belt drive should be less than 15’’. Note that each rib on an L-section Micro-V belt is 3/16’’ in width for a total of 6 ribs.

A service factor of 1.5 was selected for the belt drive, which is recommended for a driven machine such as a hydraulic gear pump [1]. Given a service factor of 1.5, the belt drive was designed to transmit 90 hp. From the belt standards outline by Gates, an L-Section Micro-V belt was chosen; this belt is shown in section 3.1.2. To accommodate the range of shaft speeds, speed ratios relatively close to 1:1 were chosen. Given the belt standards, speed ratios closer to 1:1 allow for a smaller overall design. Sheave diameters were chosen based on the selected ratios of 2:1, 1:1 and 3:4, respectively.

In order to minimize the number of belts needed, large sheaves were selected, which also maximized the horsepower rating per rib of belt. The large sheaves resulted in a trade-off between sheave diameter and sheave width. The specifications and dimensions of the selected sheaves are shown in Table B-I below.

TABLE B-I
SPECIFICATIONS AND DIMENSIONS FOR SELECTED BELT DRIVES

From/To Shaft	Speed Ratio	Sheave Diameters	Center Distance	Belt Length	Rated hp/Belt Rib	Number of Belts	Total Sheave Width
A to B	2:1	8''/16''	13.4''	65.5''	2.59	6	10.1''
B to Center	2:1	8''/16''	13.4	65.5''	4.62	4	6.7''
C to D	1:1	10''/10''	14.0''	50.0''	4.13	4	6.7''
C to Center	3:4	6''/8''	10.4''	50.0''	5.64	3	5.1''

Given these belt selections, the parameters of initial cost, maintenance, and ease of use were met. Gates estimates that belt drives should last up to 20 000 hours of operation based on their design standards [1]. However, routine inspection of belts on the load simulator should be considered to prevent catastrophic failure.

For design of the working shafts, special considerations must be given with respect to torque, bending moments and bearing loads generated by the belts. Gates outlines a series of equations for calculating shear forces on shafts. Appendix E details the specific torques, bending moments and bearing loads for each shaft. The calculations below outline the design of shaft A, as an example.

A minimum rotational speed of 350 rpm is required for this shaft to transmit the maximum power of 60 hp. The maximum torque was found to be 10 800 lbin, shown by the following calculations [2].

$$Torque\ [lbin] = 63000 * \frac{Power\ [hp]}{Speed\ [RPM]}$$

$$10800\ [lbin] = 63000 * \frac{60\ [hp]}{350\ [RPM]}$$

This value represents the maximum torque that the shaft will experience. The service factor is applied after calculating the required diameter.

The belt design manual by Gates outlines equations for calculating the tension of the slack side, $ts_,$ and the tension of the tight side, t_t, for Micro-V belts [1]. The following equation represents the maximum transverse load, t_{tot}, on the shaft.

$$t_{tot} = t_s + t_t$$

$$2291.4\ [lb] = 563.7[lbs] + 1727.7\ [lbs]$$

Each shaft is supported by two 6300 Conrad type bearings. Shafts A and B use series 6311 bearings since they have a diameter of 2 3/16''. Shafts C, D, and the center shaft have smaller diameters of 1 5/8'', and use series 6309 bearings. Given the loads on the shafts, it is estimated that all bearings should last for more than 10 000 hours of operation. Conrad bearings use ball bearings, therefore they have moderate endurance to loads along the shaft cross section, referred to as thrust loads [2]. Thrust loads are not taken into account in assessing the life of the bearings. Power will be transferred into the mechanical input assembly via universal jointed shafts. The universal joints will be reasonably lubricated, thus the thrust loads are expected to be negligible.

It is assumed that the edge of the sheave is placed 1'' from the bearing on the shaft and that the maximum transverse load is applied where the centre of the sheave is. Given that the sheave on shaft A is 10.1'' wide, the load is applied 6.05'' from the closest bearing. Fig. B-1 illustrates a free body diagram that was used in calculating the shear forces inside the shaft.

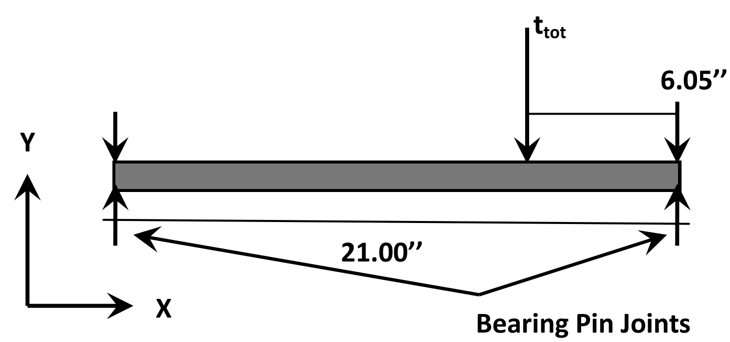

Figure B-1. Free body diagram of shaft A. Arrows indicate location of loads and bearing joints [3].

From the location of the applied load and the bearing fixtures, the reaction forces at each bearing fixture are calculated. The reaction forces are illustrated in the shear force diagram in Fig. B-2 below.

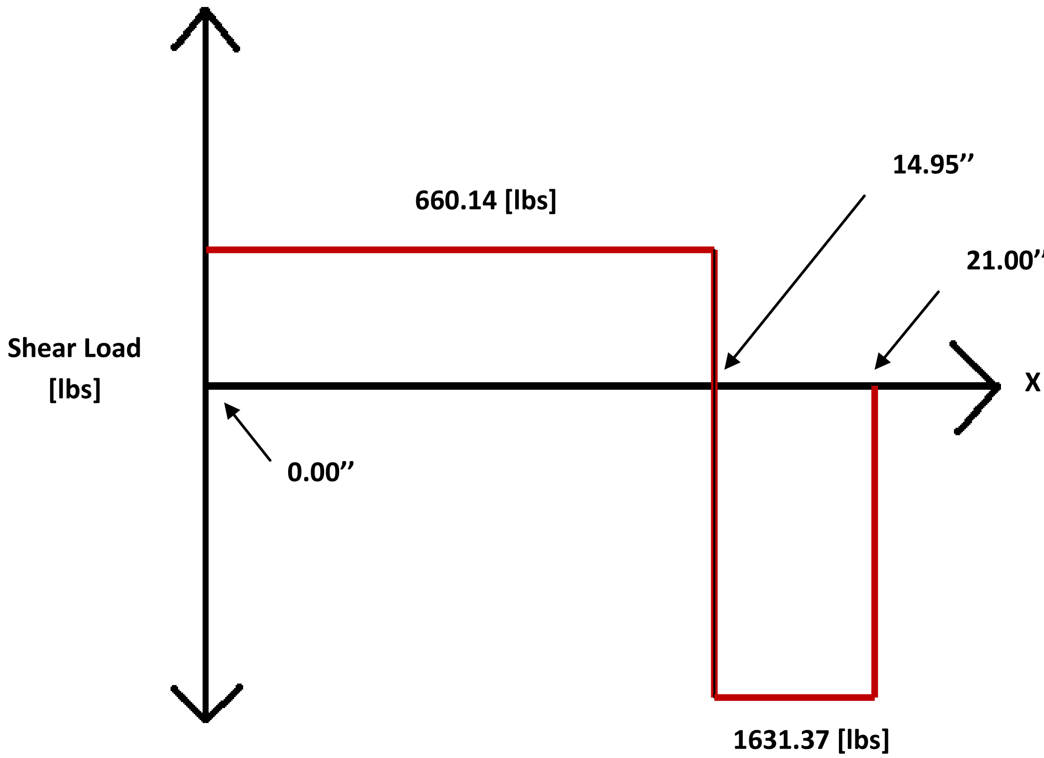

Figure B-2. Shear force diagram for shaft A [4].

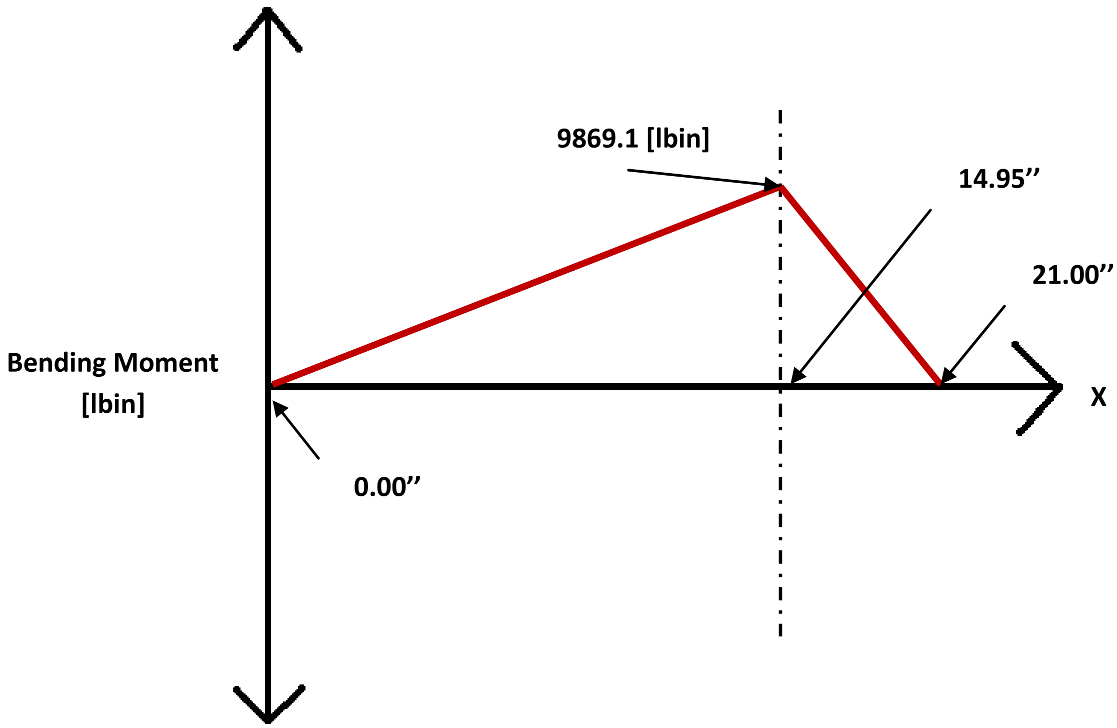

Figure B-3. Bending moment diagram for shaft A [5].

From Fig. B-2 and B-3 it can be seen that the maximum shear force and bending moments are 1631 lbs and 9861.1 lbs, respectively. From these values, the 'Basis for Shaft Design' equation for repeated reversed bending and steady torsional shear stress can be applied [2]. This equation solves for shaft diameter and assumes that the shaft is a solid structure, which is not hollow and is detailed below.

$$\left(\frac{\sigma}{s'_n}\right)^2 + \left(\frac{\tau}{S_{ys}}\right)^2 = 1$$

One variable needed to determine the shaft diameter is the torsional yield strength. The torsional yield strength of a metal can be expressed as a function of yield strength through distortion energy theory. This theorem is expressed by the following equation.

$$S_{ys} = \frac{S_y}{\sqrt{3}}$$

By combining these two theories, it is possible to solve for an equation to find the diameter, D, for a shaft given specific loading conditions and dimensions. A detailed process of deriving this diameter equation is shown in [2]. This equation is expressed below.

$$D = \left\{\left[\frac{32 * N}{\pi}\right]\sqrt{\left(\frac{K_t M}{s'_n}\right)^2 + \frac{3}{4}\left(\frac{T}{s_y}\right)^2}\right\}^{\frac{1}{3}}$$

A number of scaling factors are applied to variables in deriving the diameter equation. Table B-II below details the nomenclature in the diameter equation.

TABLE B-II
NOMENCLATURE OF DIAMETER EQUATION FOR SHAFT DESIGN [2]

Variable	Meaning	Units
D	shaft diameter	inches
N	factor of safety (for shaft design 2)	none
K_t	stress concentration factor	none
M	maximum bending moment	lbin
T	maximum applied torque	lbin
s'_n	actual endurance strength	psi
s_y	yield strength	psi

Given the above diameter equation, the minimum diameter for the shaft was found to be 2.14''. This value is rounded up to a standard shaft size of 2 3/16''. The calculation for all shaft diameters is shown in the Appendix D.

Since the overall design of the load simulator is composed of components that MacDon is familiar with and uses regularly, no special considerations are given to the manufacturing of the overall system. Should MacDon choose to implement this design, all components would be assembled in house. Therefore, no manufacturing cost is considered, since the cost would be representative to operations at MacDon.

Detailed Component Costs Breakdown

A detailed cost breakdown for each component is shown in Table B-III below. While costs for a few components are based on standards and suppliers costs, all others are based on estimates from experience and education.

TABLE B-III
COST BREAKDOWN FOR POWER INPUT ASSEMBLY

Part	Quantity	Unit Cost $CDN	Total Cost $CDN
CP180-045 Gear Pump from Sauer Danfoss [6]	1	450	450
Custom Designed 1 5/8’’shaft, 36’’ long	2	200	400
Custom Designed 2 3/16’’ shaft, 36’’ long	2	350	700
Custom Designed 1 5/8’’ shaft, 30’’ long with 6’’ end	1	250	250
Micro-V stock sheaves from Gates Corporation [1]	8	160	1280
L-section Micro-V belts from Gates Corporation [1]	4	125	500
Model 0812-0015 electric clutch from Ortlinghaus [7]	1	250	250
Model 74624/74644 Piston Motor from Eaton [8]	1	350	350
Model 350P hydraulic clutch from Logan Corporation [9]	1	300	300
Miscellaneous Components for shafts (Keys, bearings, bushings, lock rings etc.)	-	-	~900
Miscellaneous Components for hydraulic (couplings, tees, elbows, quarter turn valves etc.)	-	-	~400
Total			**$5780**

It should be noted that the cost of the above components might vary in regards to availability and location of supplier. Additional components may be required in accordance with local manufacturing and structural standards.

References

[1] Gates Heavy Duty V-Belt Drive Design Manual, Gates Standard 14995-A, 1999.

[2] L.M. Robert, Machine Elements in Mechnaical Design, 4th ed. NJ: University of Dayton, 2004.

[3] K. Morgenstern. (2011, November 17). ShaftFBD.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[4] K. Morgenstern. (2011, November 17). Shearforcediagram.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[5] K. Morgenstern. (2011, November 17). Bendingmomentdiagram.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[6] Sauer Danfoss. (2010, Aug.). “Gear Pumps and Motors” [Online]. Available: http://www.sauer-danfoss.com/Products/PistonPumpsandMotors/index.htm [November 17, 2010].

[7] Ortlinghaus. (“no date”). Electromagnetically actuated clutches and brakes, clutch/brake combined units and tooth clutches [Online]. Available: http://www.ortlinghaus.co.uk/p_electroclutches.html [November 18, 2011].

[8] Eaton. (2011, Oct.). “Medium Duty Piston Motors” [Online]. Available: http://hydraulics.eaton.com/products/motors_main.htm [November 11, 2010].

[9] Logan Clutch Corporation. (“no date”). P Series [Online]. Available: http://www.loganclutch.com/PSeries [November 14, 2011].

Appendix C: System Operating Curves for Estimating Pump Power

List of Figures

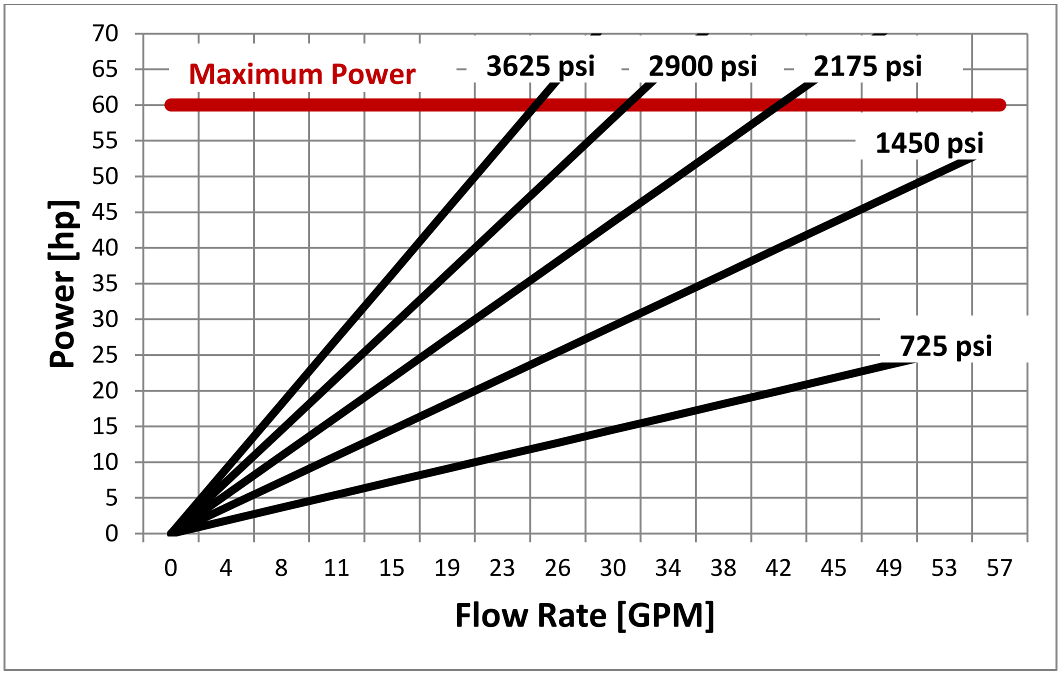

Figure C-1. System operating curve in imperial units [1].

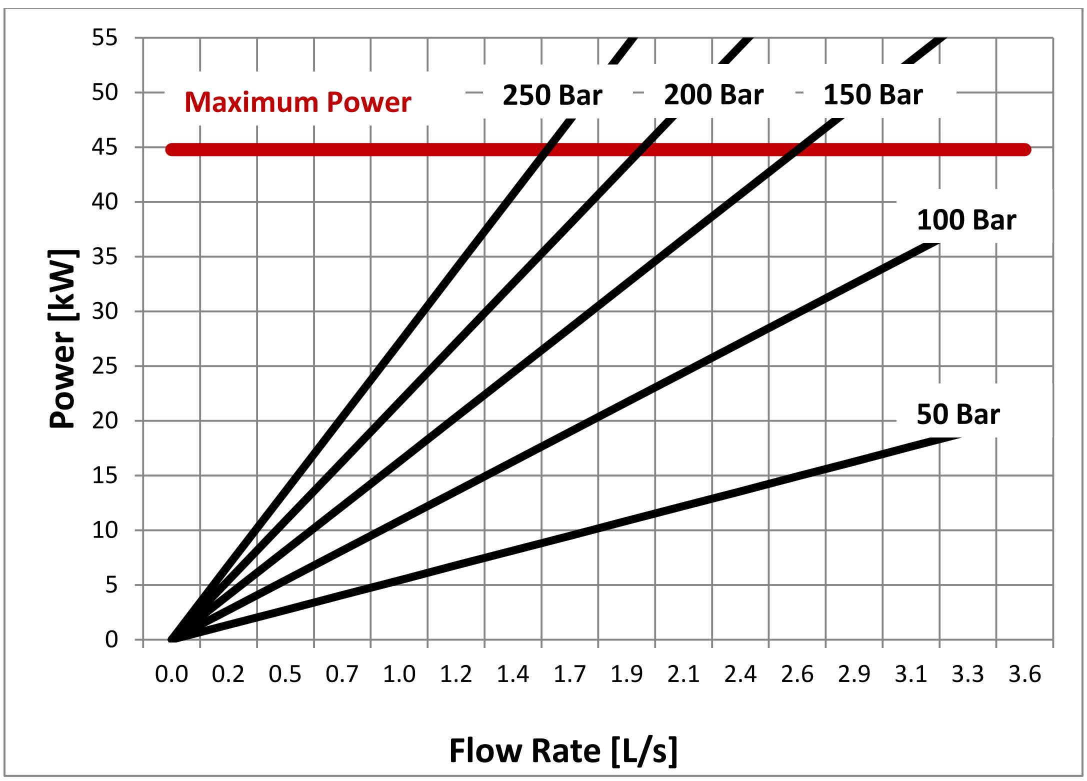

Figure C-2. System operating curve in metric units [2].

References

[1] K. Morgenstern. (2011, November 28). ImperialOperatingCurve.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[2] K. Morgenstern. (2011, November 28). MetricOperatingCurve.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

Appendix D: Shaft Calculations

List of Figures

	A	B	C	D	E	F	G	H	I	J	K	L	M	N
1		Calculation of Belt Tension												
2		All Units Imperial: lbs, in or lbin												
3														
4		Small Sheave	Large Sheave	N1	N2	C	V	D2-D1/	Arc Cont.	K PHI	HP	Tt	Ts	Tt+Ts
5	CD	6	8	1900	1400	14	3475.6	0.14	171.5	0.97	60	783.1	211.4	994.5
6	AB	8	16	700	350	13.4	1707.3	0.6	145	0.895	60	1727.7	563.7	2291.4
7	BO	8	16	1400	700	13.4	3414.6	0.6	145	0.895	60	863.8	281.8	1145.7
8	CO	10	10	1400	1400	10.4	4268.3	0	180	1	60	618.5	153.1	771.6
9														
10		N1 is speed of small sheave												
11		N2 is speed of large sheave												
12		C is center distance												
13		V is sheave velocity												
14		*Note D2-D1/C, Arc Cont. And K PHI are outline by Gates Micro-V Belt Standards												
15		Tt is tight side tension												
16		Ts is slack side tension												

Figure D-1. Values used for finding tight and slack tension in Micro-V belts [1].

	A	B	C	D	E	F	G	H	I	J	K	L	M
1		Calc											
2		All											
3													
4		Sma	Large	N1	N2	C	V	D2-D1/C	Arc Cont.	K PHI	HP	Tt	Ts
5	CD	6	8	1900	1400	14	=(B5*D5)/3.28	=(C5-B5)/F5	171.5	0.97	60	=44000*(K5/(J5*G5)	=33000*(1.33-J5)*(K5/(J5*G5))
6	AB	8	16	700	350	13.4	=(B6*D6)/3.28	=(C6-B6)/F6	145	0.895	60	=44000*(K6/(J6*G6)	=33000*(1.33-J6)*(K6/(J6*G6))
7	BO	8	16	1400	700	13.4	=(B7*D7)/3.28	=(C7-B7)/F7	145	0.895	60	=44000*(K7/(J7*G7)	=33000*(1.33-J7)*(K7/(J7*G7))
8	CO	10	10	1400	1400	10.4	=(B8*D8)/3.28	=(C8-B8)/F8	180	1	60	=44000*(K8/(J8*G8)	=33000*(1.33-J8)*(K8/(J8*G8))

Figure D-2. Equations used for finding tight and slack tension in Micro-V belts [2].

	R	S	T	U	V	W	X	Y	Z	AA	AB	AC	AD	AE
1														
2		Calculating Working Shaft Diameter based on Machine Elements in Mechanical Design 4th Ed, Robert L. Mott												
3														
4	Fac	MaxTorque	X Mome	Y Moment	Moments	Max Shear X	Max Shear Y	Bearing Forces	Design	Stress	Endu Strength	Yld Strength	Shaft Diameter	Rounded
5	A	4041.7	0	9869.1	9869.1	0	1631.3	1631.3	2	1	20655	71000	2.139	2 3/16
6	B	4656.1	3087.6	9869.1	10340.8	1145.7	1631.3	1993.4	2	1	20655	71000	2.173	2 3/16
7	O	2327.2	4228.8	0	4228.8	698.97	0	699.0	2	1	20655	71000	1.615	1 5/8
8	C	2286.6	3323.4	3430	4776.0	549.3	788.5	961.0	2	1	20655	71000	1.680	1 5/8
9	D	1714.9	0	3430	3430.0	0	788.5	788.5	2	1	20655	71000	1.505	1 5/8
10														
11		Assume shaft to be ANSI 1040 cold drawn												
12		From figure 5-8: Endurance strength is 30 ksi at tensile strength of 80 ksi												
13		Yeild strength is 71ksi												
14		With reliability of 99%, apply factor of 0.81												
15		Assume size factor of 0.85												
16		Gives actual endurance strength of 20.655ksi												
17		Design Denotes the design factor reccomended for shafts												
18		Series 6300 single row deep groove , conrad type ball bearings will give life of about 10,000 hrs.												
19		(Bearings have design factor of 6 to 10)												

Figure D-3. Values used for finding working shaft diameter [3].

S	T	U	V	W	X	Y	Z	AA	AB	AC	AD
MaxTorque	X Moment	Y Moment	Moments	Max Shear X	Max Shear Y	Bearing Forces	De:	Str	Endu Strengt	Yld Strength	Shaft Diameter
=(N5*C6)/(2*P6)	0	9869.1	=SQRT((T5^2)+(U5^2))	0	1631.3	=SQRT((W5^2)+(X5^2))	2	1	20655	71000	=(((32*Z5)/(3.14159))*SQRT(((((AA5*V5)/AB5)^2)+((3/4)*((S5/AC5)^2)))))^(1/3)
=(N6*B6)/(2*P6)	3087.6	9869.1	=SQRT((T6^2)+(U6^2))	1145.7	1631.3	=SQRT((W6^2)+(X6^2))	2	1	20655	71000	=(((32*Z6)/(3.14159))*SQRT(((((AA6*V6)/AB6)^2)+((3/4)*((S6/AC6)^2)))))^(1/3)
=(N8*B8)/(2*P8)	4228.8	0	=SQRT((T7^2)+(U7^2))	698.97	0	=SQRT((W7^2)+(X7^2))	2	1	20655	71000	=(((32*Z7)/(3.14159))*SQRT(((((AA7*V7)/AB7)^2)+((3/4)*((S7/AC7)^2)))))^(1/3)
=(N5*C5)/(2*P5)	3323.4	3430	=SQRT((T8^2)+(U8^2))	549.3	788.5	=SQRT((W8^2)+(X8^2))	2	1	20655	71000	=(((32*Z8)/(3.14159))*SQRT(((((AA8*V8)/AB8)^2)+((3/4)*((S8/AC8)^2)))))^(1/3)
=(N5*B5)/(2*P5)	0	3430	=SQRT((T9^2)+(U9^2))	0	788.5	=SQRT((W9^2)+(X9^2))	2	1	20655	71000	=(((32*Z9)/(3.14159))*SQRT(((((AA9*V9)/AB9)^2)+((3/4)*((S9/AC9)^2)))))^(1/3)

Figure D-4. Equations used for finding working shaft diameter [4].

	AE	AF	AG	AH	AI	AJ	AK	AL	AM
1			Calculating End Shaft Diamter (Taking into account fillet stress concentrations)						
2									
3			Shaft Input						
4	Rounded		Minimum RPM	HP	Max Torque		Rounded		
5	2 3/16	A	350	60	10800.0	1.389661	1 3/8 with 6 tooth spline		
6	2 3/16	B	700	60	5400.0	1.102975	1 3/8 with 21 tooth spline		
7	1 5/8	O	1500	65	2730.0	0.878662	1 keyed		
8	1 5/8	C	1400	60	2700.0	0.875432	1 keyed		
9	1 5/8	D	1900	60	1989.5	0.790703	1 keyed		

Figure D-5. Values used for finding end shaft diameter [5].

	Shaft Input			
	Minimum RPM	HP	Max Torque	Shaft Diameter
A	350	60	=63000*(AH5/AG5)	=(((32*Z5)/(3.14159))*SQRT((((3/4)*((AI5/AC5)^2)))))^(1/3)
B	700	60	=63000*(AH6/AG6)	=(((32*Z6)/(3.14159))*SQRT((((3/4)*((AI6/AC6)^2)))))^(1/3)
O	1500	65	=63000*(AH7/AG7)	=(((32*Z7)/(3.14159))*SQRT((((3/4)*((AI7/AC7)^2)))))^(1/3)
C	1400	60	=63000*(AH8/AG8)	=(((32*Z8)/(3.14159))*SQRT((((3/4)*((AI8/AC8)^2)))))^(1/3)
D	1900	60	=63000*(AH9/AG9)	=(((32*Z9)/(3.14159))*SQRT((((3/4)*((AI9/AC9)^2)))))^(1/3)

Figure D-6. Equations used for finding working shaft diameter [6].

References

[1] K. Morgenstern. (2011, November 28). TensionValues.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[2] K. Morgenstern. (2011, November 28). TensionEquations.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[3] K. Morgenstern. (2011, November 28). WorkingShaftValues.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[4] K. Morgenstern. (2011, November 28). WorkingShaftEquations.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[5] K. Morgenstern. (2011, November 28). EndShaftValues.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

[6] K. Morgenstern. (2011, November 28). EndShaftEquations.jpg. Winnipeg, Department of Mechanical and Manufacturing Engineering, University of Manitoba.

Appendix E: Concept Development

List of Tables

Introduction

The concept analysis performed for this project focuses on a selection process involving screening and scoring matrices. For a more accurate analysis, the overall system was divided into three subsystems: energy dissipation, hydraulic fluid storage, and mechanical shaft connection. The evaluations of the matrices are based on specific engineering parameters and weighted scoring. The thirty-nine engineering parameters from TRIZ were considered, and the eleven most relevant to MacDon's needs are as follows:

- Manufacturing cost
- Maintenance cost
- Volume/Shape
- Weight
- Stability
- Efficiency of time
- Durability
- Harmful effects
- Convenience of use
- Adaptability
- Complexity

Manufacturing and maintenance cost are considered in most engineering projects because cost is generally an overriding factor. Volume/Shape and weight are important because of the specific requirements laid out by MacDon, namely because the system needs to be mobile, as it must be moved with a forklift. The stability of the system is also related to the mobility, which considers the layout of the system and how the weight is distributed. Efficiency of time and convenience of use are specified towards the requirements of the input assembly and how easily it can switch between hydraulic and mechanical input. Adaptability is geared towards the subsystems following the input assembly because they are scored based on how they react to the input of the system. Complexity, durability, and harmful effects are overriding factors that affect the design. Low complexity is ideal, as it would lead to ease in operation and in some cases lower cost. Durability and harmful effects are factors considered over the life of the product and the effects on the unit itself as well as its environment. The weights applied to the matrices are based on the customer needs and these weights score the concept's relevance to the design requirements. The concept matrices will be presented the in the following sections regarding each of the subsystems.

Energy Dissipation

The first section discussed in concept selection is the energy dissipation subsystem. This subsystem is composed of two parts: heat generation and heat dissipation. The heat will be generated by the restriction of oil flow. In the heat generation section various methods of restriction will be considered. This heat must then be dissipated to the atmosphere by a radiator of which several common types will be evaluated.

Heat Generation

Multiple methods of heat generation were analyzed using a scoring matrix. The devices considered are a needle valve, a mechanical variable relief valve, an electric variable relief valves, a water break, a generator, and large fan. The eleven relevant engineering parameters were applied to the various heat generation devices. A scoring matrix was formed to evaluate the criteria numerically. Weight was applied to the scoring matrix based on the client's needs. The scoring can be seen in Table E-I below.

TABLE E-I
HEAT GENERATION SCORING MATRIX

Parameters	Weight	Needle	Variable relief mechanical	Variable relief electric	Water break	Generator	Large fan
Manufacturing Cost	**2**	+1	+1	+1	-1	-1	0
Maintenance cost	**1**	+1	+1	+1	-1	-1	+1
Volume/Shape	**2**	+1	+1	+1	0	-1	-1
Weight	**2**	+1	+1	+1	-1	-1	0
Stability	**-**	-	-	-	-	-	-
Efficiency of Time	**1**	+1	+1	+1	-1	-1	0
Durability	**1**	+1	+1	+1	0	0	0
Harmful Effects	**-**	-	-	-	-	-	-
Convenience of Use	**2**	+1	0	+1	-1	-1	0
Adaptability	**3**	-1	0	+1	0	0	0
Complexity	**1**	+1	+1	0	0	-1	+1
TOTAL		**+9**	**+10**	**+14**	**-8**	**-11**	**0**

After analyzing the heat generation methods with a scoring matrix, the electric variable relief valve was selected as the best option. This is primarily due to convenience of use and adaptability of the electric valve system.

Heat Dissipation

The justification for heat dissipation in the system can be shown by the following relationships, where E is total energy, ρ is density, V is volumetric flow rate, C_p is specific heat, and ΔT is temperature difference between the entering fluid and the ambient air.

$$E = \rho V C_p \Delta T$$

This total energy combined with time, Δt, gives the power, P, shown in the following equation.

$$P = \frac{E}{\Delta t}$$

Using the radiator selection rule of thumb the maximum potential power in the system is taken to be 30 hp [1]. By applying the given system power, an estimated temperature difference of 125°F, volumetric flow rate of 40 gpm, and standard 15W-40 oil properties, the time for the oil to reach the temperature limit is calculated to be about 10 seconds. Since MacDon plans on running the load simulator for weeks at a time, a radiator is necessary to avoid rapid overheating.

Three types of radiators were analyzed for the heat dissipation in the system. The three types considered are copper liquid-to-air, aluminum liquid-to-air, and copper liquid-to-liquid. The aluminum liquid-to-air product was deemed the most suitable option because of complexity and cost. The oil-to-water radiator requires the implementation of an external water source adding unwanted complexity and cost to the overall system. The aluminum option was chosen over the copper liquid-to-air type because of its higher cooling capacity. Based on the manufacturer performance curves, an aluminum radiator is necessary to achieve the required heat dissipation based on the flow rate of the working fluid.

Hydraulic Fluid Storage

The hydraulic fluid storage subsystem was analyzed for material selection. The materials considered were mild steel, stainless steel and aluminum. The three materials were compared based on the eleven relevant engineering parameters with the addition of corrosion resistance because of its importance in material selection.

Aluminum was given the lowest relative score primarily because of its high initial cost and its average corrosion resistance. Mild steel has the lowest initial cost and best manufacturability; however, because of its low corrosion resistance it was not chosen. Stainless steel was selected as the ideal material because of its excellent corrosion resistance. Stainless steel has high manufacturing costs; however, because of its corrosion resistance it has very low maintenance costs.

Power Input

In regards to the power input assembly, various designs were considered. These designs mainly revolved around three parameters laid out by MacDon. These three key parameters include:

1. Switching from mechanical to hydraulic input within a minute.
2. Accommodating large range of shaft speed from less than 540 to 4000 rpm.
3. Dissipating an input power of up to 60 hp.

MacDon specified that the load simulator must both have a 21 tooth spline and 1'' keyed shaft inputs. Therefore, initial concept development was focused on how to connect these two shaft inputs. Given this focus, nine options were screened using the eleven selection parameters. The following is a list of the nine designs:

- D.1: Bolt flange
- D.2: Manual Gear Box
- D.3: Disassembling pump and switching out shafts
- D.4: Cutting off old shaft and welding on new shaft
- D.5: Double chain roller coupling
- D.6: Bolt on gear coupler
- D.7: Clutch coupling
- D.8: Overrun coupler
- D.9: Adding a second hydraulic pump

Table E-II evaluates these individual designs with respect to the eleven parameters in a scoring matrix.

TABLE E-II
MECHANICAL SHAFT CONNECTION SCREENING MATRIX

Parameters	D.1	D.2	D.3	D.4	D.5	D.6	D.7	D.8	D.9
Manufacturing Cost	Low	High	Low	Low	Low	Low	High	Low	Med
Maintenance cost	Med	High	Med	High	Med	Low	High	Low	Med
Volume/Shape	Low	High	Low	Low	Low	Low	High	Low	High
Weight	Low	High	Low	Low	Low	Low	High	Low	Med
Stability	-	Low	-	-	-	-	Low	-	-
Efficiency of Time	Med	High	Low	Low	Med	Med	Low	High	High
Durability	High	Med	Med	Low	High	High	Med	High	High
Harmful Effects	Low	Med	Med	High	Low	-	Med	-	Med
Convenience of Use	Med	Low	High	High	Low	Med	High	Low	Low
Adaptability	High	Low	High	High	High	High	High	Low	High
Complexity	Low	High	Low	Med	Low	Low	High	Low	Med

The results from the screening matrix are shown in Table E-III after applying score and weight based on the client's needs.

TABLE E-III
MECHANICAL SHAFT CONNECTION SCORING MATRIX

Parameters	Weight	D.1	D.2	D.3	D.4	D.5	D.6	D.7	D.8	D.9
Manufacturing Cost	**4**	+1	-1	+1	+1	+1	+1	-1	+1	0
Maintenance cost	**2**	0	-1	0	-1	0	+1	-1	+1	0
Volume/Shape	**1**	+1	-1	+1	+1	+1	+1	-1	+1	-1
Weight	**2**	+1	-1	+1	+1	+1	+1	-1	+1	0
Stability	**1**	-	-1	-	-	-	-	-1	-	-
Efficiency of Time	**4**	0	+1	-1	-1	0	0	-1	+1	+1
Durability	**2**	+1	0	0	-1	+1	+1	0	+1	+1
Harmful Effects	**1**	+1	0	0	-1	+1	-	0	-	0
Convenience of Use	**2**	0	-1	+1	+1	-1	0	+1	-1	-1
Adaptability	**1**	+1	-1	+1	+1	+1	+1	+1	+1	-1
Complexity	**2**	+1	-1	+1	0	+1	+1	-1	+1	0
TOTALS		**+13**	**-11**	**+8**	**+1**	**+11**	**+14**	**-13**	**+16**	**+2**

The overrun shaft coupler, D.8, was selected because it had the highest score in the scoring matrix. These couplers are simple, single piece components, which are expected to last thousands of hours in operation. The male and females ends are available in all common shaft sizes and styles and weigh approximately 1.5 kg. The cost of each overrun coupler varies depending on the exact shaft dimensions used, but they are estimated to be less than $100 each. Since, the final load simulator design incorporates four shaft connections; the nine designs from the concept development phase were disregarded. These four shaft connections meet MacDon's specifications by having a 21 tooth spline and two 1'' keyed shafts.

Given the variety of industrial hydraulic pumps available, the 126 hp model from Sauer Danfoss was chosen because it can operate at high speeds of up to 3000 rpm. Given the maximum continuous pressure rating of 3600 psi, the required 60 hp can be transmitted at a much lower speed of 1400 rpm. Essentially, this pump allows for a 1600 rpm range of input speeds, which is much closer to accommodating the desired 500 to 4000 rpm range. Given this, the design of the belt drive system was much less complicated.

Commercially, there are a number of options available for stepping down the shaft speed. In order to accomplish this, enclosed gear boxes were closely examined. Through a general search of suppliers, it was found that gearboxes that could transmit 60 hp were limited to a maximum speed of 1500 rpm. Also, gearboxes found to be capable of transmitting up to 4000 rpm were not rated for the power requirements of 60 hp.

A chain drive assembly was also considered, however given chain drive standards it was found that for speeds in excess of 2500 rpm, oil stream lubrication would be required [3]. An oil stream lubrication system would add complexity and increase maintenance costs.

While automotive or custom transmissions are able to transmit high torques at high rpms, the purpose of such transmissions is different than that required in the design of the load simulator. In most transmissions, the input speed varies slightly, while the output speed may range considerably. The load simulator requires the opposite approach; it will have a large range of input speeds and must output a relatively narrow speed range for the pump. A custom design of a variable speed transmission would have proven far too costly for the $15 000 budget.

Given the transmission options, a drive train of belts and sheaves was chosen to manage the wide range to speeds. Belt drives are relatively inexpensive with respect to manufacturing and maintenance costs.

The hydraulic pump circuit in the load simulator offers a convenient means of testing hydraulic input loads, however it was specified by MacDon that this circuit must not be hydraulically linked to the tractor circuit. Linking both circuits would present high risk to the tractor being tested if contaminants were to have entered the load simulator circuit. A motor was chosen to transmit power, since motors can be readily integrated into the mechanical input assembly. Given this ease of use, the hydraulic input assembly was also designed to be able to drive an alternate motor for further testing.

Given that it must be possible to quickly switch from the mechanical input to the hydraulic inputs, the motor, which facilitates the hydraulic input, must be permanently installed. Installation of a hydraulic motor would take longer than a minute. However, implementing a clutch system would be much faster and quicker than physically disconnecting a shaft or drive.

The components that compose the bank of hydraulic couplers attached to the hydraulic motor are not considered crucial in the load simulator design. Given the enormous variety of hydraulic tees and couplings available, a design other than illustrated in this report could be built with other hydraulic components. One major advantage of the current coupler design is that all inputs for both hydraulic and mechanical inputs face the same direction. This provides great ease of use and convenience for the operators.

References

[1] Thermal Transfer Products. (2011, Nov). “BOL-400 Radiator” [Online]. Available: http://www.thermasys.com/thermal-transfer-products-site/thermal-transfer-products-home/products/industrial-hydraulic-oil-coolers/ [Nov. 19].

[2] L.M. Robert, Machine Elements in Mechanical Design, 4th ed. NJ: University of Dayton, 2004.

Appendix F: Radiator Performances Curve

List of Figure

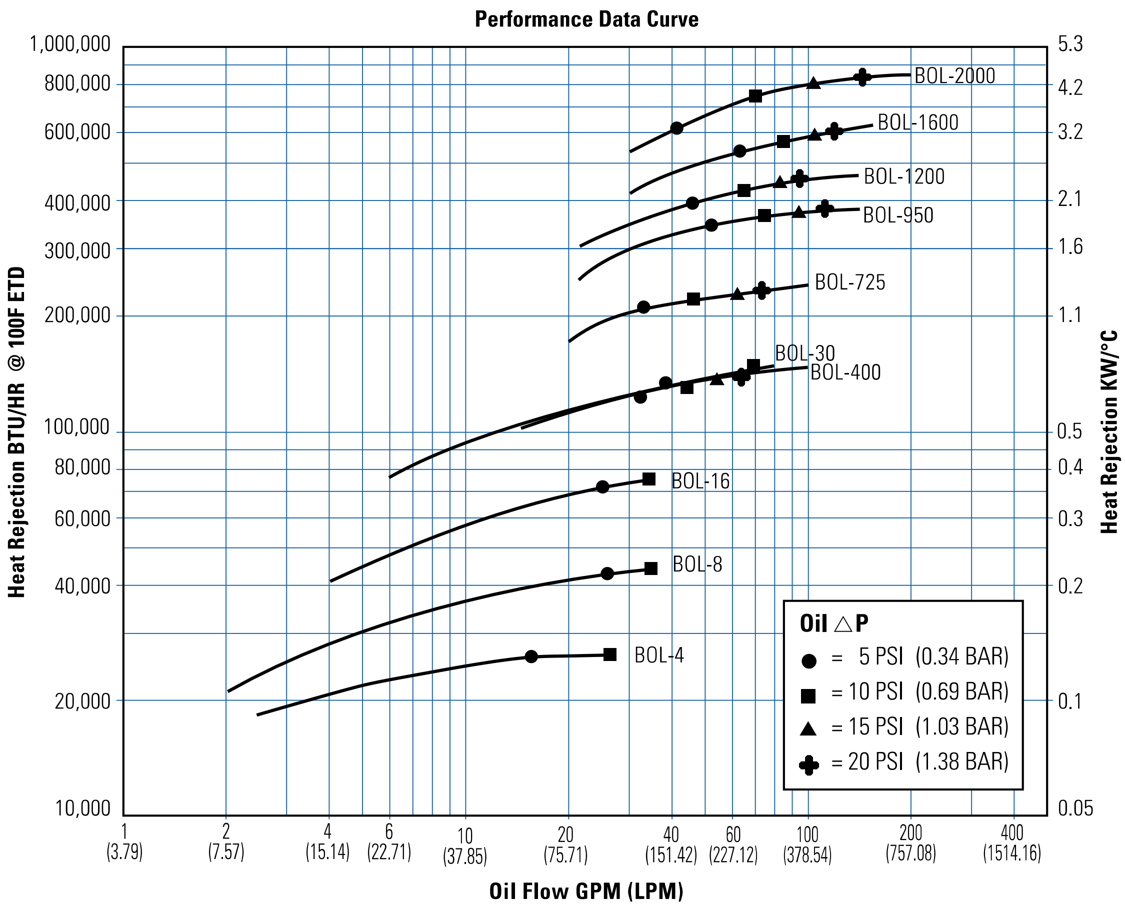

Figure F-1. Radiator performance curve [1].

Reference

[1] Thermal Transfer Products. (2011, Nov). “BOL-400 Radiator” [Online]. Available: http://www.thermasys.com/thermal-transfer-products-site/thermal-transfer-products-home/products/industrial-hydraulic-oil-coolers [Nov. 19].