Mechanical Engineering 450
Winter 2008
Custom Dry Sump Scavenge Pump for Honda CBR600F4i

Final Report
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EXECUTIVE SUMMARY

Formula SAE is an intercollegiate design series that slates students to finance, design, manufacture, test and race a small, open wheeled racecar in an annual competition sponsored by the Society of Automotive Engineers (SAE). Traditionally, MRacing houses a CBR600F4i motorcycle engine to power their car. They desire that the Center of Gravity (CG) of the car is lowered and the stock water pump is used. We have been assigned by MRacing, the Michigan Formula SAE team to design an oil scavenging system inside the CBR600F4i engine to help them achieve their set goals and improve the performance of their car. All surveyed scavenging products available in industry are oversized and do not fit inside the engine. We have spoken to Nichols Portland and were directed to Marc Goulet, Design Specialist for Automotive Oiling Systems as well as Douglas Hunter, Lead Program Engineer at BorgWarner. Both have given us their technical support in fundamentals of oil pump design. Three types of designs are considered: using internal (Gerotor or Crescent) or external (Spur Gear) gear systems.

The MRacing team requires that adequate flow rate is achieved in a small, lightweight pump assembly that will fit inside the crankcase without major modifications to the crankcase or engine block. It must be easy to manufacture, made using readily available gears and easy to remove and service. To achieve this goal we have determined engineering specifications which include a 25 liters/min/kRPM flow rate goal in a pump assembly that will fit in a 6.5x3x3 inch box inside the crankcase. The housing material will be 6061 Aluminum and will be machined using the in house machine shop. The scavenged oil will be routed through the existing 34mm hole in the side of the crankcase to the oil reservoir.

We have completed the tasks assigned. The component selection and packaging of the assembly as well as the oil routing have been addressed. We have built a final working prototype, which will be tested on the MRacing dynamometer stand. Once results are logged and analyzed, we will proceed with in-car testing until the SAE competition in May.

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1 INTRODUCTION

In 2004, MRacing implemented its first dry sump system and has experienced tremendous success in competitions ever since. The chief objective of a dry sump lubrication system is to maintain engine oil in an external reservoir, providing continuous flow to the engine lubrication supply pump in extreme driving conditions. The engine used in the MRacing racecar is the Honda CBR600F4i and the stock lubrication system utilizes a wet sump, which uses of a 2.5" deep oil pan. This system works well on the Honda CBR motorcycle, but is insufficient in the MRacing racecar for two reasons discussed below.

The foremost goal for the Oiling sub-system in 2008 is to reduce the height of the oil pan so that ultimately the engine will be mounted lower than in 2007. To achieve this goal, the thickness of the oil pan is to be less than half an inch. Lowering the overall center of gravity (CG) of the racecar will improve its performance during competition and this can only be achieved by using a dry sump oiling system. In addition, the Cooling sub-system requires that the stock water pump be used in 2008 because it is more reliable and more lightweight. To successfully implement these designs we need to design and manufacture a custom, dry-sump oil scavenge pump for the Honda CBR600F4i engine for the MRacing racecar.

1.1 Problems with Stock Oil Pan

Utilizing the stock 2.5" deep oil pan, the target vehicle CG height cannot be achieved. A higher CG height will in turn lead to decreased performance in competition, which is not desired by the team sponsor. This oil pan is used because motorcycle operating conditions vary greatly from Formula SAE race conditions. For example, during the Formula SAE competition endurance event, the engine is subject to lateral forces as high as 2g which in turn causes the engine oil to slosh and shift within the crankcase. A dry sump oiling system cannot efficiently recover engine oil in these conditions because losses in oil pressure and oil flow are caused and are detrimental to the engine. In previous years, oil that collected in the pan was gravity fed to an external oil scavenge pump using the oil pan seen in Figure 1, below. The same scavenge method is impossible this year and oil must be scavenged inside the engine. This has driven the team to design a completely flat oil pan with no outlet channels used in previous years.

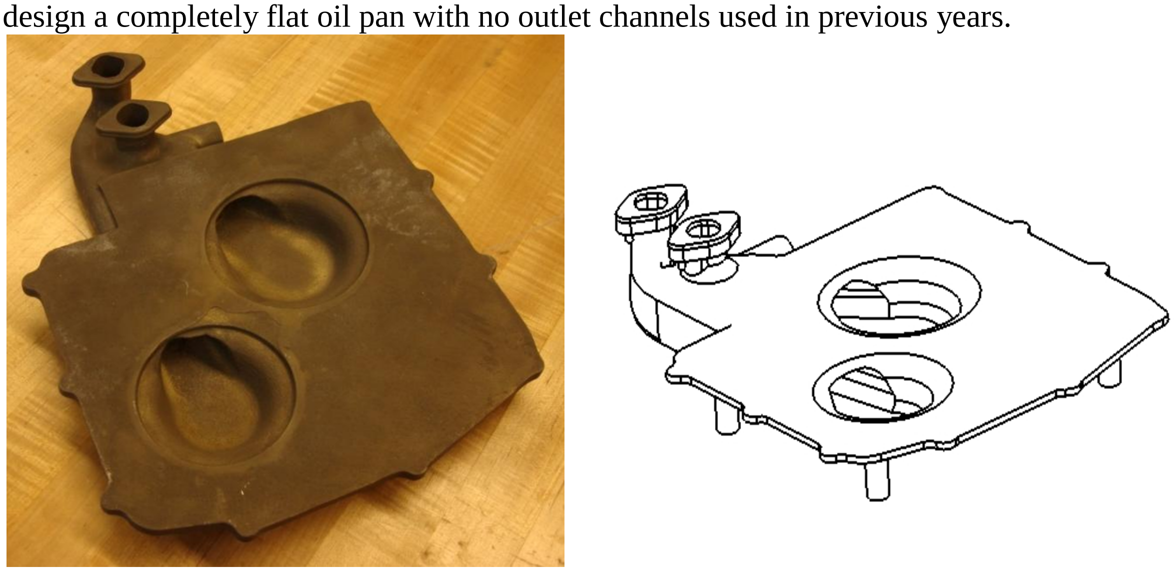


Figure 1: Picture (left) and schematic (right) of the 2007 oil pan showing oil collector pockets

1.2 Commercial Product Availability Problems

In the industrial sector, there is a very limited availability of scavenge pumps for custom applications. A general industry/manufacturer search has shown us that while there are companies that specialize in the fabrication of such oil pump systems, none offers a manufactured, off-the-shelf component that will fulfill the target set forth by MRacing in regards to the oiling system.

2 PROBLEM STATEMENT

We have been assigned by MRacing, the Michigan Formula SAE team, to solve the problems they face when implementing a dry sump lubrication system in their CBR600F4i engine system. Specifically, they require that we come up with a way to scavenge oil from inside the engine and route it to the oil reservoir. The purpose of this scavenge pump is two-fold:

a) to enable a dry-sump oiling system to be used
b) to ensure that the engine supply pump is not starved of oil

This will enable MRacing to achieve the upmost target of any racing team: to develop the fastest performing race-car. Using a completely flat oil pan, MRacing can place the engine lower than last year's car, consequently lowering the entire vehicle's CG. This will increase the overall performance of the vehicle. However, to successfully implement a flat oil pan, a scavenging mechanism must be designed, which is the task our team will undertake. In other words, our design for ME450 will begin where engine oil is collected at the bottom of the oil pan and will end where that engine oil is routed to the oil reservoir.

3 CUSTOMER REQUIREMENTS

Early on in the design process for the 2008 race-car several requirements have been established. These are analyzed below, and are discussed in the context of the entire race-car project.

3.1 Packaging

The assembly must fit an un-modified crankcase. Since MRacing re-uses engines for many years, and the oiling system components may change in the future, it is crucial the crankcase is not modified to accommodate the required scavenge pump.

3.2 Weight Reduction

One of the most important requirements in every sub-system of the 2008 MRacing race-car is to reduce overall weight. All components need to be designed in a way that minimizes their weight, while their functionality is not compromised. Following this guideline, the oiling sub-system, including the scavenge pump, must be as lightweight as possible.

3.3 Use Stock Water Pump

In previous years, the water cooling system was powered by an after-market electric water pump, and the stock water pump was not used. The goal for the 2008 Cooling sub-system is to switch back to using the stock water pump. This design choice is driven by the un-reliability of electric water pumps, the decreased complexity of the Electrical sub-system and the weight of electric water pumps. To achieve this goal, our design must therefore work in conjunction with the stock water pump.

3.4 Use Flat Oil Pan

As discussed above, to reduce the CG of the 2008 vehicle, the engine must be mounted as low as possible. To achieve this, a flat (less than 0.5 inch) oil pan must be used in conjunction with a dry sump oiling system. Our designed pump must scavenge at an average rate higher than that of the supply pump which is 10 liters/min/kRPM.

3.5 MRacing Sponsor Availability

The design, fabrication and testing process must fully utilize the team sponsor expertise, manufacturing capacity and part availability. MRacing is a student team project that is run with the help of sponsor companies, who offer financial and technical help as well as help with fabrication. As such, and since the production schedule is very time intensive, our team should use all available resources. Information sources are discussed in detail in Section 4.

4 INFORMATION SOURCES

The results of our communication with industry specialists as well as industrial availability of scavenge pumps are discussed below.

4.1 Industry Sponsors

MRacing has never designed a custom pump and does not possess much knowledge about scavenge pump design. In order to get started, we contacted several professionals in industry to aid in the development of the custom pump. We contacted Nichols Portland and were directed to Marc Goulet, Design Specialist for Automotive Oiling Systems. Nichols Portland is a division of the Parker Hannifin Corporation and has worked with McLaren F1, Ducati Motorcycles, Honda ALMS, Aircraft Engines and Harley Davidson and designed rotors for the Corvette C6.
We have also contacted Douglas Hunter, Lead Program Engineer at BorgWarner. Both have been very generous and spoke with us about fundamentals of oil pump design. Two types of designs were discussed, which are commonly used in automotive applications: using internal or external gears. Using an internal gear system entails using a gerotor (derived from Generated Rotor) pump, and using an external gear system entails a spur gear pump. Both are shown in Figure 2 below.

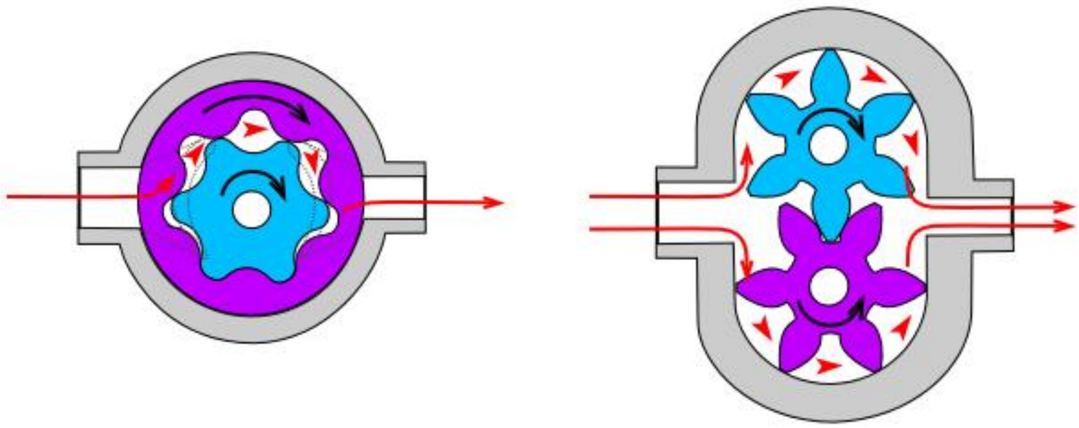


Figure 2: Gerotor Pump (left) and Spur Gear Pump (right) fluid flow schematics (Wikipedia, 2008).

4.2 Industrial Availability

It was also determined that there is no available design from industry suppliers that will fulfill both the packaging and the performance criteria that are needed for our design. Our application requires a scavenge pump for a dry-sump oiling system, and dry-sump systems are common in racing applications, but each application is unique in both flow-rate required and physical sizing, based on the corresponding engine it is used in. However, if a custom design is manufactured, then we can control the design of most pump specifications. By selecting the number of gears, type of gear system, material and assembly procedure we can adjust the scavenge pump to meet all engineering specifications. The way with which this was addressed is further discussed in Section 5.

5 ENGINEERING SPECIFICATIONS

Below are detailed descriptions of the engineering specifications that our team has decided upon. These are related to the sponsor needs, as summarized and ranked in the Quality Function Deployment (QFD) on page 7.

We have set the engineering specifications for the project, as shown in Table 1 below. To achieve this goal we have determined engineering specifications which include a 25 liters/min/kRPM flow rate goal in a pump assembly that will fit in a 6.5x3x3 inch box inside the crankcase. The housing material will be lightweight and easily machinable, and the oil will be routed through the existing 34mm hole in the side of the crankcase.

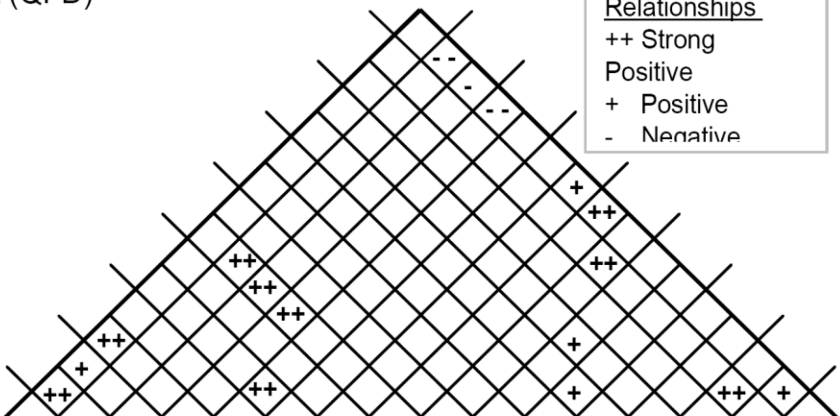
Table 1: List of Project Engineering Specifications and Justification for their selection

Specification	Target Value	Justification
Fluid Flow Rate	25 l/min/kRPM	Safety factor of 2.5 x Supply Rate, section 5.1
Gear Thickness	1 inch	At most 1/3 total pump thickness
Gear OD	2 inches	Enough to satisfy flow rate, section 5.1
# of Gears, # of Rotating Shafts	2-stage gerotors, 3 rotating shafts	Determined for alpha-design to satisfy flow rate
Gear Material	Steel	Industry Standard
Max Gear RPM	< 10 kRPM	The driving shaft will rotate at a maximum of 7 kRPM
Housing Material	7075 Aluminum	Team Availability
Housing Dimensions	6.5x3x3 inch	Maximum available space, section 5.2
Input and Output Channel Dimensions	2 Inputs, 1 Output	Determined by pump sizing, as big as possible
Machining Operations	< 10 CNC	Time and machine limitations
Number of Parts	< 10 parts	Determined by machinability
Overall Weight	< 3.1 lbs	Oiling System Goal

Custom Dry Sump Scavenge Pump for Honda CBR600F4i

Quality Function Deployment (QFD)

Relationships
++ Strong Positive
+ Positive
- Negative



Customer Requirements	Weight	Fluid Flow Rate (l/min/kRPI	Gear Thickness (in)	Gear Outer Diameter (in)	Number of Gears (#)	Gear Material	Max Gear RPM (RPM)	Housing Material	Housing Dimensions(in)	Rotating Shafts (#)	Output Channel Diameter (	Input Channel Diameter (in	Number of Input channels (	Number of Output Channel	Machining Operations Req	Number of Parts (#)	Overall Weight (lb)
Adequate flow rate	10	10	10	8	10		8			7	5	5	7	7			
Small size	10		5	7	7				10	5							5
Fits location	10				6				10	6			4	4			
Lightweight	5		6	6	7	6		6	10	5						7	10
Powered by existing shaft	7	8			8		10			8							
Routes oil through crankcase hole	10	5									8	8	7	7			
Easy to manufacture housing	8							9	5						10	8	
Complies to gear availability	9	10	6	6	6	6	8										
Easy to remove/service	8								5	3						6	
Total		296	234	234	375	84	222	102	330	285	130	130	180	180	80	147	100
Normalized		0.10	0.08	0.08	0.12	0.03	0.07	0.03	0.11	0.09	0.04	0.04	0.06	0.06	0.03	0.05	0.03
Importance Rating		**3**	**5**	**6**	**1**	**16**	**7**	**15**	**2**	**4**	**11**	**12**	**8**	**9**	**14**	**10**	**13**
Target Value		25 l/min/kRPM	0.5 in	1.5 in	open	open	10000 RPM	6061 Al	max 3" wide	open	open	open	open	open	less than 10	less than 10	max. 3.5 lb

5.1 Fluid flow rate

The engineering specifications that directly relate to fluid flow rate are discussed below. These include the gear thickness, gear outer diameter (OD), the number of gears and the maximum operating RPM of the gear sets.

The custom scavenge pump must be able to scavenge 25 liters/min/kRPM. Currently, the stock supply pump provides the engine with oil at a rate of 10 liters/min/kRPM. So the flow rate target for the scavenge pump must take into account a safety factor of 2.5 to account for the shifting of the oil in the engine when the car experiences forces more than 1g. Pump selection should be guided by fluid flow and not pressure, since oil pressure is of lesser significance at this stage.

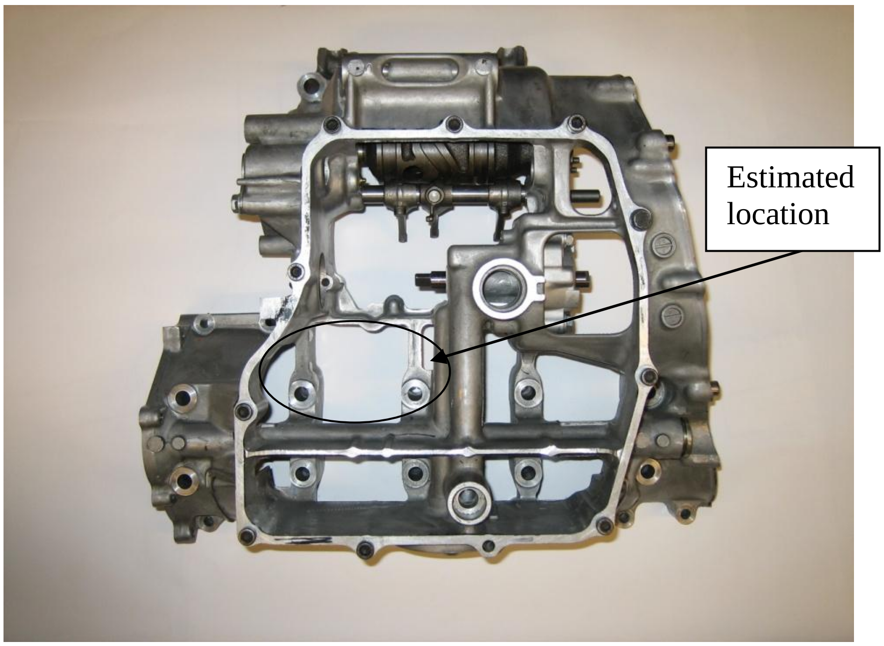


Figure 3: Estimated location of maximum oil scavenging capacity

5.1.1 Gear Thickness and Outer Diameter (OD)

The swept volume calculation (which is related to the maximum possible flow rate) is linearly correlated to both the thickness of a gear (spur or gerotor) and its outer diameter. The pump style can vary from internal gear pumps (gerotors) or external gear pumps (spur gear) or a combination of the above. In effect, the thicker a gear is selected to be, the more flow it can achieve, and similarly a larger OD provides a larger area between gear teeth for oil to flow through. However, there are limitations to this for our design, based on the packaging constraints discussed in section 5.2.

5.1.2 Number of Gears and Input Channels

We know from previous years that the engine oil collects in the oil pan in two locations. The scavenge pump should scavenge with suction from multiple points in the engine. It is estimated that 75% of the oil will come from the front of the pan directly below the main journal bearings as shown in Figure 3 above.

5.1.3 Max Operating Revolutions per Minute (RPM)

It is known that the driving shaft that will be used to drive the pump assembly rotates at half the speed of the engine. We know that the engine will rotate at a maximum of 10 kRPM, so the lower bound on maximum RPM of the gear system is 5 kRPM.

5.1.4 Input and output routing channels

The pump must deliver oil to the external reservoir without any modifications to stock engine parts; i.e. the crankcase should not be modified. Specifically, the oil should be directed through the 34mm diameter hole (concentric with pump driveshaft, shown in Figure 4 below) where the water pump sits.

5.2 Pump Housing Dimensions

The entire assembly must fit within the crankcase space available. In addition, it is required that the assembly locate within the crankcase. The tolerances in engine components approach 0.002" and since the pump will be powered via an existing shaft, it needs to be positioned and mounted accurately and precisely. The only place the pump can be mounted without modifications to the crankcase is the oil pan.

The entire assembly consisting of pump gears, power transmission system and gear housing is required to fit in a 6.5 x 3 x 3 inch box. This dimensioning constraint is shown below in Figure 4.

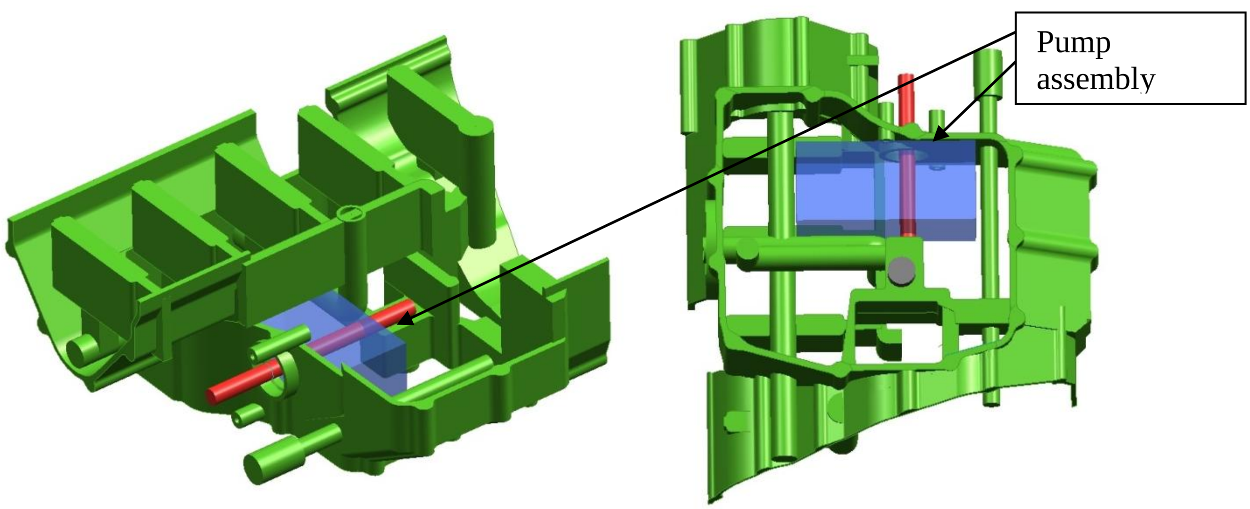


Figure 4: Isometric (left) and bottom (right) views of the required pump assembly location.

5.3 Power Transmission and Number of Rotating Shafts

The pump should ideally be powered by the existing supply pump and water pump shaft, shown in Figure 5 below. This is a 0.5" diameter shaft that rotates at half the engine speed and can be used to transfer power to the pump gears using a gear train or a chain drive system.

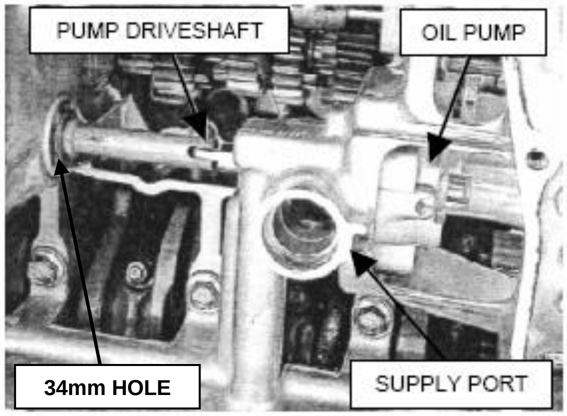


Figure 5: Points of interest in the crankcase

5.4 Overall Weight

One of the most important efforts in the MRacing team is to reduce the weight of the car. As a result, each component must be manufactured to be as light as physically possible while maintaining its function. As a result, the scavenge pump must be as lightweight as possible, an estimated goal being 3.1 lbs.

5.4.1 Gear and Housing Material

To reduce weight, we have to consider the densities and volumes of the component parts of the scavenge pump. Gears are typically made of a steel material, and comprise the densest component in the assembly. However, we can consider using a lightweight material such as aluminum to consolidate weight on the pump housing. In addition, material availability suggests the housing material be constructed out of 7075 Aluminum alloy.

5.5 Pump Housing Manufacture and Number of Parts

Since it is critical to the entire MRacing team that the pump is machined in time, and because of limitations of machine shop facilities, the manufacturing of the pump must be done using the available equipment as soon as possible. So the pump housing must be complete in as few operations as possible and the available equipment include the mills, lathes and machine tools available at the Wilson Student Project Team Center (WSTPC) and the Walter E. Lay Automotive Laboratory machine shop.

5.6 Serviceability

The desired result of the scavenge pump is an assembly that can be removable and serviceable in an easy manner. So, it is desired to remove the scavenge pump from the engine without removing the engine from the car and/or rebuilding the engine.

6 CONCEPT GENERATION PROCESS

In this section, the methods and procedures used to generate and select a working concept are described.

6.1 Level I Functional Decomposition

The problem of scavenging fluid for the aforementioned application is extremely specialized. The scavenge pump was designed specifically for the Honda CBR 600 F4i for use in the MRacing Formula SAE racecar. The principal function of our design will be to transform energy into fluid motion. A level I functional decomposition is shown below in Figure 6.

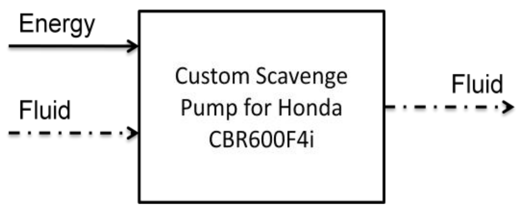


Figure 6: Level I functional decomposition illustrates the need for a device that uses energy to transport fluid.

6.2 Detailed Functional Decomposition

A pump is defined as a mechanism that creates a pressure gradient, and this is the central part of our design, however we must also provide power and fluid to the pump. A level II functional decomposition, shown below in Figure 7, illustrates the additional functional requirements. The two inputs of our system are energy and fluid. Energy has to be transmitted from the source to the pump, which pressurizes the fluid. Fluid, which flows from engine bearings into the crankcase, must be removed to replenish the dry sump reservoir. However, before fluid can be pump, we need to ensure that oil is directed toward the inlet. After fluid passes through the pump, it becomes pressurized and needs to be routed outside the engine, to the oil reservoir. The location of scavenge and expulsion points are extremely important because the distribution of oil into the crankcase is very uneven. More than 75% of the oil comes from the main crankshaft, whereas less than 10% comes from the transmission. These are the two most prominent components above the oil pan. Ideally, the pump should scavenge from every point in the engine, but since this is impractical, the scavenge points should be placed strategically at points with higher flow. The location of the pump inlet and outlet designates whether or not oil is removed from and directed to the proper locations. These sub-functions are broken down in a level III functional decomposition, shown in Figure 7 on page 12.

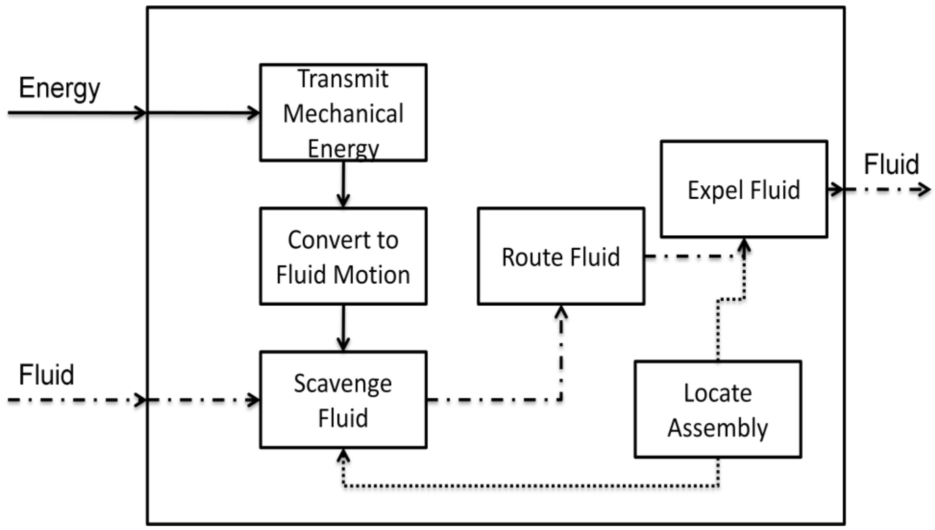


Figure 7: Level II functional decomposition illustrates the specific tasks that need to be accomplished.

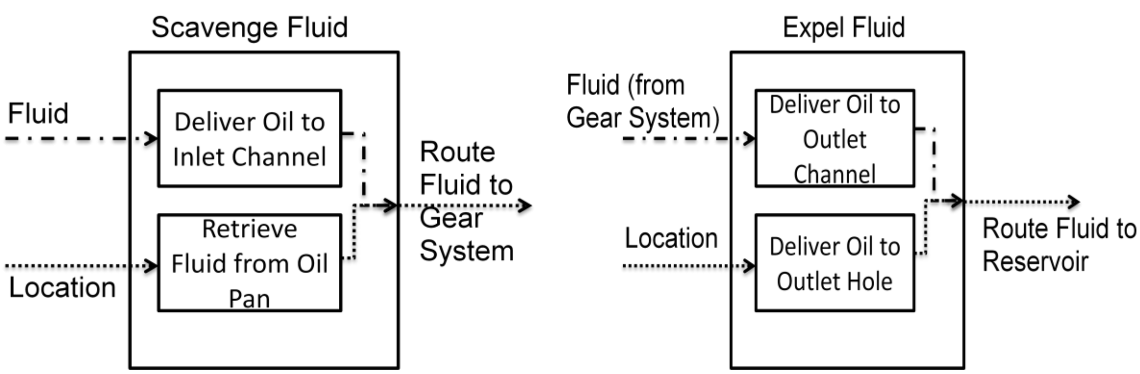


Figure 8: Level III functional decomposition demonstrates further classification of scavenge requirements.

7 GENERATED CONCEPTS

We are faced with several strict constraints with the design of the pump, such as time, availability of materials and manufacturing processes, packaging, weight, and capacity. All these factors were highly influential in our design concept. The motivation for a custom scavenge pump is the Formula SAE competition this May, which means we need to manufacture and test our design well before then. MRacing is limited to manufacturing processes available on campus, which include CNC machines in the Mechanical Engineering Graduate Shop and Wilson Student Project Center. We have very limited access to advanced manufacturing processes. In order to satisfy the 2008 oil system goal to reduce the thickness of the oil pan and height of the engine, the scavenge pump and supporting components must fit inside the engine. In order to satisfy the 2008 MRacing team goal to reduce vehicle weight, the scavenge pump must be as light as possible. Finally, the average flow should be equal to the constant 10 liters per 1000 rpm flow of the supply pump.

When we initially set the goal to develop a custom scavenge pump, there were a few decisions that were made immediately. The most prominent of these was the decision to use an existing shaft in the crankcase to power the scavenge pump. Another immediate decision was to only consider oil pump designs that are conventionally used in automotive powertrain applications. We did not evaluate any others since we are constrained with an aggressive timeline.

The morphological chart is shown below on Table 2. It summarizes the concepts we explored to satisfy each function described in the functional decomposition. A detailed review of each concept is in section 6.

Function	Options			
Transmit Mechanical Energy	Chains and Sprockets	Belts and Sheaves	Gears	Direct Drive
Convert to Fluid Motion	Gerotor	Gear Pump	Crescent Pump	
Deliver Oil to Inlet Channel	Baffling	Channels in Oil Pan		
Route Fluid	Channels in Housing	Flexible Hose	Metal Tubing	
Expel Fluid	Water Pump Adapter	Modify Crankcase	Line from Oil Pan	Direct Route to Supply Pump

Table 2: Morphological chart summarizes concepts for each function

7.1 Convert to Fluid Motion

There were three pump designs we considered to pressurize the fluid. These were a gerotor pump, a spur gear pump, and a crescent pump. All three designs are conventionally used in automotive oiling applications.

7.1.1 Gerotor Style Pump

A gerotor is a positive displacement unit consisting of two elements, an inner rotor and an outer rotor. The outer rotor has one more tooth than the inner rotor and both rotate about separate axes. As they rotate, fluid enters the enlarging chamber to a maximum volume and is then forced out as chamber volume decreases. The process occurs continuously for each chamber, providing smooth pumping action. A gerotor pump can be seen in Figure 12, below.

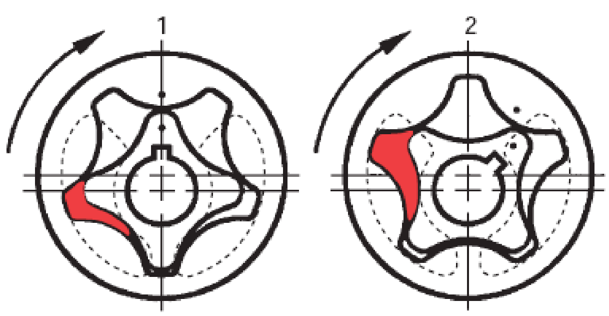


Figure 9: Gerotor style pump uses two rotors and one driveshaft to continuously pump fluid in and out of the chambers. [1]

7.1.2 Spur Gear Style Pump

Spur gear pumps consist of two or more meshed gears that rotate inside a housing. As the teeth pass the inlet port, liquid is trapped between the teeth and the housing. This liquid is carried around the housing to the outlet port. As the teeth mesh again, the liquid between the teeth is pushed into the outlet port. This action produces a positive flow of liquid into the system. A spur gear style pump is shown in Figure 13, below.

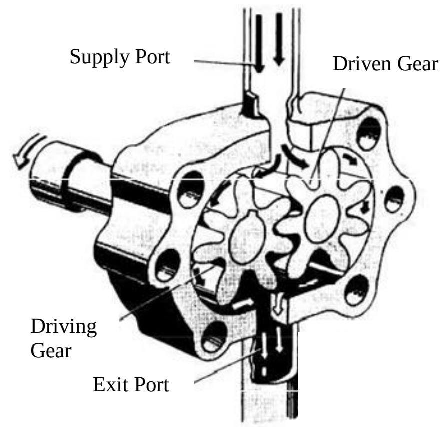


Figure 10: Spur gear style pumps use two meshed gears, one driveshaft and one driven shaft to pump fluid through housing. [2]

7.1.3 Crescent Style Pump

Crescent style pumps are internal gear pumps and rotate on eccentric axes similar to the gerotor pump. They are also similar to spur gears in the way the fluid is pushed through the pump by gear teeth. They are advantageous because the tooth designs are less complex. The fluid is sucked in axially where the teeth open up. The fluid travels along a crescent between the gear teeth and rotates around to the outlet where it is pushed out, creating the pressure gradient. A crescent style pump is shown in Figure 14, below.

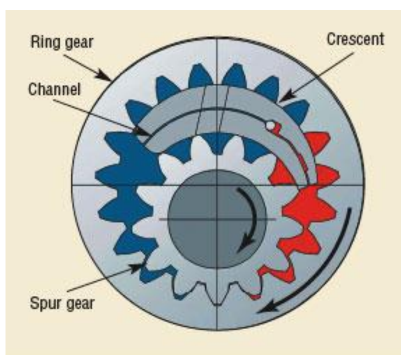


Figure 11: Crescent style pumps use internal gears and one driveshaft, similar to the gerotor. [3]

Spur gears have more packaging flexibility because spur gears can be stacked axially, like a gerotor, but also radially, which would take advantage of the wide opening in the crankcase. A gerotor requires the least amount of machining and uses only one shaft and two bearings. Spur gears involve more manufacturing and have more components. It is driven by a single shaft, but requires at least one more shaft for each radially stacked gear. Both style pumps require very close tolerances and complex porting designs.

In our search for pump components, we were referred to Nichols Portland. we spoke to Marc Goulet, their Automotive Oiling Systems design specialist, and was informed that he could provide us with any standard Nichols Portland gerotor sets free of charge. The catalog of standard sizes is shown in Appendix B. This persuaded us to seriously consider gerotors for our final design. In addition, our contact and newfound MRacing sponsor gave us a lot of insight about oil pump design.

7.2 Transmit Mechanical Energy

One issue that drove the design of a custom scavenge pump was the decision to use the stock water pump. In previous seasons, a commercial oil scavenge pump was mounted outside the engine and powered by the shaft that drives the stock water pump. Since it took the place of the water pump, an electric pump was used. For the 2008 season, the cooling system designer would like to use the stock water pump, and that is why we need to locate the scavenge pump inside the engine.

The water pump mounts to the engine such that the driving shaft mates with the main driveshaft of the stock oil supply pump. The stock oil supply pump is located in the center of the engine and the water pump mounts outside the engine. Since they are both driven on the same shaft, the shaft spans 3.1” across a section of the crankcase. This is shown in Figure 9 below. This is the section of shaft we plan to use to power the scavenge pump. The shaft is driven by a chain that connects to the primary engine transmission shaft and rotates at approximately half engine-speed. We expect that this shaft will be adequate to power the scavenge pump since it is already driving an oil pump. In addition, it is located only 2” above the oil pan, and has a lot of open space around it to fit the pump housing. This space is illustrated in Figure 3, on page 8.

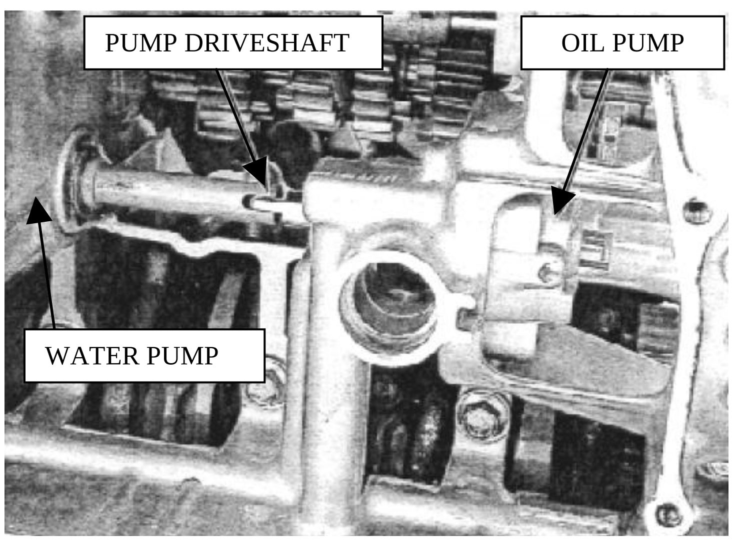


Figure 12: Stock Oil System Points of Interest

Since the shaft already powers the oil supply pump, and has a small enough diameter to directly power a pump, it would be feasible to use a direct drive. We explored a direct drive in our initial concept, which is shown below in Figure 10. We observed several problems with this design.

Since the length, of useful shaft space is only 3.1 inches, only a single gerotor could be mounted to the shaft. Use of only one stage is a risky design since there is only one scavenge point and no safety factor. In addition, the gerotor sits high off the oil pan and oil must travel 1.5" against gravity before it reaches the pump. We decided a more reliable approach would be to have multiple scavenge stages and place multiple gerotors closer to the oil pan.

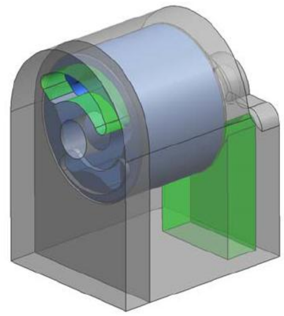


Figure 13: Concept utilizes a direct drive method exhibits a single stage and long inlet channel.

The trade-off for using multiple gerotors is the need for multiple driveshafts and each driveshaft needs to rotate. We explored three methods for transmitting power from the main driveshaft to smaller gerotor driveshafts. These methods include chains and sprockets, belts and sheaves, and gears. We immediately eliminated belts and sheaves since the system is immersed in oil. We took an in depth look at chains versus gears to determine the optimal transmission method. An analysis of the geometry and configuration of the gerotors in the crankcase were crucial in solving this problem. We utilized Autodesk AutoCAD 2D drafting software to examine the distance between shafts and what that meant for use of gears and chains and sprockets. Since the gerotors would have to be separated by at least the radius of the outer gerotor, driving gears would have to be larger than the gerotors. Such large gears would be heavy and make packaging more difficult. Due to the large difference between shaft distances, we cannot power the shafts with equal sized gears and would be forced to step the speed of the gerotors, which was undesirable. In our research of chains, we found that a tensioning mechanism is not needed if the distance between sprockets is an even number of chain pitches. We were able to accommodate such a geometry with #25 chain, which has a ¼" pitch. This is the smallest standard chain that is commercially available. A sketch of the final driving configuration is shown below in Figure 11.

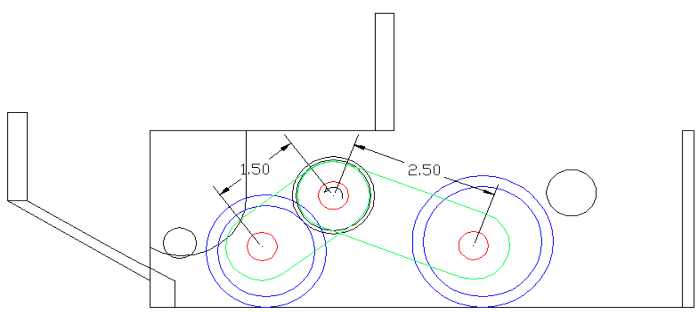


Figure 14: Sketch of shaft and gerotor configuration was used to determine the optimal power transmission method.

7.3 Oil Delivery

The pump is the principle component in the system, but will only provide the required capacity if oil is properly delivered and expelled. All the oil that goes through the engine falls into the oil pan and needs to be scavenged. Ideally, we would like to scavenge from every point in the pan. Formula 1 Racing engines have eight or more scavenge pumps, one located beneath each cylinder in the engine. This would be very impractical as it requires complex housing and power transmission designs. Instead of moving the oil pumps to the oil, we are challenged to direct the oil to the scavenge pump inlet. We attempted to design the pump with a high capacity inlet directly below the main crankshaft because more than 75% of the oil that flows into the pan falls from the crankshaft. However, these measures will not be enough and we must design baffling or channels in the oil pan to direct the flow. In the past, MRacing has guided oil flow with channels as shown in the 2007 cast magnesium oil pan in Figure 1 on page 3. However, this will be very difficult to reproduce since the oil pan this year is half the thickness. We have more closely analyzed the benefit of baffling, which accomplishes a slightly different objective. Instead of guiding oil to the scavenge inlet, baffling will prevent sloshing in the pan though the creation of partitions. This will be effective if baffling is placed around the scavenge inlet. When the vehicle encounters low accelerations, the oil will flow into the crankcase partitions, and the oil will remain between the partitions even under high acceleration. Under high accelerations without baffling, oil would slosh away from the pump inlet and delay scavenging, which could ultimately starve the supply pump.

7.4 Oil Outlet Routing and Expulsion

The final function that is required of the scavenging system is to route the oil to a central location in the engine where it can exit and replenish the reservoir. We explored three options for routing oil within the engine which included channels in the oil pump housing, soft hose with threaded hose ends, or hard aluminum tubing. We determined that channels in the housing would be difficult to manufacture initially, but require little to no assembly, while hoses and lines would require simple initial manufacturing, but complex assembly. We decided to manufacture channels in the housing, which would offer a more consolidated final product.

In a wet sump, oil never exits the engine under normal operation. In previous years' dry sump oil systems, oil exited the engine through the bottom of the oil pan as shown in Figure 1 on page 3. Since one of the main goals for 2008 is to reduce the height of the engine, the same exit point would be impossible. Since there is no provision for expulsion in the stock system, the routing of oil outside the engine posed a significant challenge. We could simply drill a hole through the crankcase, but did not want to make any permanent modifications to the crankcase. MRacing uses crankcases year after year and any specialized modifications would make it useless in future seasons. After a thorough analysis of every existing hole in the crankcase and engine block, we concluded the 35mm hole where the water pump mounts would be the closest and most practical hole to route oil through. This hole is shown in Figure 4, on page 9.

8 ALPHA DESIGN

After a thorough analysis of each function and several solutions, we developed the α-Design, which is shown below in figures 15 and 16. The central driveshaft mates between the stock supply pump shaft and stock water pump shaft. Two sprockets are mounted to the shaft that drives a chain for each gerotor. The housing construction will be CNC machined from 7075 Aluminum and will be constructed with five pieces. The central piece will support the main driveshaft and contain pockets for the respective gerotors. Two 1/8” Aluminum sheets will cover the front face of each gerotor pocket to constrain the gerotor axially as well as support the shafts. The inlet and outlet channel will mount behind the central housing. Channels will be made to support the inflow and outflow of oil through the gerotors. The outlet will also support bearings for the gerotor shafts. The outlet and central housing will be separated with a sheet of 0.032” 4130 Steel, which has cutouts for the inlet and outlet ports. Finally, there will be an adapter, which mounts external to the crankcase between the water pump and engine wall. This will seal with the outlet of the housing so that oil can flow freely through the crankcase and back to the reservoir.

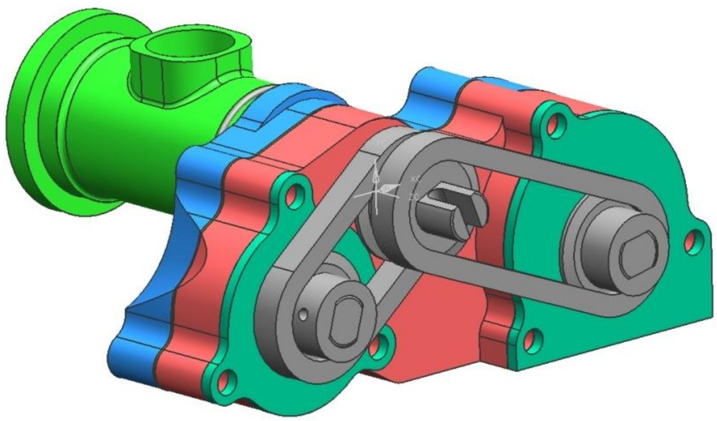


Figure 15: α-Design

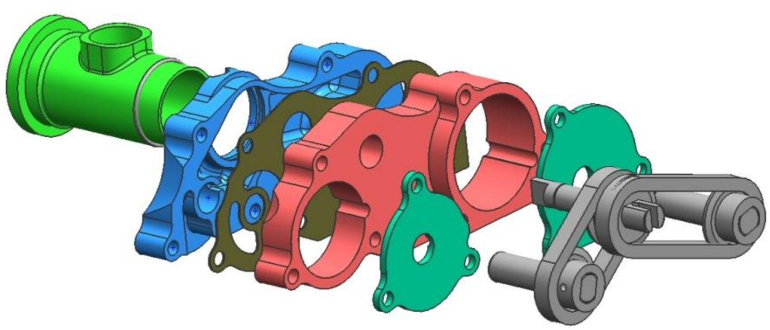


Figure 16: Exploded α-Design

9 PARAMETER ANALYSIS

The design of the scavenge pump was a very high paced process with an increasing amount of constraints as the project progressed. Constraints continued to develop as analyses were performed, parts were ordered and the design evolved. The entire design was not finalized at once, but certain aspects of the design were established, while others were still being developed.

9.1 Supply Pump Capacity Evaluation

The first objective of the pump was to fulfill a flow capacity 2.5 times greater than the flow of the supply pump. This figure was pre-determined by our Formula SAE predecessors and is explained in detail in the design goals section of this report. Preliminary tests of the supply pump flow were performed in past years, and through research of team archives, we were able to find a record of 7 liters per krpm. However, there is no record of how the measurement was recorded and we cannot verify this flow with our existing equipment. We could however, calculate the theoretical displacement based on the size of the gerotor in the supply pump. The equations for incremental displacement for gerotor pumps, spur gear pumps and crescent pumps were given to us by Douglas Hunter, Borg Warner.

In order to calculate the theoretical displacement of the existing oil pump, we used the tip diameter, TDi, root diameter, RDi, and thickness, T, of the inner in inches. The eccentricity, e, of the gerotor set can be calculated with equation 1 below. The pitch diameter, p, of the outer rotor can be calculated with equation 2, below. The pitch diameter is the inscribed circle which is tangent to the lobes.

$$e = \frac{TDi - RDi}{4} \qquad \textbf{(1)}$$

$$P = \frac{TDi - RDi}{2} \qquad \textbf{(2)}$$

The incremental displacement, D, in cubic inches per inner gear revolution per inch of rotor thickness is defined by equation 3, below. Finally, the theoretical displacement in cubic inches per revolution, Qt, is defined by equation 4.

$$D = 2\pi Pe \qquad \textbf{(3)}$$

$$Qt = D \times T \qquad \textbf{(4)}$$

These values for the Honda CBR600F4i stock oil pump are listed below in Table 3.

Parameter	Value
Tip Diameter, TDi	1.172 in
Root Diameter, RDi	0.732 in
Thickness, T	0.788 in
Eccentricity, e	0.110 in
Pitch, P	0.951 in
Incremental Displacement, D	0.654 in^3/rev/in
Theoretical Displacement, Qt	0.516 in^3/rev

Table 3: Stock Honda Oil Pump Gerotor Set Dimensions and Theoretical Displacement

The theoretical displacement converts to 8.456 liters per krpm, which proves that the 7 liter per krpm measurement from a previous season may be correct after losses are considered.

9.2 Spur Gear and Crescent Pump Incremental Displacement Formulas

The incremental displacement, D, of a crescent type pump can be calculated with the inner gear tip circle radius + eccentricity, r_2, the outer gear tip circle radius, r_1, and equation 5 below.

$$D = \pi(r_2^{\,2} - r_1^{\,2}) \tag{5}$$

The incremental displacement, D, of an external spur gear pump can be calculated with the depth of the teeth, d, the circular tooth thickness at pitch diameter, t, and the number of teeth on one gear, n, the number of gears k, and equation 6 below.

$$D = k(d \times n \times t) \tag{6}$$

9.3 Pump Selection

We used theoretical displacement equations to size gerotors since the displacement specifications for the Nichols Portland standard gerotors are all theoretical. In order to achieve the goal to design for 2.5 times the capacity of the supply pump, we determined the scavenge pump requires a theoretical displacement of 1.29 cubic inches per revolution, which converts to 21.14 liters per krpm.

After we determined the required capacity, we researched gear suppliers. All three pump designs use complex manufacturing processes, which include machining complex geometries, grinding, and heat treating. We cannot fabricate the gears in our existing facilities and must acquire these parts from a supplier. We also searched for a vendor that we could purchase stand alone gears from and not an entire oil pump assembly. We contacted Nichols Portland and were offered any of their standard gears. A list of their standard gerotor sets and specifications is listed in Appendix B.

9.4 Housing Design and Evolution

At first, we analyzed the use of a single 1.5 inch thick, Nichols 4086 gerotor. The theoretical displacement is 1.29 cubic inches per revolution and would satisfy our requirements. A CAD model of a design, which makes use of a single 1.5 inch thick Nichols 4086 and a direct drive off the driveshaft is shown in Figure 17, below. The housing consists of two pieces, which allow for installation and removal of gerotor as well as simplified manufacturing.

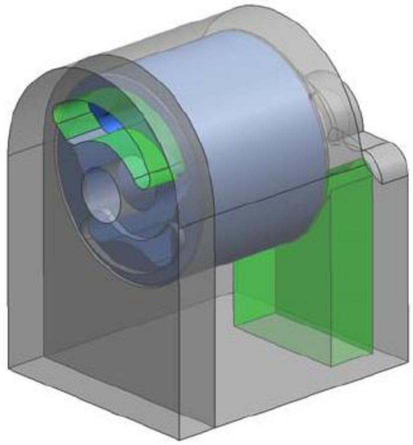


Figure 17: First iteration of scavenge pump satisfies capacity requirements, but falls short of other requirements.

After further analysis of this design, we determined that packaging a 1.5 inch thick gerotor would be very difficult. The design was flawed for several reasons. First, a single gerotor can only have a single inlet, which requires all the oil in the engine to flow to one spot and move through a single inlet. Second, the gerotor was raised too high off the oil pan, and oil needed to flow against gravity almost 2 inches before the pump inlet. Last, a single gerotor system does not offer any safety factor in the design. If the gerotor fails, the entire system fails, and can lead to catastrophic failure of engine bearings.

We quickly decided to evaluate a two stage pump, which makes use of two gerotors. We considered two configurations. The first consisted of gerotors stacked axially and the second used gerotors placed side by side. Most commercial oil pumps, including the Pace Products pump used in previous MRacing dry sump oil systems, have gerotors stacked axially. The Pace Products pump, shown below in Figure 18, has multiple inlets and outlets located at each stage. The housing geometry is complex, the separate outlets create problematic routing through the crankcase, and the total thickness is almost six inches. This design is advantageous because the gerotors are powered by a single shaft, but is unfavorable in our application with only 3 inches of usable shaft length. The Pace Products pump was oversized for our application and costs more than $500.

Figure 18: Pace Products multi-stage gerotor scavenge pump

We immediately found several advantages to placing gerotors side by side. Upon evaluating a system with side by side gerotors, other issues such as inlet placement, outlet routing, and elegant packaging seemed easily resolved. We envisioned two equal sized gerotor sets, placed on either side of the shaft, close to the pan. The inlets would be spread out and could scavenge on separate sides of the crankcase, while a single channel could join the discharge ports and route oil through the crankcase. The first iteration of our final design is shown below in Figure 19. This design was rigorously modified, but the general concept evolved into the alpha design.

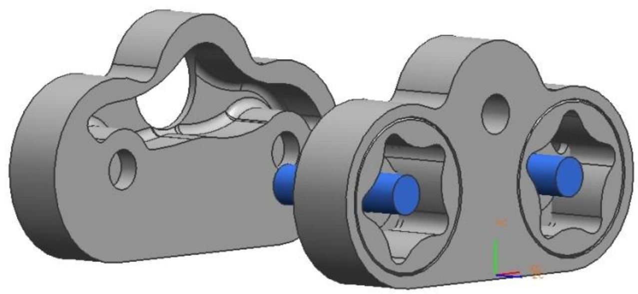


Figure 19: Exploded view of the first two-stage iteration highlights two side-by-side gerotors and an elegant outlet channel.

After further evaluation of the two-stage design we made many adjustments. The CAD model of the first iteration did not package inside the crankcase. We were forced to use a smaller gerotor on one side, which in turn required us to compensate with a larger gerotor on the other side. We discovered an additional benefit to using different sized gerotors. The smaller gerotor had to be placed under the transmission and the larger gerotor was placed below the main crankshaft. This was convenient since most of the the oil falls from the main crankshaft. We used an excel spreadsheet to assess different Nichols standard gerotor combinations and thicknesses to achieve close relative capacities of 25% from the transmission side and 75% from the crankshaft side. We settled on a design with a 0.400" Nichols 4065 gerotor and a 0.800" Nichols 4158 gerotor, shown below in Figure 20. The 4065 and 4158 would have respective capacities of 0.26 cubic inches per revolution 1.12 cubic inches per revolution. The 4065 handled 20% of the flow and the 4158 handled the other 80%.

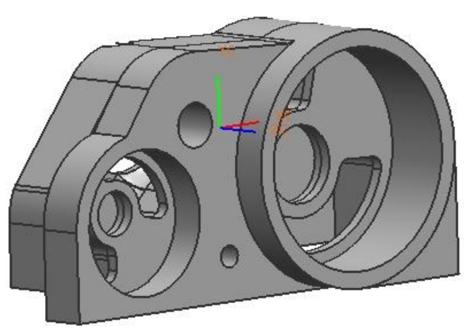


Figure 20: Design which makes use of different sized gerotors to match stream of oil in crankcase.

Unfortunately, when we requested the 4158 gerotor from Nichols, they informed us that they do not get many orders for the 4158 and did not have any in their inventory. We were forced to revise our design to accommodate more common sizes such as the Nichols 4065 and 4086. We revised the spreadsheet and settled on a .650" 4065 gerotor and 1.05" 4086 gerotor, which handled respective capacities of .42 cubic inches per revolution and .90 cubic inches per revolution. We checked to see that these sized gerotors worked in CAD and ordered them. The lead time for gerotors was a few weeks and we wanted to order them as soon as possible. Even though this aspect of the design was established, we did not have all the details with shafts, fasteners, chains and sprockets. We made preliminary estimates and developed concepts for these features, but still had to resolve many details in the housing.

9.5 Material Selection

Using the CES material selector, we wanted to determine a suitable material for the pump housing. The major constraint was the cost of the material, as we would be purchasing it ourselves. Also, because the housing acts as a large pressure vessel, we needed a relatively high strength material so that the pump can withstand the pumping pressure. Furthermore, the material must be non-flammable because it is being used in a relatively high temperature application. Along these lines, the pump must also have a low coefficient of thermal expansion so that the pump will not significantly change its size over its large operating temperature range of around 30-300° F. Finally, the pump needs to have a moderate to high hardness so that the pump does not wear significantly. Using these constraints, CES suggested using various wrought and cast aluminum alloys, as well as some high strength titanium alloys. From these, we selected 7075-T651. This aluminum alloy was selected because it is a high strength, light weight and machinable alloy that fit all of our criteria very well. Also, we already have a large supply of 7075-T651, so we did not need to buy any more material.

We also used CES to determine a material for was the gerotor covers. These covers have many of the same constraints as the pump housing, as they also act as part of a pressure vessel holding the gerotors. The covers, however, do not need to have as high a wear resistance, as there is no part rubbing against it, as is the case with the pump housing. Thus, we chose 6061-T651 aluminum for the gerotor covers, as this is a readily accessible, multipurpose aluminum alloy. Also, we have an abundant stock of 6061 aluminum sheet, so we would again not need to buy more material.

9.6 Design for Assembly

The theoretical time for assembly is 275 seconds or 4.6 minutes. This was determined with the Design for Assembly worksheet shown in Appendix F. The actual time for assembly of the scavenge pump was approximately 20 minutes because it was not assembled on a line. Since we could not eliminate any parts, the theoretical minimum number of parts in the assembly is 17. The assembly efficiency equation is defined as three times the ratio of the theoretical minimum number of parts to the actual assembly time in seconds. This equates to 4.25%.

We performed a test for the minimum number of parts and a test for part elimination and were unable to reduce the number of parts. There are several parts that cannot be combined because they move relative to each other. These include shafts, bearings, the inner and outer gears of the gerotor sets, chains and sprockets. Additionally, there are many different materials in the construction of the scavenge pump to satisfy, temperature, hardness, machinability and weight requirements. Finally, the combination of any two parts would prevent proper assembly of the scavenge pump. The gerotors need to be installed inside the housing, and channels and precise holes need to be bored within the housing. The gerotors are pressed onto shafts, which have to be inserted into the housing and sprockets have to be mounted to the shafts outside the housing.

9.7 Design for Environmental Sustainability

For the parts we have manufactured in-house we have decided to use two aluminum alloys, 7057-T651 for manufacturing the pump and outlet housing and 6061-T651 for the manufacture of the gerotor cover plates. The middle sheet was machined out of 4130 steel sheet. For everything else, we used industry standard parts made of steel. The corresponding weights and materials are summarized in Appendix F.

Although aluminum requires a lot of energy to manufacture, it is one of the most easily machinable and lightweight materials available and is widely used in automotive applications. In addition, our design should last as long or even outlast the engine. This is because the pump is not a structural member that undergoes cyclical loading so it should not fail by fatigue.
By itself our pump does not produce any air emissions because it does not convert any forms of chemical energy. It is driven off an existing shaft in the engine which powers the water pump.

9.8 Design for safety

The major risks during the fabrication of our design are standard risks associated with machine shop usage and assembly space cleanliness. The persons that are undergoing these risks are the design and assembly engineers which include the members of our team and the chief engineer of MRacing, Jason Moscetti. These are detailed below and no unexpected risk has come up from our design safe analysis.

In MRacing, we have come to accept that anything can break catastrophically at any time. Zero risk is impossible to achieve so precaution must be taken at all times. This philosophy begins at the design phase, where parts are designed with large safety factors and all designs are approved by the Chief Engineer and Team Leader, and continue until the car is retired.

9.9 Manufacturing Process Selection

The first manufacturing process selected was for the pump housing. Using CES, the pump housing was specified to be 7075-T651 aluminum. The pump housing is a solid three dimensional part, and the holes for the gerotors must hold a fairly high tolerance of 0.002”. Also, the required tooling should be as inexpensive as possible, as any tooling would be purchased by the team. Using these constraints, CES suggested the manufacturing processes in Table 4.

Manufacturing Process
Abrasive Jet Machining
Drilling
Grinding
Milling

Table 4: CES suggested manufacturing processes for pump housing.

From these processes, we selected milling. Because of the complex shape of the housing, CNC machining is needed. While abrasive jet machining is a viable option, we have no access to such a machine, so it was ruled out quickly. Furthermore, because we have access to a CNC mill, we chose milling as the manufacturing process for the pump housing.

The second manufacturing process selected using CES was for the gerotor covers. These covers constrain the gerotors axially and help to seal the pump. After using CES to select 6061-T651 aluminum as the material, we again used CES to find an appropriate manufacturing process. The main constraint for an acceptable machining process was that it must be able to produce a flat sheet part between 0.049” and 0.125” thick. This thickness range was chosen for its availability, while the actual part thickness would later be determined by FEA. Also, the expected production run would be around 10,000-25,000 units, with other Formula SAE teams and small motorsports applications as the main expected customers. With these constraints, CES suggested the manufacturing processes in Table 5.

Manufacturing Process	
Abrasive Water Jet Cutting	Hot Wire Cutting
Abrasive Jet Machining	Laser Machining
BNC Molding	Press Forming
Band Saw	Stamping
SMC Molding	

Table 5: CES suggested manufacturing processes for gerotor covers.

From these processes, we selected laser machining because it is not labor intensive and it had a very small cost to us. We have a longstanding relationship with a machining company in the area that will laser cut parts for us for free. Thus, the only cost to us was purchasing the sheet metal. Also, laser cutting is a very quick way to manufacture parts that consist only of a two dimensional profile.

10 FINAL DESIGN FEATURES

The final design is shown below in Figure 21. Dimensioned drawings of the housing and driveshafts can be found in Appendix D. There are eight major components, which make up the housing assembly, not including the gerotors and hardware.

No.	Component
1	Housing
2	Inlet/Outlet Channels
3	Separation Plate
4	Water Pump Adapter
5	4086 Cover Plate
6	4065 Cover Plate
7	Shafts
8	Chains
9	Sprockets

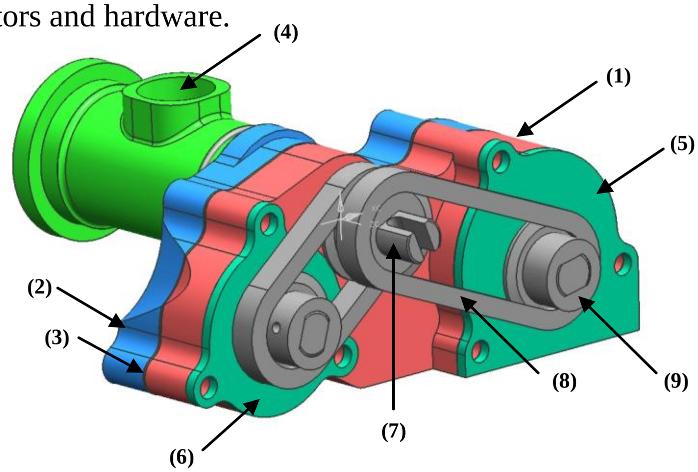


Figure 21: Housing Assembly and Components

10.1 Housing

The final design for the housing is shown below in Figure 22. The principle function of this component is to hold the gerotors and provide support for the main driveshaft. The pockets for the gerotors have very accurate tolerances. The Nichols Portland catalog specifies an outer diameter clearance of .005 to .009 inches. The axial clearance should be .002 to .004 inches. Each pocket must also have an oil supply groove to lubricate the gerotor. The groove is directly connected to the inlet and outlet channels to provide a constant supply of oil to the gerotor pocket. The housing is assembled with the other components with 6 ¼-28 NAS bolts. There are three clearance holes around each gerotor to maintain an effective seal between housing components. The housing mounts and locates to the oil pan with ¼-28 NAS bolts and 5/16" hollow dowels. The dowels ensure accurate location and repeatable installation of the scavenge pump. The location of the pump must be extremely accurate since the shaft connects to an existing shaft. The housing must also support the central driveshaft and achieves this with a 7/16" I.D. brass bushing. The housing is constructed from 7075-T651 Aluminum and CNC machined. Almost every corner on the outside geometry is filleted to eliminate stress risers and assigned radii to accommodate machining tools.

No.	Feature
1	4065 gerotor pocket
2	Threaded mounting hole, ¼-28 NAS Bolt and 5/16" dowel
3	Gerotor lubrication channel
4	4086 gerotor pocket
5	Clearance hole ¼-28 NAS Bolt
6	Driveshaft support bushing hole

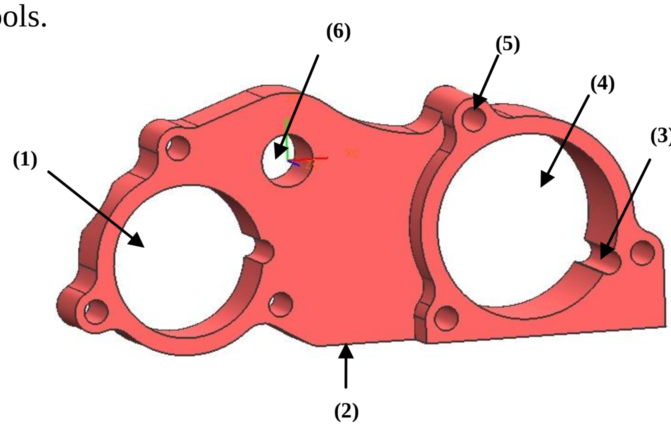


Figure 22: Housing features

10.2 Inlets and Outlets

The final design for the inlet and outlet component is shown below in Figure 23. The principle function of this piece is to provide a path for oil to enter and exit the gerotors. The outlet channel also serves to route oil from the outlets to the water pump adapter, through the crankcase wall. The depth of the inlet and outlet channels was the foremost constraining factor in this design. The depth of the port should be such that it does not restrict the pump flow, especially on the inlet side. A general rule of thumb is a port depth equal to the thickness of the gerotor. The final depth of the inlet and outlet channels is 0.55 inches. This should be adequate for the 4065 gerotor outlet, but we expect losses in the outlet port of the 4086 gerotor. The 4086 gerotor inlet is open straight back to allow free flow. Since the 4065 does not have as large a capacity, the inlet channel is more defined. The cross-sectional area of the inlet channels were designed to prevent cavitation and keep the inlet velocity below 6 feet per second. Cavitation occurs when the local inlet pressure is below the vapor pressure of the oil. When cavitation occurs, air bubbles form at the inlet and carry over to the pressurized outlet, where they can implode and cause vibrations, damage to the pump, and losses in outflow. Since this is a scavenge pump we don't expect to see any cavitation because the inlet fluid will be at least 30% air. However, we have taken this precautionary measure in case the inlet channel is drowned in oil during extreme conditions.

No.	Feature
1	4065 gerotor outlet channel
2	Pocket to clear protrusion in crankcase
3	4065 gerotor inlet channel
4	Journal bearing for 4065 shaft
5	Pocket for weight reduction
6	4086 gerotor inlet channel
7	Threaded through hole ¼-28 NAS Bolt
8	Pocket for needle bearing; 4086 shaft
9	4086 gerotor outlet channel
10	Outlet channel to water pump adapter
11	O-ring groove

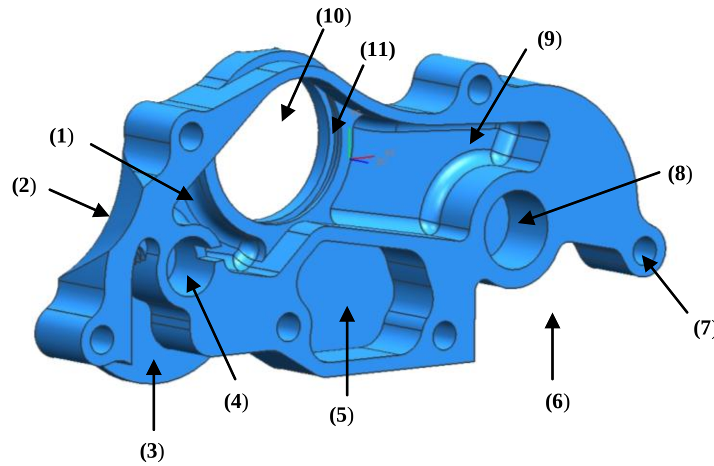


Figure 23: Inlet outlet channel features

The optimal geometry of the ports for each gerotor is specified by Nichols Portland. Since there is no barrier between inlet channels and the gerotor pocket, we used a 0.032" steel separation plate, which mounts between the housing and inlet/outlet channel to define the inlet and outlet ports. This component is shown in Figure 24, below. The oil must enter through these specific port geometries after is flows through the inlet channel and before it discharges into the outlet channel. The cross-sectional areas of the 4065 and 4086 ports are 0.366 in^2 and 0.320 in^2 respectively. The exact dimensions for each port are shown in Appendix C.

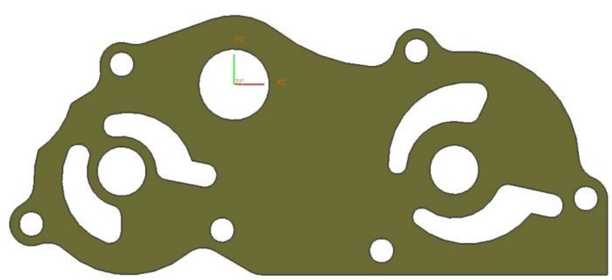


Figure 24: Separation plate has Nichols Portland defined porting geometries

The inlet outlet channel mates with the water pump adapter and has a 35mm opening in the back with an O-ring groove to seal. Oil that discharges from the gerotors meet at this point and exit through the water pump adapter. The inlet outlet channel also must support the individual gerotor shafts. We wanted to make pockets to use ½" I.D. needle bearings for both shafts, but were unable to package a bearing in the 4065 side. We used a needle bearing for the 4086 shaft, but substituted a journal bearing on the 4065, which receives oil from the outlet channel through a small groove. The inlet outlet channel has threaded holes with helical inserts to accept ¼-28 NAS bolts. Helical inserts were used to keep the bolts from vibrating loose. The pump will be hidden inside the engine and we will not be able to perform periodic bolt checks. There are two material cutouts in the inlet outlet channel. The pocket in the center is strictly for weight reduction and the one in the curved surface on the back is to clear a protrusion in the crankcase. The inlet outlet channel is constructed from 7075-T651 Aluminum and CNC machined.

10.3 Water Pump Spacer

The final design for the water pump spacer is shown below in Figure 25. The principle function of this piece is to provide a path for oil exit the engine. The spacer offsets the water pump 1 inch off the stock mounting position on the engine. We modified the water pump to mate with the water pump spacer and the spacer is clamped between the water pump and the crankcase. The groove on the water pump side sits around a web on the pump to constrain the spacer circumferentially. We will use existing water pump seals to create a seal between the oil and water flow. The end of the spacer that mounts to the crankcase was modeled after the water pump, but extended to seal with the scavenge pump inside the engine. A 0.5 inch I.D. bearing will be pressed into the water pump spacer to support the water pump shaft. Finally, a threaded hole on top accepts a -10 AN fitting to attach to an oil line.

No.	Feature
1	Circumferentially constraining groove
2	Ball bearing for shaft support
3	Oil outlet to reservoir
4	Oil inlet and pump mating surface

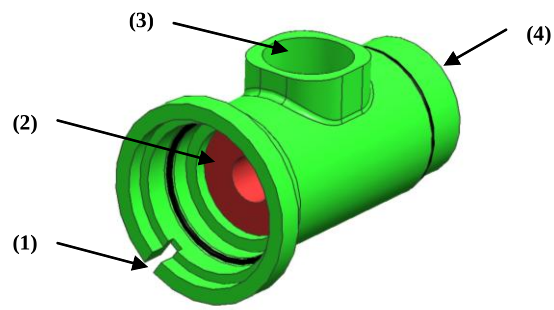


Figure 25: Water pump spacer features

10.4 Covers

The final design for the housing covers are shown below in Figure 26. The principle function of these pieces is to close the gerotors inside the housing. Since the covers mount up against the gerotors, shadow ports are included to help balance the pump axially and reduce viscous forces. The shadow ports are 0.032 inches deep and the geometry is a mirror image of the inlet and outlet ports. The covers also need to help support the gerotor shafts. Small grooves were designed to route oil to the shaft and create a journal bearing.

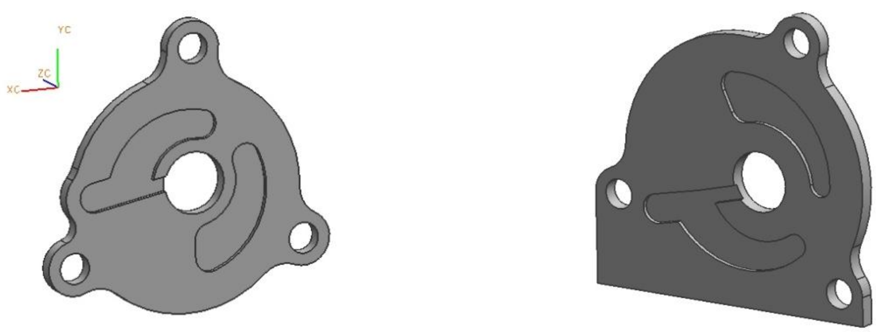


Figure 26: Housing covers

We wanted to make the covers as thin as possible, but were unsure how they would perform with up to 50 psi of fluid pressure in the gerotors. In order to verify the strength of the thin walls behind the shadow ports, we performed a simple FEA analysis. We modeled a preliminary drawing of the covers and constrained it at the bolt holes and shaft support hole, while applying pressures between 50-100 psi. We then varied wall thicknesses and recorded stress data. The results prove that a 0.125 inch sheet of aluminum would be adequate.

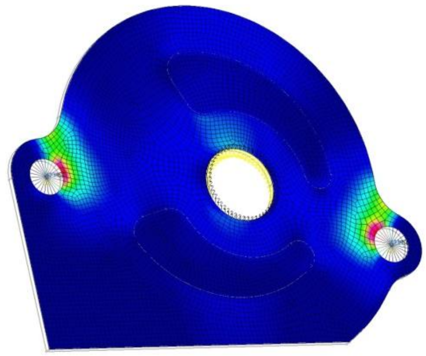


Figure 27: FEA analysis of housing covers

10.5 Shafts

The final designs for shafts are shown below in Figure 28. The gerotor shafts are pressed into the gerotors to transfer power. The gerotor driveshafts are constrained axially by the gerotors themselves and only constrained radially with bearings. The main driveshaft does not have a gerotor and needs to be constrained axially. This is achieved with a snap ring and a step down in diameter at the bushing. All three shafts have D-grooves to mount sprockets. The main driveshaft has notches on either end to mate with the stock supply pump shaft and the water pump shaft. The stock supply pump shaft will transmit power to the main driveshaft, which will transmit power to the water pump shaft. All three shafts are constructed from 8620 High Alloy Steel, and have been manufactured for us by Nichols Portland. Detailed drawings of the shafts are shown in Appendix D.

No.	Feature
1	Male projection to drive shaft
2	Diameter to fit in bushing
3	Snap ring groove
4	D-groove for sprocket
5	Female cutout to drive shaft
6	D-groove for sprocket
7	D-groove for sprocket
8	Diameter to fit needle bearing

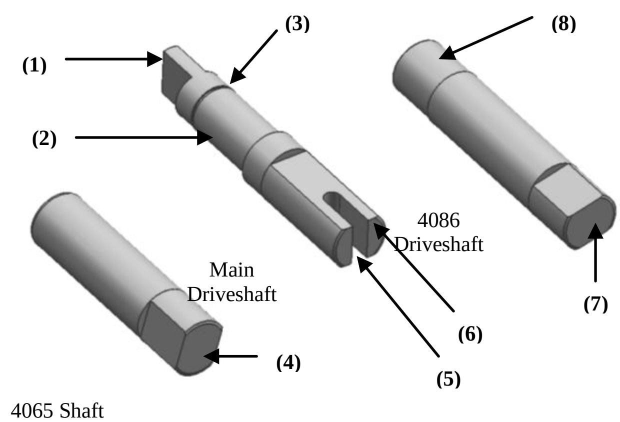


Figure 28: Driveshaft features

10.6 Chains and Sprockets

Chains and sprockets were selected with packaging as the primary objective. The smallest standard size chain is #25 and that is what we selected. #25 chain can transmit a maximum load of 100 lbs, which is well above what we expect in the scavenge pump. We are using 14 tooth sprockets to drive the shafts, which each have an OD of 1.25 inches. Each gerotor shaft has one sprocket and the main driveshaft has two sprockets so that each gerotor relies on a separate chain. This provides a safety factor in case one chain fails. We will not be able to drive the car with only one stage, but it will prevent catastrophic failure in the engine.

10.7 Prototype Description

The design detailed above will be prototype completely and exactly as described above. A parts list is shown in Appendix E. This includes parts that we will manufacture in-house and parts that sponsors will provide us as well as hardware that we have purchased.

10.8 Manufacturing Processes

To manufacture the prototype, we researched a few different options. For mass produced pumps, the housing is generally cast. This allows multiple housings to be produced inexpensively. For our prototype, casting was not as beneficial because the setup costs are too large for a single casting. Also, we were concerned about outsourcing such a larger part of the prototype. With such an abbreviated project timetable, the time required do make a cast part led us to an alternative manufacturing process. Ultimately, we decided to make the pump housing completely out of machined parts. Machining the housing allows us to have complete control of the manufacturing of the housing and thus allows us to set our own manufacturing schedule without having to rely on outside companies. The pump was designed as an assembly of six parts: gerotor housing, gerotor inlet/outlet, water pump adapter, dividing plate, and two gerotor covers. The gerotor covers and pump spacer are two dimensional sheet metal parts, so they were laser cut by one of our sponsors. So, we had to manufacture the gerotor housing, gerotor inlet/outlet, and the water pump adapter.

The gerotor housing and inlet/outlet were manufactured entirely using milling operations. Using GibbsCAM computer aided manufacturing software, g-codes were generated for the exterior profile of the parts as well as the complex inlet and outlet geometry. Using these g-codes, the two parts were machined on a CNC mill. The holes for the gerotors and the bearings need to hold a high tolerance, so they were machined using a boring head on the mill. A boring head allows you to slowly increase the size of the hole you are making, and thus gives more control of the tolerances of the process. Also, the boring head ensures that you cut a perfect circle.

The water pump adapter was manufactured using both lathe and mill operations. The inner and outer diameters were turned on a manual lathe. When turning the outer diameter, we needed to leave enough extra stock to be able to later machine the outlet line bung, pictured below in Figure 29.

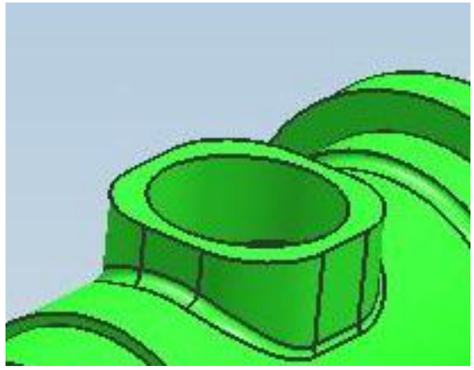


Figure 29: Outlet Line Bung on Water Pump Adapter

GibbsCAM was used again to generate g-code for the outlet line bung. This was then machined on the CNC mill. By machining the water pump adapter out of one piece instead of using a weldment, we were able to eliminate any warping that would occur during the welding process. For a thin walled part, the heat generated by the welding process can cause the part to warp if sufficient fixturing is not used. The water pump adapter must seal with the engine, so any warping of the adapter will compromise that seal and cause oil leakage. Also, a bearing for the water pump shaft must be pressed into the adapter, so warping due to welding would affect the close tolerances needed for the interference fit.

11 PROTOTYPE VALIDATION

In order to demonstrate that our design works and that all our customer requirements and engineering specifications were met, we performed a series of qualitative tests. Due to a dynamometer engine piston failure testing has been delayed severely.

11.1 Qualitative Tests

We performed several validations prior to installing the pump in the engine which include packaging verifications, proper function of the driving system, and weight. One of the customer requirements was that the housing can be manufactured with the equipment on the University of Michigan campus and the fabrication stage has been successfully completed. We assembled the entire pump and mounted it in the engine with the oil pan, water pump adapter, and stock oil pump to ensure a proper fit. The weight of the pump assembly is 2.82 pounds, which fulfills the minimum weight customer requirement of 3.1 pounds. Table 6 below shows the comparison between target and actual values for the prototype.

Specification	**Target Value**	**Actual Value**
Gear Thickness	1 inch	0.65” and 1.05” (two-stage pump) set specifications by Nichols Portland availability
Gear OD	2 inches	0.65” and 1.05” (two-stage pump) set specifications by Nichols Portland availability
# of Gears, # of Rotating Shafts	2-stage gerotors, 3 rotating shafts	2-stage gerotors, 3 rotating shafts
Gear Material	Steel	Steel, set specification of Nichols Portland
Max Gear RPM	< 10 kRPM	<10 kRPM, according to engine RPMs
Housing Material	7075 Aluminum	7075 Aluminum, set by team material availability

Housing Dimensions	6.5x3x3 inch	Maximum available space, section 5.2
Input and Output Channel Dimensions	2 Inputs, 1 Output	2 Inputs, 1 Output
Machining Operations	< 10 CNC	6 CNC operations,
Number of Parts	< 10 parts	10 (not incl. fasteners, o-rings and bushings)
Overall Weight	< 3.1 lbs	2.9 lbs (approx.)

Table 6: Comparison between customer requirements and actual values

11.2 Dynamometer Tests

In order to test our designs we will utilize MRacing's engine dynamometer in the Walter E. Lay Automotive Laboratory. Our pump and all other necessary components will be setup on the engine test bed. This is not an exact representation for what will be seen while our pump setup is used on the MRacing vehicle. The engine test bed represents a steady state environment, meaning the system will not be subject to lateral and longitudinal acceleration forces seen when the vehicle is in operation. In order to simulate this we plan to elevate different sides of the dynamometer to specified angles in order to test how these acceleration forces will affect the performance of the pump. This still is not a perfect representation to the vehicle setup as we cannot test the transient affects of actual driving.

The components for the 2008 Engine, Oiling, Cooling and Electronics systems have all been completed and the CBR600F4i engine has been assembled to test statically. The entire oiling system is made up of an oil reservoir, pan, cooler, supply and scavenge pump. It was supposed to be tested last week, but due to a piston failure, we have been delayed. This testing will determine whether or not our gerotor pumps work to deliver adequate flow to the reservoir as well as to validate the chain and sprocket driving system that we plan to use.

Regarding our specific project, we will obtain data on oil pressure coming out of the scavenge pump at all operating engine speeds. We will then calculate flow rate and compare this with the engineering specification for flow rate defined in Section 5. The flow rate out of the scavenge pump and into the reservoir must be at least 25 L/min/kRPM. In addition, we will evaluate the performance of our proposed chain and sprocket driving system. We cannot perform any experiment for this, but static testing will reveal if this component will fail. Also, decreased performance can be attributed to the driving system if upon disassembly we determine that either the chain or sprockets show signs of wear.

11.3 In Car Testing

It is common practice for the MRacing team to conduct dynamic testing in any available locations, such as the GM Proving Grounds. During dynamic testing, MRacing tries to simulate competition events such as endurance, acceleration and skid-pad. During these tests all components will be shaken down and this will enable us to see which component, if any, is prone to failure.

Dynamic testing occurs once the car is assembled completely and is used to validate all systems overall. As is detailed in Section 11.2 Challenges and Problems, we are expecting to face some difficulty during long left turns. This is because of the sloshing effect of engine oil in a flat oil pan and because of the location of our scavenge pump. However, this forms the basis of our large safety factor in flow rate (2.5 times the supply rate). We hope that during a long left turn, the engine will be supplied enough oil from the reservoir and not be starved. This is something we can only test with dynamic testing. Specifically, we will collect track data on engine pressure and lateral acceleration using the engine control unit (ECU) and MEMS accelerometers on the car, respectively. Using the combination of lateral acceleration and engine pressure data, we can look at how much oil is delivered to the engine during left turns.

12 DESIGN CRITIQUE

Overall, our design and prototype met most of our design goals. First, the final design was 0.28 lbs under our design weight goal, and thus is the smallest pump that would be acceptable for our application. Also, we managed to fit the pump inside of the engine and use an existing hole in the crankcase to route oil out of the engine. Because of this, we were able to reduce the center of gravity of the entire car about 0.25".

However, our design also has some drawbacks that would need to be addressed in future designs. First off, some small modifications were needed to fit the pump into the engine. While these were made without problems, the pump should be redesigned to prevent this in the future. If every engine that used the pump had to be machined, that would be a strong deterrent to adopting the pump. Also, in order to allow the pump to fit, the pump machining was very complex and labor intensive. A different manufacturing process like casting might allow for easier manufacturing, and would be much more cost effective if we were to produce larger quantities.

13 SUMMARY AND CONCLUSIONS

We have been assigned by MRacing, the Michigan Formula SAE team, to design an oil scavenging system inside the CBR600F4i engine to solve the problems they face when implementing a dry sump scavenge system. They desire that the CG of the car is lowered and the stock water pump is used. All aftermarket products are oversized and do not fit in the crankcase. We have spoken to industry sponsors of the team and have determined to consider either a Gerotor gear pump system or a spur gear pump system. The MRacing team requires that adequate flow rate is achieved in a small, lightweight pump assembly that will fit inside the crankcase without major modifications to the crankcase or engine block. It must be easy to manufacture, made using readily available gears and easy to remove and service.

To achieve this goal we have determined engineering specifications which include a 25 liters/min/kRPM flow rate goal in a pump assembly that will fit in a 6.5x3x3 inch box inside the crankcase. The housing material will be lightweight and easily machinable, and the oil will be routed through the existing 34mm hole in the side of the crankcase. We have completed the tasks assigned. The component selection and packaging of the assembly as well as the oil routing have been addressed. We have built a final working prototype, which will be tested on the MRacing dynamometer stand. Once results are logged and analyzed, we will proceed with in-car testing until the SAE competition in May.

14 ACKNOWLEDGMENTS

We would like to acknowledge the contributions of: Volker Sick, Professor – Mechanical Engineering, University of Michigan; Jason Moscetti, 2008 MRacing Chief Engineer, University of Michigan; Steve Rumple, Arvin Meritor; Douglas Hunter, Lead Program Engineer, BorgWarner; Marc Goulet, Design Specialist - Automotive Oiling Systems, Nichols Portland.

15 REFERENCES

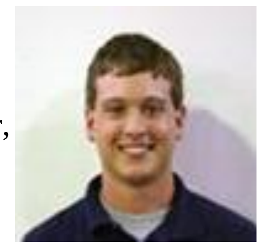
[1] “Honda CBR600F4i Service Manual” 2001-2003.
[2] Munson, Bruce R., Donald F. Young and Theodore H. Okiishi, Fundamentals of Fluid Mechanics: Fifth Edition. John Wiley & Sons, Inc., Ames, Iowa, 2006.
[3] “Jagg designed oil coolers by Setrab oil coolers”, http://www.setrabusa.com/jagg, Setrab USA, Inc. Johnstown, Ohio .
[4] Gerotor Selection and Pump Design v1.2. Nichols Portland, Portland, ME
[5] Engine Mechanics: Fluid Power, Spur Gear Pumps “Integrated Publishing: The Most Informative Site on the Internet” http://www.tpub.com/

APPENDIX A **- BIOGRAPHCAL NOTES**

Tasos Charalambides

Tasos was born in Paphos, Cyprus in 1985 to Mary and Costas Charalambides. After graduating high school in 2002 he joined the Cypriot National Guard, where he served for two years. His interest for problem solving brought him to the School of Engineering at Michigan, from where he will graduate in 2008. After graduation he will pursue graduate education and aims to one day have a Ph.D. in Mechanical Eng. The first vehicle he was licensed for was an armored personnel carrier called the Leonidas APC.

Anthony Ferrara



Anthony was born in London, England in 1986 to Al and Rita Ferrara. Soon after, he moved to Pittsburgh, where he lived for most of his childhood. As a child, Anthony was very interested in the University of Michigan football team, which was his first exposure to the University. As he got older, more and more interested in the University because of its ties to the automotive industry. At Michigan, Anthony joined the Formula SAE team, and has been responsible for the design of various components on the powertrain system. Also, he currently works at the University of Michigan 3-D and Visualization Lab in the Duderstadt Center.

Alexander L. Hutz

Alex was born in Pittsburgh, Pennsylvania in 1986 to James and Mary Hutz. Ever since birth he has been interested in how things are put together and manufactured. He graduated from Hampton High School in Allison Park, PA and chose to attend the University of Michigan's College of Engineering. At UofM he joined the Formula SAE Team. He has designed the braking system for the past two years and for 2008 chose to be the Team Captain and Project Manager. He has a passion for cars and manufacturing and has spent the past four years of his life devoting himself to these things. He has also had the opportunity to drive the International CXT, the largest pickup truck in the world.

Elliot Kruk

Elliot Kruk is a senior mechanical engineering major with a concentration in thermodynamics and fluid mechanics. Originally from Stamford, CT, Elliot came to the University of Michigan for its strong ties to the automotive industry. Elliot has been a member of the University of Michigan Formula SAE team for three years and held two executive design positions. He designed the fuel system in 2007 and is presently the oil system engineer.

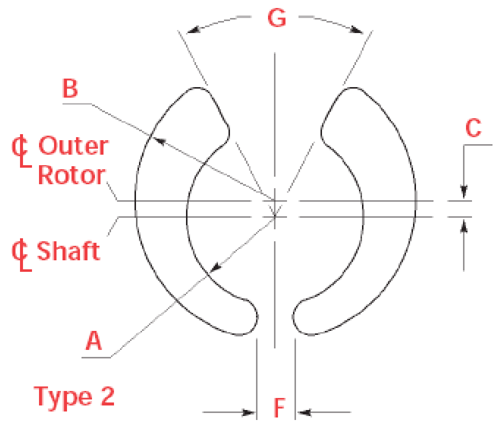
APPENDIX B - **NICHOLS PORTLAND STANDARD GEROTOR LIST**

Nichols Portland Standard Gerotors and Specifications*

(English units table)

Gerotor Type	Maximum Operating Speed**	Thickness Min.	Nominal Range Max.	Nominal O.D. Standard	Nominal O.D. Minimum Recommended***	Nominal I.D. Standard	Nominal I.D. Maximum Recommended**
	(rpm)	(inch)	(inch)	(inch)	(inch)	(inch)	(inch)
6010	21556	0.100	0.360	0.805	0.805	0.202	0.202
10010	17250	0.125	0.500	0.998	0.873	0.313	0.375
6020	16000	0.125	0.750	1.123	1.123	0.313	0.313
6022	14500	0.125	0.625	1.123	1.123	0.313	0.313
8030	11250	0.125	0.625	1.498	1.373	0.500	0.625
10060	7250	0.188	1.250	2.248	2.123	0.625	1.000
6063	9000	0.188	1.250	1.998	1.748	0.625	0.688
4065	10000	0.188	1.250	1.748	1.624	0.500	0.500
4086	7850	0.188	1.250	1.969	1.969	0.500	0.500
6095	7250	0.188	1.500	2.248	2.248	0.750	0.750
12131	4500	0.250	2.000	3.248	3.123	1.000	1.750
4158	6250	0.250	1.375	2.473	2.473	0.543	0.875
14162	3750	0.250	1.500	3.998	3.748	1.625	2.125
6170	5500	0.250	2.000	2.998	2.873	1.000	1.125
6166	5000	0.250	1.375	2.998	2.998	1.000	1.125
4180	6000	0.250	1.375	2.627	2.627	0.563	0.875
6280	4250	0.250	3.000	3.998	3.750	1.250	1.500
8369	3250	0.375	3.000	4.498	4.498	1.750	2.125
8384	3250	0.375	1.750	4.498	4.498	1.750	2.125
10397	2800	0.375	1.000	5.000	5.000	2.000	2.400

APPENDIX C **- PORTING GEOMETRY**



Standard Gerotor Porting Information (See Figure 4)

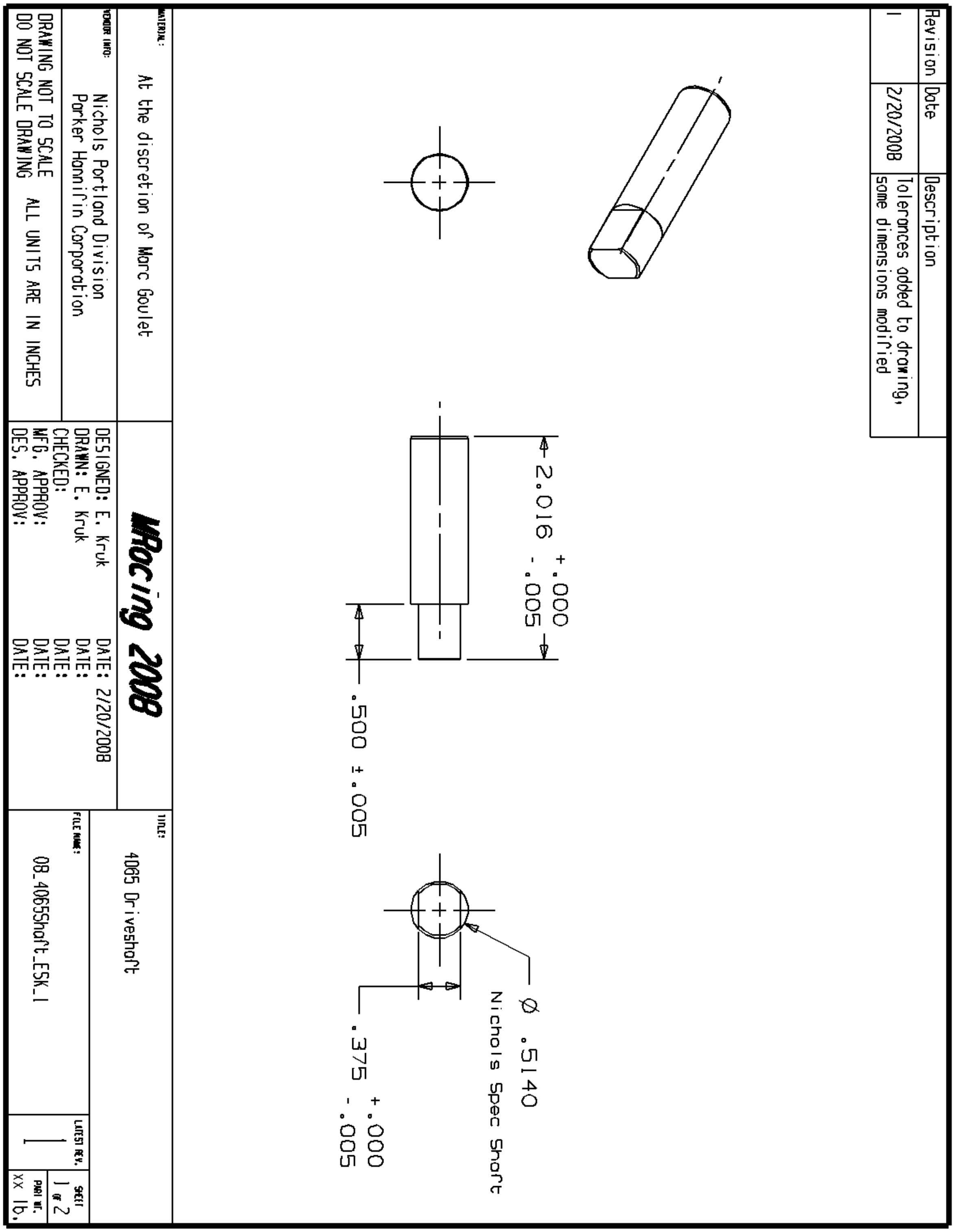
Gerotor Type	Radius A	Radius B	Ecc. C	Width D	Land E	Land F	Angle G	Port Area
	Nominal Porting Dimensions							
	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(degrees)	(mm^2)
6010	5.0	7.9	0.9	2.6	5.0	2.8	52.4	44.0
10010	6.9	9.2	0.660	2.0	4.4	2.0	34.5	49.0
6020	6.6	10.8	1.321	4.0	6.8	2.9	54.0	88.4
6022	7.3	11.5	1.321	4.0	7.5	3.4	54.0	94.8
8030	10.6	14.8	1.321	4.0	8.4	3.6	42.0	138.1
10060	17.3	22.6	1.651	5.0	11.0	5.6	34.5	277.4
6063	11.9	19.1	2.286	6.9	12.6	4.7	55.0	263.2
4065	9.3	17.9	2.794	8.4	14.0	6.5	79.5	236.1
4086	12.9	21.6	2.819	8.5	18.7	10.4	79.5	206.5
6095	14.7	23.4	2.794	8.4	15.5	6.0	55.0	391.6
12131	27.2	34.3	2.286	6.9	14.2	5.9	28.5	601.3
4158	15.1	27.9	4.191	12.6	22.2	11.9	77.5	551.6
14162	34.1	41.2	2.286	6.9	15.2	7.9	24.5	748.4
6170	19.1	30.7	3.810	11.4	19.8	8.1	54.0	692.3
6166	22.2	32.7	3.429	10.3	22.3	14.6	52	649.7
4180	14.6	29.2	4.775	14.3	22.1	9.8	79.0	638.7
6280	25.4	39.9	4.775	14.3	26.7	10.2	55.0	1125.8
8369	34.9	49.5	4.775	14.3	28.1	8.5	43.0	1572.3
8384	36.6	51.2	4.775	14.3	29.7	9.4	43.0	1638.7
10397	42.9	57.4	4.343	12.7	27.4	10.4	34.5	1740.0

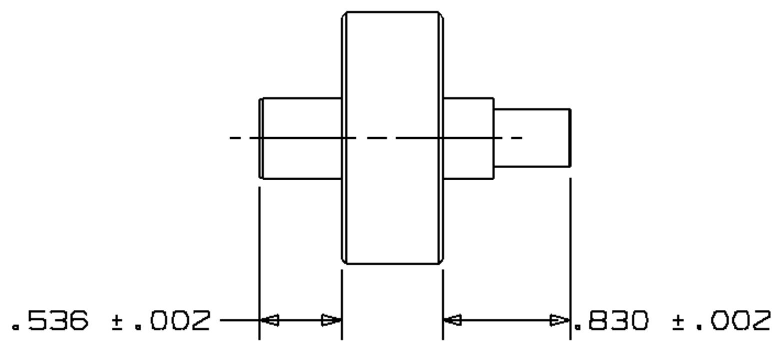
*Note: Some gerotors are better suited to some applications than others. Please contact Nichols Portland Product Engineering prior to final selection.

** Recommended maximum operating speed (see Section 6D).

*** These minimum and maximum recommended feature sizes are for reference only. This does NOT imply that tooling is available to produce these sizes, and an additional tooling charge may be required in order to produce non-standard O.D.s or I.D.s. Please contact Nichols Portland Product Engineering prior to considering a non-standard I.D. or O.D.

APPENDIX D **- DIMENSIONED CAD DRAWINGS**





MATERIAL:	MRacing 2008	TITLE: 4065 Gerotor Shaft Assembly
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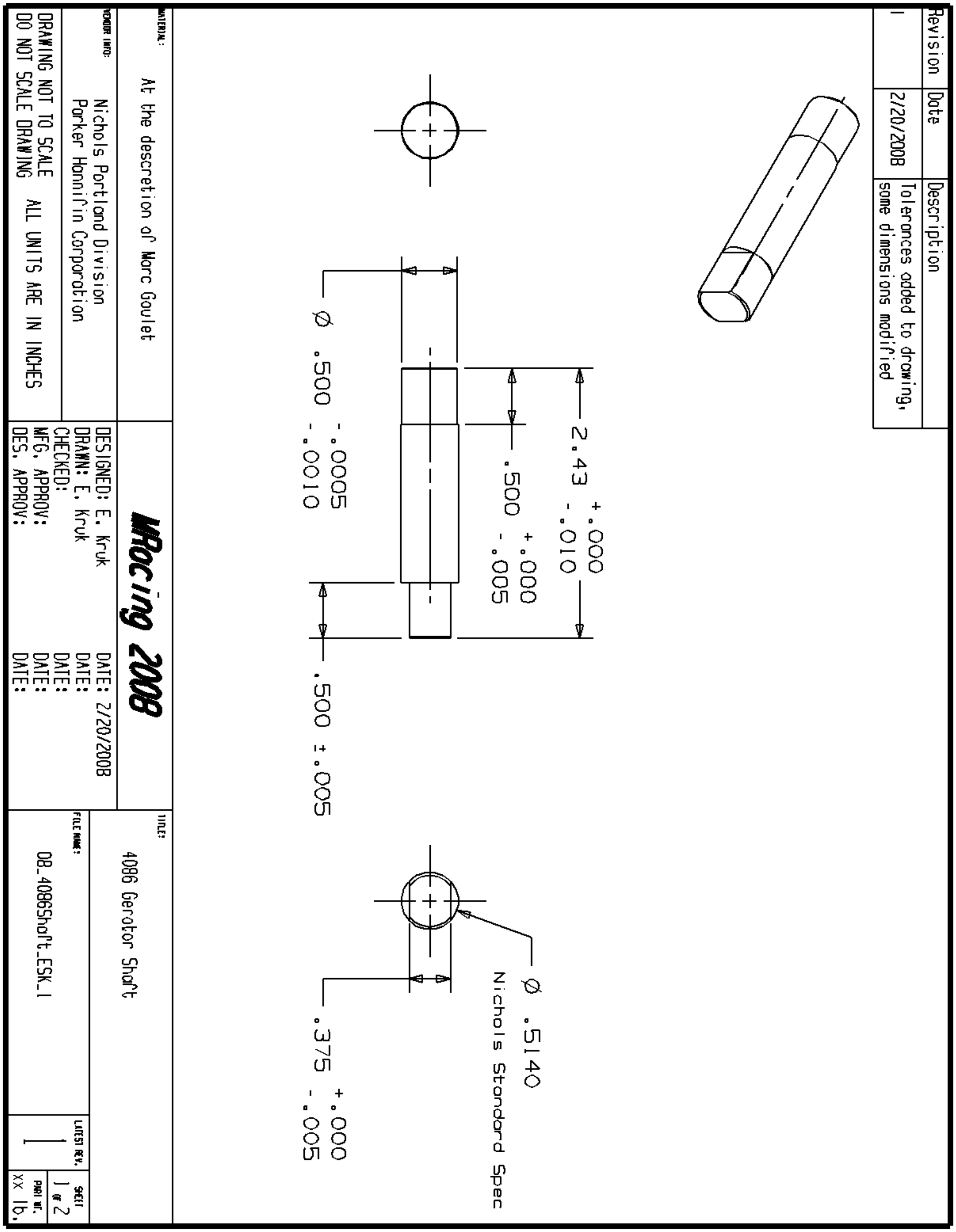
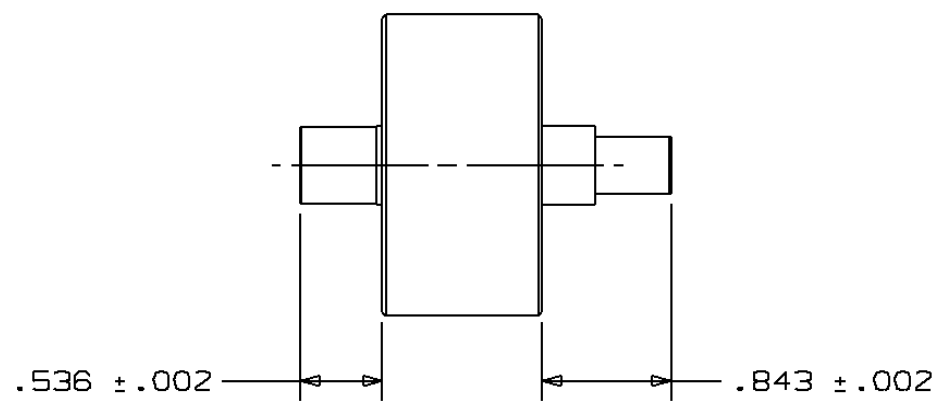
DRAWING NOT TO SCALE DO NOT SCALE DRAWING ALL UNITS ARE IN INCHES		LATEST REV. 1 / SHEET 2 of 2 / PART WT. xx lb.



Revision
Date
Description
1
2/20/2008
Tolerances added to drawing, some dimensions modified
2.43 +.000 -.010
.500 +.000 -.005
Ø .500 -.0005 -.0010
.500 ±.005
Ø .5140
Nichols Standard Spec
.375 +.000 -.005
MATERIAL:
At the descretion of Marc Goulet
VENDOR INFO:
Nichols Portland Division
Parker Hannifin Corporation
DRAWING NOT TO SCALE
DO NOT SCALE DRAWING
ALL UNITS ARE IN INCHES
WRacing 2008
DESIGNED: E. Kruk
DATE: 2/20/2008
DRAWN: E. Kruk
DATE:
CHECKED:
DATE:
MFG. APPROV:
DATE:
DES. APPROV:
DATE:
TITLE:
4086 Gerotor Shaft
FILE NAME:
08_4086Shaft_ESK_1
LATEST REV.
1
SHEET
1 of 2
PART WT.
xx lb.



MATERIAL:

MRacing 2008

TITLE:

4086 Gerotor Shaft Assembly

VENDOR INFO:
Nichold Portland Division
Parker Hannifin Corporation

DESIGNED: E. Kruk DATE: 2/20/2008
DRAWN: E. Kruk DATE:
CHECKED: DATE:
MFG. APPROV: DATE:
DES. APPROV: DATE:

FILE NAME:

LATEST REV. 1

SHEET 2 of 2

PART WT. xx lb.

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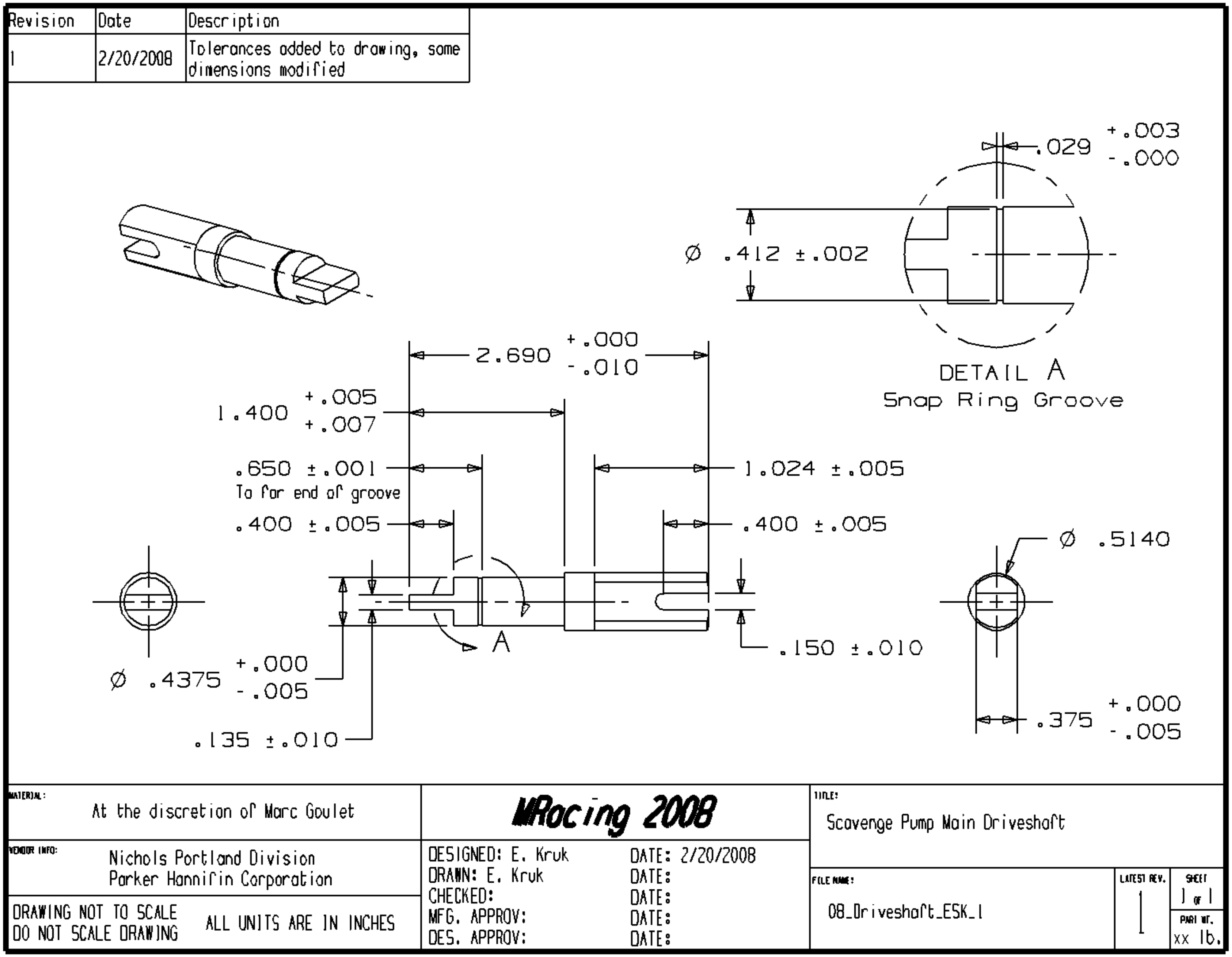
DO NOT SCALE DRAWING
ALL UNITS ARE IN INCHES


Revision | Date | Description
1 | 2/20/2008 | Tolerances added to drawing, some dimensions modified
.029 +.003 -.000
Ø .412 ±.002
DETAIL A
Snap Ring Groove
2.690 +.000 -.010
1.400 +.005 +.007
.650 ±.001
To far end of groove
.400 ±.005
1.024 ±.005
.400 ±.005
Ø .5140
A
.150 ±.010
Ø .4375 +.000 -.005
.135 ±.010
.375 +.000 -.005
MATERIAL:
At the discretion of Marc Goulet
MRacing 2008
TITLE:
Scavenge Pump Main Driveshaft
VENDOR INFO:
Nichols Portland Division
Parker Hannifin Corporation
DESIGNED: E. Kruk DATE: 2/20/2008
DRAWN: E. Kruk DATE:
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MFG. APPROV: DATE:
DES. APPROV: DATE:
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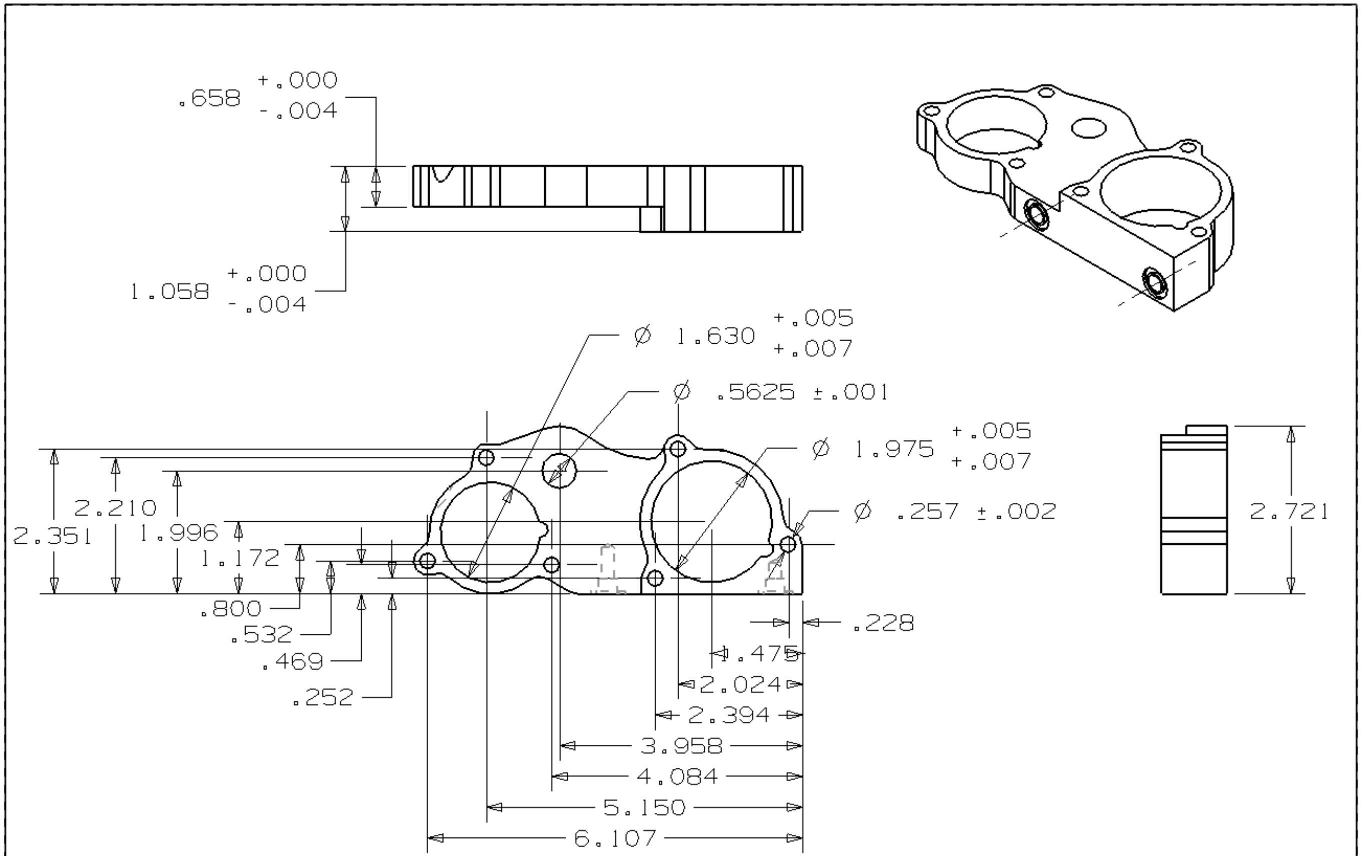

MATERIAL: 7075-T651 Aluminum

MRacing 2008

TITLE: Scavnege Pump Housing

VENDOR INFO:

DESIGNED: Team 32 DATE: 3/7/08
DRAWN: E. Kruk DATE:
CHECKED: DATE:
MFG. APPROV: A. Hutz DATE:
DES. APPROV: DATE:

FILE NAME: 08_Scavenge_Pump_ESK_11.prt

LATEST REV. 1

SHEET 1 OF 1

PART WT. xx lb.

DRAWING NOT TO SCALE
DO NOT SCALE DRAWING

UNITS INCHES [mm]

TOLERANCES UNLESS OTHERWISE SPECIFIED
X.XX ± 0.06 [1.6]
X.XXX ± 0.031 [0.79]

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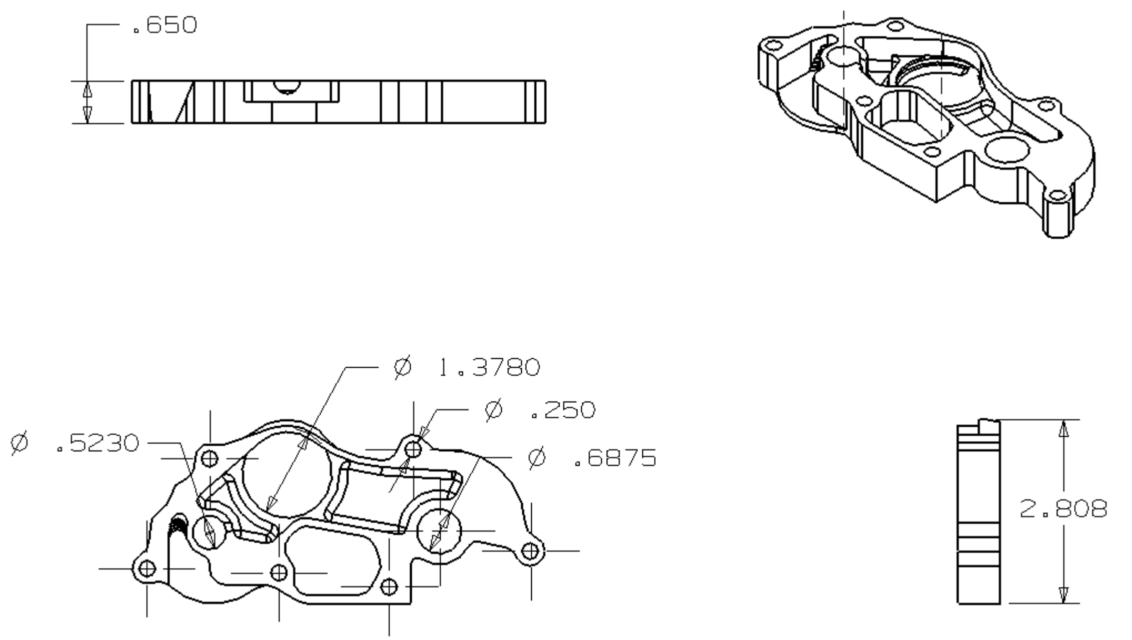

MATERIAL: 7075-T651 Aluminum	MRacing 2008	TITLE: Inlet and Outlet
VENDOR INFO:	DESIGNED: Team 32 DATE: 3/7/08 DRAWN: E. Kruk DATE: CHECKED: DATE: MFG. APPROV: A. Hutz DATE: DES. APPROV: DATE:	FILE NAME: 08_Scavenge_Pump_Outlet_ESK_12.prt
DRAWING NOT TO SCALE DO NOT SCALE DRAWING UNITS INCHES [mm] TOLERANCES UNLESS OTHERWISE SPECIFIED X.XX ± 0.06 [1.6] X.XXX ± 0.03 [0.79]		LATEST REV. 1 SHEET 1 OF 1 PART WT. xx lb.

SH1 (DWG) WORK (OUT-OF-DATE)

APPENDIX E - **BILL OF MATERIALS**

Team 32
Bill of Materials

Part #	Part Name	Qty	Material	Size	Manuf. Process	Function	Supplier/Sponsor
				(inches)			
001	Pump Housing	1	7075 Aluminum		CNC Mill	Houses Gerotors	In-House
002	Outlet Housing	1	7075 Aluminum		CNC Mill	Routes Fluid to Outlet	In-House
003	Water Pump Adaptor	1	7075 Aluminum		Manual Lathe	Allows Oil to Leave the Engine	In-House
004	NAS Fasteners	6	Steel	1/4" 28	None	Fasten Pump Components	McMaster
005	Helicoils	6	Stainless Steel	1/4" 28	None	Locks the fasteners	McMaster
006	O-rings	2	Buna-N	Multiple sizes	None	Seals the fluid	McMaster
007	Bushings	1	Steel		None	Supports shaft	McMaster
008	Needle Bearings	1	Steel		None	Supports shaft	McMaster
009	Sprocket	3	Steel	14 tooth, 1/4" pitch	Mill modifications	Drives shafts	Motion Industries
010	Chain	2	Steel	#25, (1/4") pitch	None	Drives shafts	Motion Industries
011	Shafts	3	8620 Steel		Provided by spronsor	Drives gerotors	Nichols Portland
012	Gerotor	1	Steel	Nichols Portland 4065	Provided by spronsor	Pump fluid	Nichols Portland
013	Gerotor	1	Steel	Nichols Portland4086	Provided by spronsor	Pump fluid	Nichols Portland
014	Cover Plates	2	6061 Aluminum	1/8" thick plate	Laser Cut - CNC Milled	Increase flow efficiency	Technique
015	Separator Sheet	1	Steel	0.032" steel plate	Laser Cut	Separates Flow between Housing and Outlets	Technique

APPENDIX F **- DESIGN ANALYSIS MATERIALS**

1	2	3	4	5	6	7	8	9			
Part ID Number	**Number of Times the Operation is Carried out Consecutively**	**Two-digit manual handling code**	**Manual handling time per part**	**Two-digit manual insertion code**	**Manual insertion time per part**	**Operation time (seconds)**	**Operation cost (cents)**	**Figure for estimation of theoretical minimum parts**	Scavenge Pump	**alpha angle**	**beta angle**
1	1	30	1.95	00	1.5	3.45	1.38		Housing	360	360
2	1	30	1.95	06	5.5	7.45	2.98		Inlet/Outlet Channels	360	360
3	1	30	1.95	06	5.5	7.45	2.98		Separation Plate	360	360
4	1	30	1.95	07	6.5	8.45	3.38		Water Pump Adapter	360	360
5	1	30	1.95	06	5.5	7.45	2.98		4086 Cover Plate	360	360
6	1	30	1.95	06	5.5	7.45	2.98		4065 Cover Plate	360	360
7	3	10	1.50	00	2	10.5	4.2		Shafts	360	0
8	2	91	3.00	31	5	16	6.4		Chains	360	360
9	4	20	1.80	00	2	15.2	6.08		Sprockets	360	180
10	6	10	1.50	59	12	81	32.4		Helicoils	360	0
11	6	10	1.50	92	5	39	15.6		Bolts	360	0
12	1	00	1.13	01	2.5	3.63	1.452		O-ring	0	180
13	1	00	1.13	51	9	10.13	4.052		Needle Bearing	180	0
14	1	10	1.50	51	9	10.5	4.2		Bushing	360	0
15	4	11	1.80	92	5	27.2	10.88		Set Screw	360	0
16	1	00	1.13	98	9	10.13	4.052		4086 Gerotor	0	180
17	1	00	1.13	98	9	10.13	4.052		4065 Gerotor	0	180
						275.12	**110.048**				

Table F-1: Design for Assembly Worksheet

Component	Material	Density (lb/cu in)	Volume	Weight
Chain 1	Steel	0.283	0.6	0.1698
Chain 2	Steel	0.283	0.46	0.13018
Sprocket 1	Steel	0.283	0.2	0.0566
Sprocket 2	Steel	0.283	0.2	0.0566
Sprocket 3	Steel	0.283	0.1	0.0283
Sprocket 4	Steel	0.283	0.2	0.0566
Main Housing	7075 Aluminum	0.102	5.78	0.56644
Housing Outlet	7075 Aluminum	0.102	3.81	0.37338
Middle Sheet	4130 Sheet	0.283	0.34	0.09622
4065 Cover	6061Aluminum	0.102	0.38	0.03724
4086 Cover	6061 Aluminum	0.102	0.58	0.05684
4065 Gerotor	Steel	0.283	1.22	0.34526
4086 Gerotor	Steel	0.283	2.25	0.63675
4065 Shaft	Steel	0.283	0.41	0.11603
4086 Shaft	Steel	0.283	0.49	0.13867

Table F – 2: Design for Environmental Sustainability Material Weights and Volumes

DesignSafe Report

Application:	Team 32 Risk Assessment	Analyst Name(s):	Tasos Charalambides, Elliot Kruk, Tony Ferrara, Alex Hutz
Description:	Risk Assessment for Custom Oil Scavenge Pump	Company:	MRacing
Product Identifier:		Facility Location:	Wilson Center
Assessment Type:	Preliminary		

Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
Chief Engineer inspect parts	mechanical : head bump on overhead objects Head bump on chassis bar	Minimal Frequent Possible	Moderate	Follow machine shop and OSEH rules closely. Be careful and patient.			
Chief Engineer inspect parts	slips / trips / falls : slip Slip because of engine oil leak	Minimal Occasional Unlikely	Low	Follow machine shop and OSEH rules closely. Be careful and patient.			
Chief Engineer inspect parts	ergonomics / human factors : posture Trying to reach hard-to-reach spots	Minimal Frequent Possible	Moderate	Follow machine shop and OSEH rules closely. Be careful and patient.			
Chief Engineer inspect parts	heat / temperature : burns / scalds Touching hot surfaces	Serious Remote Negligible	Low	Follow machine shop and OSEH rules closely. Be careful and patient.			
Chief Engineer trouble-shooting / problem solving	mechanical : head bump on overhead objects Head bump on chassis bar	Minimal Occasional Possible	Moderate	Follow machine shop and OSEH rules closely. Be careful and patient.			
Chief Engineer trouble-shooting / problem solving	mechanical : break up during operation Chain will break during operation	Catastrophic Remote Possible	High	Follow machine shop and OSEH rules closely. Be careful and patient.			
Design and Assembly Engineer inspect machinery	mechanical : head bump on overhead objects Head bump on chassis bar	Minimal Occasional Unlikely	Low	Follow machine shop and OSEH rules closely. Be careful and patient.			
Design and Assembly Engineer inspect machinery	heat / temperature : burns / scalds Touching hot surfaces	Slight Remote Negligible	Low	Follow machine shop and OSEH rules closely. Be careful and patient.			
Design and Assembly Engineer inspect machinery	chemical : irritant chemicals Handling engine oil	Minimal Frequent Negligible	Low	Follow machine shop and OSEH rules closely. Be careful and patient.			

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
Design and Assembly Engineer conduct tests	mechanical : head bump on overhead objects Head bump on chassis bar	Minimal Frequent Possible	Moderate	Follow machine shop and OSEH rules closely. Be careful and patient.			
Design and Assembly Engineer conduct tests	slips / trips / falls : slip Slip because of engine oil leak	Minimal Occasional Unlikely	Low	Follow machine shop and OSEH rules closely. Be careful and patient.			
Design and Assembly Engineer conduct tests	ergonomics / human factors : posture Trying to reach hard-to-reach spots	Minimal Occasional Possible	Moderate	Follow machine shop and OSEH rules closely. Be careful and patient.			
Design and Assembly Engineer conduct tests	heat / temperature : burns / scalds Touching hot surfaces	Serious Remote Negligible	Low	Follow machine shop and OSEH rules closely. Be careful and patient.			
Design and Assembly Engineer conduct tests	environmental / industrial hygiene : carcinogens Handling engine oil	Serious Frequent Negligible	Moderate	Follow machine shop and OSEH rules closely. Be careful and patient.			

APPENDIX G - FINAL PROTOTYPE PICTURES