Design, Construction, and Testing of a Model Hydrokinetic Prototype

By

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ABSTRACT OF THE THESIS

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Tidal power represents an excellent renewable energy resource for the United States, but its economics must be reassessed in such a way that it becomes cost competitive with fossil fuels. In order to reduce operating costs, Sunlight Photonics Inc., in conjunction with Rutgers University, has designed and tested a modified tidal current system which utilizes an underwater hydraulic energy transfer system instead of the current underwater turbine-generator assembly, a design which experiences high failure rates and is expensive to build and maintain. In the modified design, generators and all electrical components are relegated to on-land electrical stations, while the underwater system consists of a tidal turbine and hydraulic pump assembly; as the tidal turbine spins, a hydraulic pump attached to its drive shaft creates high pressure fluid. This high pressure fluid is sent to land, where it can produce electricity in a well-controlled environment.

The prototype was designed with knowledge gained from extensive research into centrifugal and positive displacement pumps, gearboxes, and hydraulic fluids. The prototype utilizes a radial piston pump engaged to a 20 horsepower motor via a 14.63:1 reduction helical gearbox to simulate the power potential of the tides. An axial piston motor linked to a 460 volt three-phase alternating current generator and resistive load bank helps to simulate the effects of an active power plant on the hydraulic circuit. Testing showed that a heavily loaded generator produces greater pressure differentials between the high and low pressure sides of the hydraulic circuit than an unloaded generator. System efficiency of 20%, which is lower than anticipated, is traced to an underperforming generator and improperly sized hydraulic circuit. Recommended design modifications include a resized axial piston motor and generator assembly to help increase system efficiency to competitive levels of 70% or greater.

Acknowledgements

My deepest thanks first go to my advisor, Dr. Kimberly Cook-Chennault, for her countless hours of counsel and mentoring. Dr. Cook's impact upon my time at Rutgers can be summed in one powerful quote from John Quincy Adams: "If your actions inspire others to dream more, learn more, do more and become more, you are a leader." From components needed for the experiment, to advice on academic and professional endeavors, Dr. Cook was always ready to help in any way possible. She provided the motivation, guidance and enthusiasm which made the whole project a reality and whether she realizes it or not, her role as my advisor has had a dramatic impact on my life. Thank you.

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Any good prototype would not be possible without the numerous people who made its construction a reality. To wit, I extend my thanks to Melvin Braxton and Cali, the electrician of Rutgers Facilities, for their countless hours planning and installing the complicated electrical circuitry needed in the testing room. I also extend my thanks to Ron from Academy Rigging for assistance in assembling (more than once) the prototype.

As in any prototype, last minute changes and adjustments to components are bound to occur. During this frenetic period, a capable and quick machine shop can be the difference between downtimes of hours and weeks. For that, I extend my sincerest thanks to Bill Schneider and his team of highly professional, qualified, and experienced machinists at the Physics Machine Shop on Busch Campus of Rutgers University for their advice, guidance, and quick turn-around on part build and modification. They are truly invaluable.

Whenever tools were needed or odd questions were to be answered, John Petrowski and Joseph Vanderveer of the Mechanical Engineering Department at Rutgers University were ready to help, and for that I give my thanks.

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Chapter 1.

Introduction

1.1. The Challenge of Providing Safe, Efficient and Reliable Power

Tidal energy is currently too expensive compared to fossil fuels to be worth the infrastructure investment [1]. As a nation seeking energy independence, tidal power represents an excellent source of power for the United States, but its economics must be reassessed in such a way that it becomes cost competitive with fossil fuels. Companies have recently changed from damming tidal flows to using run of the river methods, which are considerably less expensive to build and faster to bring on-line [2]. However, even with this progress, tidal energy is expensive to build and maintain and is only active twice a day, often not during peak power demand periods making it disruptive and irrelevant to national energy needs [3].

One can compare the relative cost of different power plant technologies through two parameters: capital cost and cost of energy. Capital cost considers all the funding needed to build and bring a power plant technology online. Measured in dollars per kilowatt hour of capacity, capital cost for tidal stream technologies is almost double that of coal power plants. However, the cost of energy (also known as the daily costs to operate the facility) is similar between the two technologies, as seen in Table 1. This key point shows that tidal energy is a feasible alternative to non-renewables, but needs to be engineered so as to have lower installation costs.

Table 1: Cost comparison between non-renewable and renewable power plants.

Power Plant	Capital Cost (Dollars Per kW Capacity)	Cost of Energy (Cents per kW produced)
Coal [4]	1275	4.2
Tidal In Stream [4]	2000	4-6.5
Wind [4]	1150	4.7-6.5
Solar Thermal Trough [4]	3300	18

To make tidal power less expensive to install and more economically sound and relevant, Sunlight Photonics Inc., in conjunction with Rutgers University, proposes a tidal current system which avoids underwater electricity generation and distribution via wire conduit. Instead, a hydraulic system would pressurize fluid with the kinetic energy of the tides. This system would then transfer the kinetic energy of the tides to an on-shore generation station, where the complicated components of electricity generation can harness the energy at a fraction of the cost of underwater units. The hydraulic transfer unit also allows for construction of high pressure tanks, wherein pressurized fluid could be stored until needed during peak demand hours, increasing the flexibility and relevance of tidal energy [5]. With these modifications, it is possible that tidal energy could become cost-competitive with fossil fuels, helping to lead a renewable energy revolution. To this end, Sunlight Photonics Inc. and Rutgers University have built and tested a prototype small-scale version of the initial concept to explore the validity and practicality of this proposed concept.

In this thesis, the first chapter provides extensive data and findings on major renewable energy sources, in order to provide the basis and importance of tidal current energy as a viable alternative energy source. Chapter two explores the initial concept in detail, including its origin, its competitiveness with currently available tidal current technologies, and the specific details of its construction that will make tidal current technology a feasible energy source.

A detailed description and analysis of the prototype is covered in chapter three, wherein the aim is to provide validity to the initial concept. In this chapter, the rationale behind why each component was chosen is provided. Furthermore, the final hydraulic circuit for the prototype is described and an explanation for the method of design is described. For example, an overview of the selection and assembly of pipes, requisite fittings and ancillary components is given. In chapter four, empirical results from initial, intermediate, and final testing, along with consequential prototype modifications, are described. Chapter five summarizes the findings of the project and provides a feasibility assessment for future development.

1.2. The Current State of Energy Use in the United States

Currently, near 92% of energy consumption in the United State is derived from non-renewable energy sources, of which more than 50% are the carbon intensive sources coal and petroleum, as shown in Figure 1 [6]. The finite nature of non-renewable energy sources, above all else, indicates an insecure future wherein a challenge to available non-renewable supplies may occur. Petroleum, for example, is over 65% imported, putting the nation in a position of compromised energy security [7]. With this dependence, the

United States is vulnerable to disruptions from political agendas of petroleum rich nations, international demand, and natural disasters [7]. Developing renewable, domestically-sourced energy resources is becoming increasingly important to the long-term stability of the United States energy portfolio.

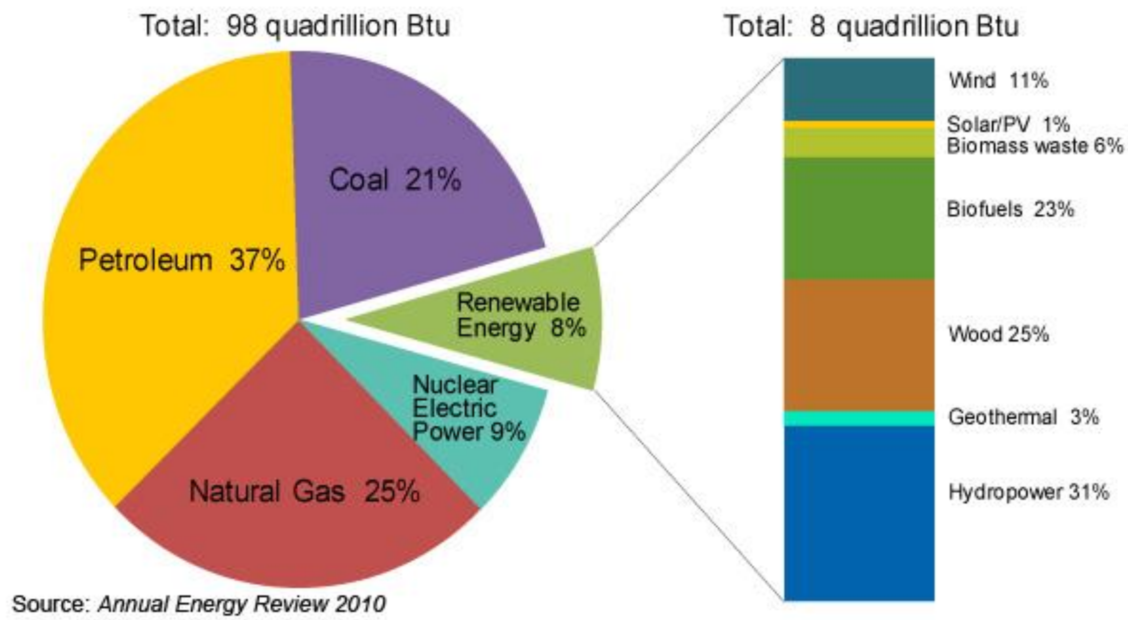


Figure 1: Energy Source Distribution for the United States as of 2010. Courtesy of the United States Department of Energy [6].

Additionally, the carbon intensive nature of some non-renewable energy sources is adding significant amounts of carbon particulates to the atmosphere [8]. These particulates, as shown in Figure 2, are expected to climb significantly over the next several decades as developing nations like India and China become dominant members of the world economy. Several renewable energy sources, including solar, wind, geothermal, and tidal current technologies emit no carbon particulates in production of electricity [9]. Additionally, these sources are all domestically available, adding stability to the national energy portfolio. Finally, all renewable energy sources, managed effectively, are infinite in nature and can provide a reliable source of power for future generations [9].

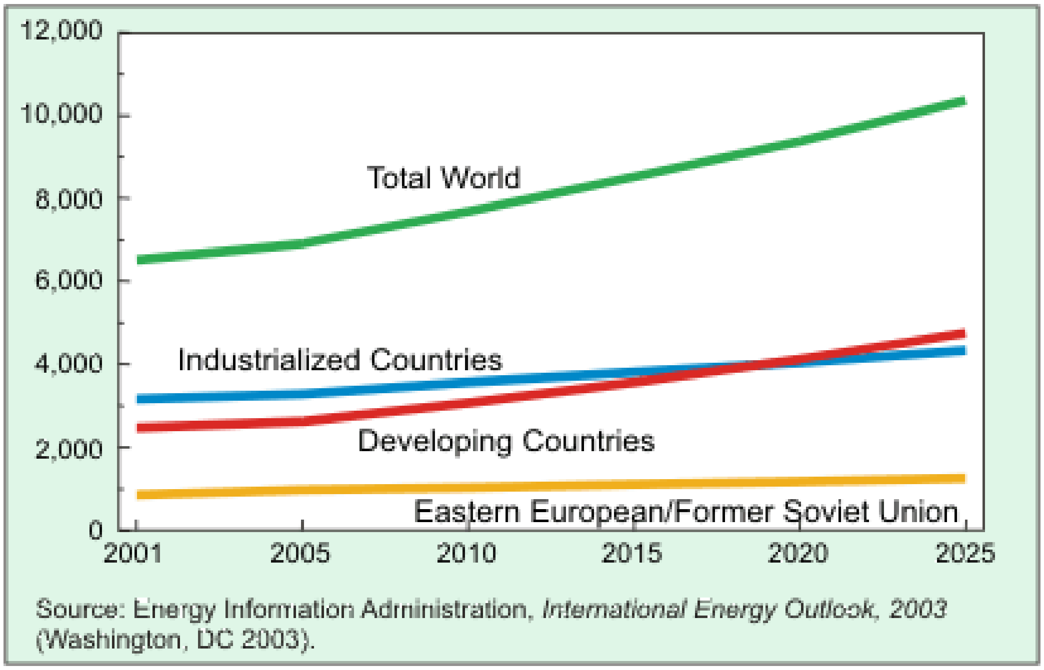


Figure 2: Carbon Emissions Predictions Through 2025 (in millions of tons). Courtesy of the United States Department of Energy [10].

1.3. Wind Power

Wind power is a renewable that shows intense promise for the future. Recent studies at Stanford University have shown that harnessing just one fifth of the United States wind energy would provide up to seven times the current energy needs of the world [11]. While promising, wind power is not without its problems. When wind sources temporarily stop, so does electricity production. Noise from the turbines is also prevalent, especially in older models. Many residents near wind farms also complain of light flickering effects, a disturbing phenomenon which occurs as the Sun rises and sets behind a rotating turbine. These minor effects all add to create a not-in-my-back-yard (NIMBY) effect that has caused challenges for the wind industry as a whole [12].

1.4. Solar Power

Solar energy resources are not without their problems either. While often considered the flagship in the "green" movement, photovoltaic solar panels are notoriously inefficient. Operating at less than 20% capacity due to design limitations, these panels add a tremendous cost and physical space requirement for electricity generation equivalent to non-renewables [13]. Currently, non-renewables such as coal and natural gas give electricity costs of about 4 to 5 cents per kilowatt; solar provides electricity at 12 to 14 cents per kilowatt hour [14]. In addition to this, solar has only experienced growth in recent years due to large subsidies from governments; without the subsidies, the cost of solar (along with many technologies) is prohibitive to the average consumer [15].

1.5. Geothermal Energy

Geothermal technologies utilize intense heat sources from the Earth which have been around for millions of years, making it the most stable renewable of the group. Geothermal energy works by one of several methods. The first sends coolant pipes into hot, dry rock where the coolant is heated until it boils, at which point it is re-circulated to the surface. This boiling coolant's heat is extracted, creating power. The next method entails directly pulling hot water from pressurized water pockets located deep underground, whereupon again, the heat is extracted. Finally, the most common and least expensive method is to utilize near surface reservoirs of hot water. These reservoirs are hotter than most due to proximity to merging or emerging tectonic plates, plates which expose magma to nearby water sources. This hot water, as with other methods, has its heat extracted for power generation [16].

Geothermal sources fall into two main categories of development. The first category represents sources which are readily available. Such sources, such as surface reservoirs, have largely been tapped by countries on tectonic plate boundaries, like Iceland. This form of energy has expanded to its peak, and will grow no further in the future. It is also highly location dependent, and requires considerable infrastructure development to spread its generated electricity to all parts of the nation. The second category represents sources where significant drilling or research must be performed before the source can be used. Pressurized hot water pockets and hot, dry rock sources fall into this category. While these sources can provide local, clean, reliable power to most parts of many nations, they are currently very expensive and rarely, if ever, attempted. Further research and expansive budgets will be necessary for long-term success [17].

1.6. Hydropower

Hydropower is yet another viable, albeit environmentally impactful solution for renewable energy. Here, we define hydropower as any electricity generation source which utilizes a dammed source of water forced to flow through a turbine or equivalent generation source. Currently, the success rate of hydropower has been very high, with approximately 20% of the world's electricity derived from hydropower [18]. However, hydropower has several key negative attributes which must be fully understood before the benefits can be realized. They are:

1) Cost: Hydropower dams are extremely expensive to build, and require many years to be built properly. Hence, they often experience cost overruns. [19].

2) Eminent Domain: Areas upriver of the dam must be evacuated permanently to make room for land which will be flooded. Such displacement often causes negative publicity, changes to the local economy, and hardship for citizens [19].

3) Failure: Dams that fail can cause extensive and widespread flooding and even death to those citizens downstream of the breach [19].

4) River Rights: Rivers which flow through multiple nations, when dammed, cause changes in flow rates through neighboring nations. This can lead to contentious debates over rights to the river, and destroy international relationships [19].

5) Ecology Change: Poorly planned hydroelectric dams can significantly alter downriver ecology by reducing available water supply. For example, the Three

Gorges Dam in China currently experiences severe drought and reduced vegetation downriver of the dam. [20]

1.7. *Oceanic Wave Energy Solutions*

1.7.1. Wave Formation

Oceanic wave energy is a renewable energy source most are familiar with, and are exposed to regularly. Waves, especially ocean waves, are a product of prevailing winds skirting across large bodies of water. Since winds are almost never stable, the strength and intensity of waves is likewise difficult to predict or model. Some understood behaviors of waves are that: 1) during oceanic storms, waves become stronger, 2) when winds are calmer, waves are weaker, and 3) when waves reach shore, they become shorter, slower, and taller in response to the sloping landscape [21].

1.7.2. Literature Review: Wave Technologies

Both tidal and wave technologies seek to generate usable power from oceanic water bodies. How they accomplish this goal, while different on the macro level, has in fact many similarities to the way energy is extracted at the micro level. For example, the Wave Dragon from the Danish firm Wave Dragon ApS, creates electricity by pooling wave water in a container (referred to as “overtopping”) and forcing it through a turbine to exit the container, as shown in Figure 3 [22]. Power produced by turbine rotation is a common theme in hydrokinetic devices, but the Wave Dragon departs from standard hydroelectric dams by its emphasis on reducing environmental impact [22]. Special mesh coverings prevent marine life from being harmed by the turbines, and water-hydraulics are used to prevent oil contamination of the ecosphere should a leak occur [22].

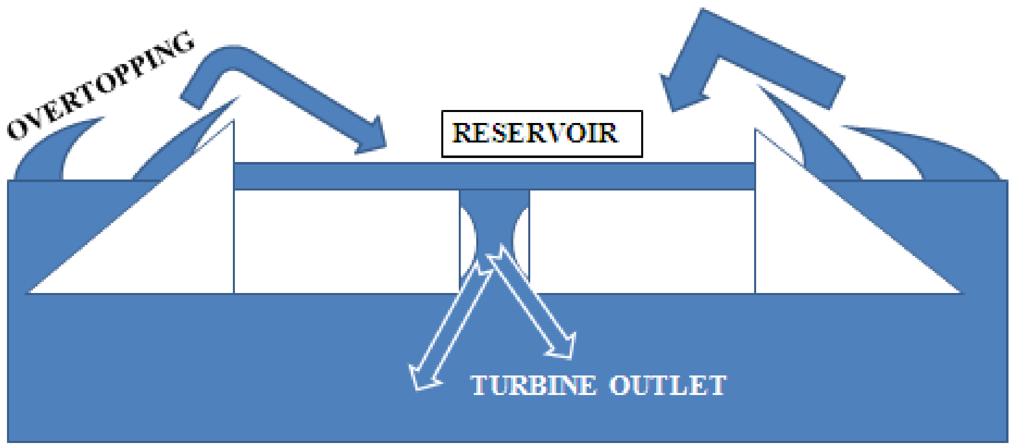


Figure 3: Wave Dragon device pools water in a reservoir and only allows drainage through the turbine outlet.

The Pelamis wave energy converter works differently from the Wave Dragon, but emphasizes environmentally sound hydraulics as well. Instead of pooling water and harnessing power akin to a dam, Pelamis is shaped like a multi-sectioned snake that rocks and bobs in the waves, as shown in Figure 4 [23]. Inside of Pelamis, pistons connected to the rotating joints compress hydraulic fluid in response to the energy of the waves. This compressed hydraulic fluid then powers a generator [23]. The sealed hydraulic system is further encased in the watertight snake body of Pelamis, thereby providing additional environmental protection [24].

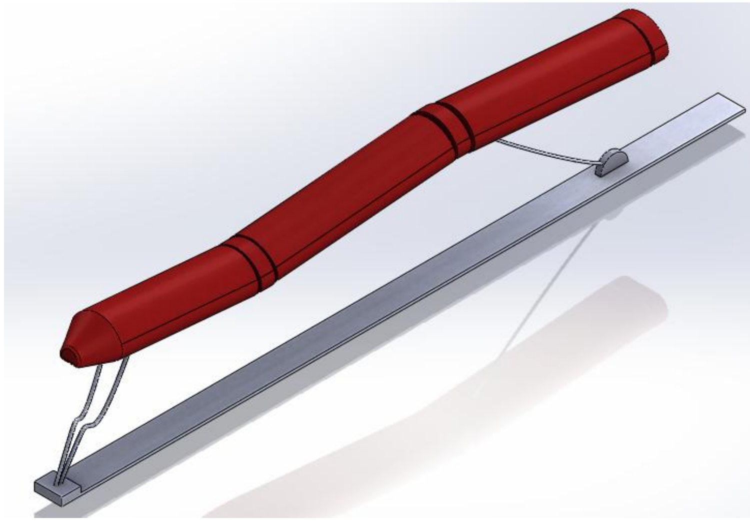


Figure 4: Pelamis wave energy converter is comprised of several sections that rock independently of each other. Electrical production is entirely self-contained.

Also utilizing hydraulics are the BioWAVE and Aquamarine Oyster Power devices, which are conceptually similar. In both devices, a large steel or reinforced plastic panel literally rocks back and forth under power of the waves, near the shoreline. Attached to this panel, on the ocean floor, is the mounting carriage and an energy conversion device [25] [26]. At this point, the systems differ significantly. BioWAVE takes the kinetic energy transferred to the panel and uses it to compress and pressurize hydraulic fluid at a depth of up to 45 meters, as seen in Figure 5. This pressurized fluid then spins an electric generator, producing power at the source. The power created is then transferred to land via subsea electric cabling [25].

Figure 5: BioWAVE energy capture device rocks fore and aft, compressing fluid and generating electricity all at the source.

Aquamarine uses a system which eliminates electrical cabling under water, a newer concept in the field of marine hydrokinetics. In this system, the rocking panel (similar in appearance to BioWAVE) transfers its kinetic energy to two hydraulic pistons at a depth of up to 15 meters. These pistons pressurize water, which is then transferred to land via high pressure hoses, shown in Figure 6 [26]. Immediately on-land is a power generation station, which takes the high pressure fluid and utilizes it to power the conventional hydroelectric system located at the station. In this design, all the electrical components are located on land, where these corrosion-susceptible components can be easily managed and maintained [26]. Unlike the Pelamis, Wave Dragon, and bioWAVE, the Aquamarine Oyster system keeps a majority of its complex internal components out of the water, thereby reducing operating costs significantly. Repairs which must be assessed at the site also do not require SCUBA trained electricians, a rare and expensive personnel requirement for the other technologies [26].

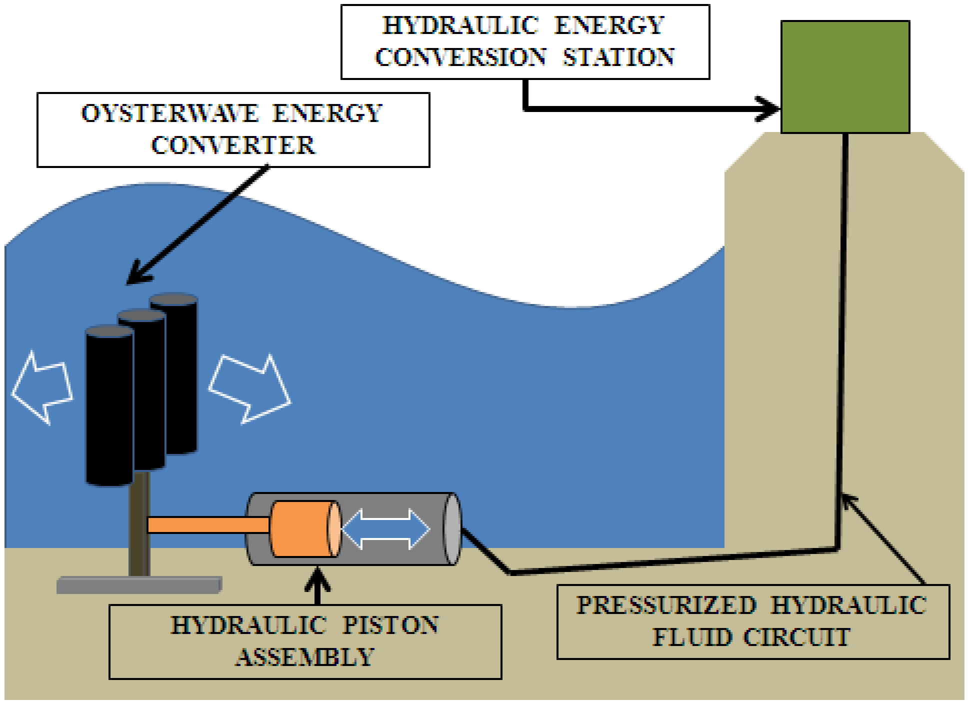


Figure 6: With a look similar to BioWAVE, Aquamarine Oysterwave creates high pressure fluid from the rocking motion of its energy converter; it then transfers high pressure hydraulic fluid to land for electricity generation.

Ocean wave column energy harnessing devices, or OWCs for short, represent the other major type of wave technology. These devices sit on top of the waves, and allow waves to rise and fall within a container located at the bottom of the device. As the waves rise and fall, they cause an air pocket in the top of the container to compress or expand (Figure 7). In order to prevent a vacuum, a ventilation shaft at the top of the container, with a turbine-generator assembly attached, allows air flow. When a wave falls, air must be let into the container, thereby rotating the generator. When a wave rises, air in the container compresses and pressure is released via the same port, also turning the turbine-generator assembly [27].

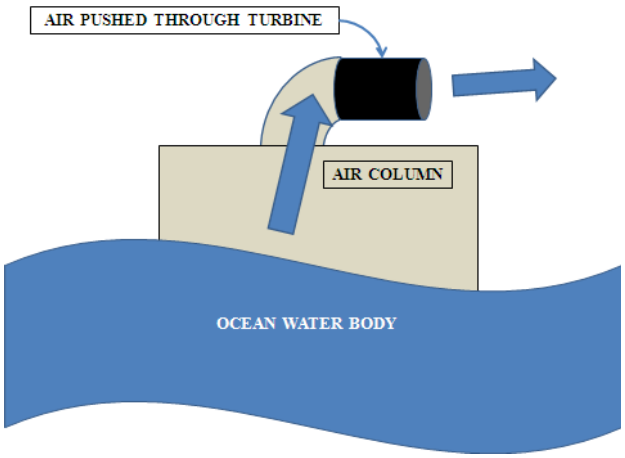


Figure 7: Typical ocean water column device. Air column is compressed or expanded depending on height of water inside column.

The Voith Energy Group's WaveGen technology is a water-column technology that utilizes a Wells-style turbine to achieve desired efficiency (Figure 8). Since a turbine that can harness air flow in both forward and reverse directions is ideal, the Wells-style turbine is a good fit. This type of turbine utilizes a double pitch blade in the shape of a V. As air flows in either direction, the turbine only rotates in one direction [27].

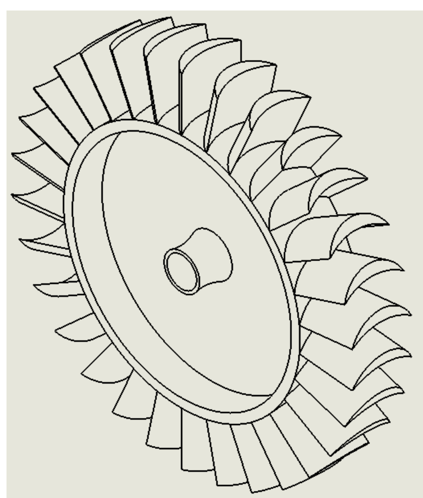


Figure 8: Wells turbine, commonly used on ocean water columns.

Conversely, the airWAVE OWC from Oceanlinx is conceptually similar to

WaveGen, but instead of a Wells turbine, utilizes a Dennis-Auld style turbine to harness moving air [28]. The Dennis-Auld turbine is a variable pitch turbine, controlled remotely via a computer program to control motion, servo motors to twist the blades as needed, and sensors to monitor the performance. Efficiency is as high as 60% (compared to 30% for fixed pitch blade units like the Wells turbine model) [28]. Ideally, the airWAVE unit is able to spin continuously in one direction by reversing turbine blade direction when air flow direction changes. Additionally, as waves become stronger, the Dennis-Auld unit can increase or decrease blade pitch as needed to maximize energy capture [28].

Despite the plethora of wave technologies currently available, the key fact remains that waves are an unpredictable force. Days of strong waves, such as during hurricanes and other storms, could overwhelm many of the wave technologies presented [29]. In order for these technologies to stay operational they need to be built out of excessively robust materials compared to tidal technologies. Where wave technologies are “hammered” by the effects of the waves, tidal technologies sit well below the surface of the water, avoiding much of the tumult that wave harnessing equipment is subject to.

Additionally, wave power is an uneven, unpredictable power source [29]. During calm days, electrical production will be low. However, during storms, electrical production will rise significantly, provided the harnessing equipment is not damaged. These surges make wave technologies difficult to integrate into the electrical grid. Tidal flows, in comparison, are predictable and steady, giving electrical utilities the ability to integrate the technology more easily. The variability of wave energy makes it a poor choice for mass deployment. Only when the power can be produced in a smooth, predictable manner will wave technology be a viable option.

1.8. Tidal Range Power

Power from the tides can be harnessed in a multitude of ways. One previously used, well-understood method of power generation is *tidal range* power. This precursor to tidal current technologies has provided much of the valuable information which underpins tidal current technologies today. Tidal range power helped advance knowledge of tidal power potential, and was the avenue though which many power harnessing methods were tested and employed, as shown in the section. Unfortunately, due to its ecological impact and high construction costs, tidal range power has largely been eschewed for the more ecologically benign tidal current system.

Tidal range power originates at an inlet, which is classified as a body of water which has only one entrance and exit to the ocean. One such location is the Severn Estuary in Great Britain. In these zones, when a barrage is erected at the opening, the creation of a *basin* is established. It is in such a location that a tidal range power plant can be built. The barrage used to create the basin has turbines installed at its inlet and outlet ports [30].

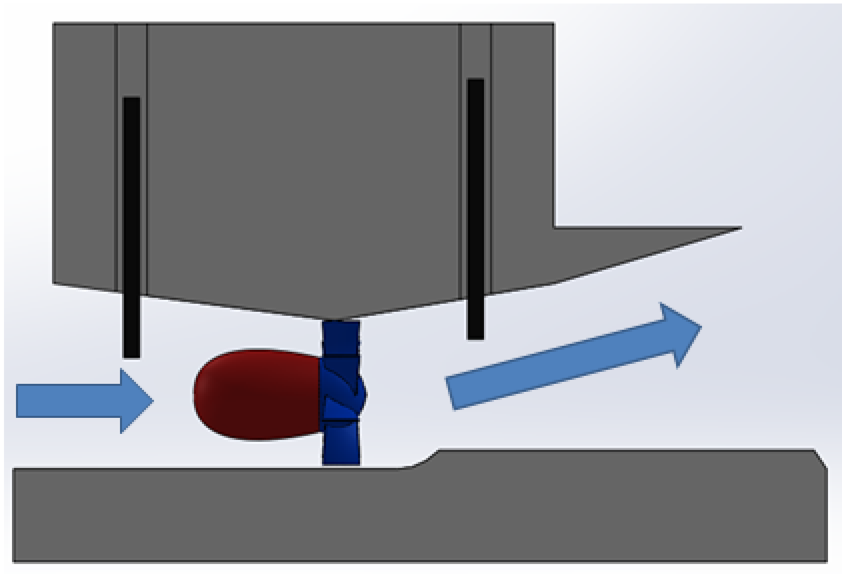


Figure 9: Tidal barrage uses oceanic barrier (in grey) to force water through port. Water flow is demarcated via arrows; notice that flow is past bulb-style generator/turbine unit.

Operation of the tidal range power plant is quite simple. As the tides rise, they cause water to flow through the inlet ports of the barrage into the basin area. As high tide begins to recede, the barrage prevents quick escape of the water from the basin to the ocean. The water is forced to flow through the ports (Figure 9) causing rotation of the turbine blades, and hence, generation of electricity. Variations on design of the turbines, ports, and basin all contribute to electricity output variations [31].

1.8.1. Turbine Flow Considerations

One major consideration is turbine flow direction. On the surface, it appears that having the turbines generate electricity on both the ebb and flow sides of the cycle would be highly advantageous. In such a system, it appears that electricity would be generated during all times of water flow, giving maximum output. However, this is never the case, and actually provides insight into the complexity of these turbines [32].

In order for the turbines to effectively provide power, their vanes must be oriented in such a way as to maximize rotation in the water flow. Having turbines with rotatable vanes increases the mass and complexity of the turbine. With increased mass comes increased inertia, making it more difficult to get the turbine to rotate. This makes it much harder to reverse direction once water flow has reduced and the reverse direction is desired. Additionally, rotation of the turbine vanes takes considerable time, a period of time when power generation will drop notably. In summary, the inclusion of extra weight and cost exacts a monetary increment upon total system costs of 10 to 15%, a value which at this economic point in time makes such design propositions a moot point [32].

1.8.2. Turbine Port Design

Port design represents another area of considerable modification, with most modifications being concerned with the cost-benefit relationship of the turbine style. One such design places the turbine and generation unit at the center of the port. Simple shafts connect to the top and bottom of the port; they serve to stabilize the turbine/generator unit and house electrical cabling for the unit. This design, known as the *bulb model,* requires that more ports be installed than for other designs; the bulky center bulb restricts water flow through the port (Figure 10). Additionally, maintenance and system adjustments require draining of the basin *and* an exterior blockade to keep tidal flows from entering the area during serving [33].

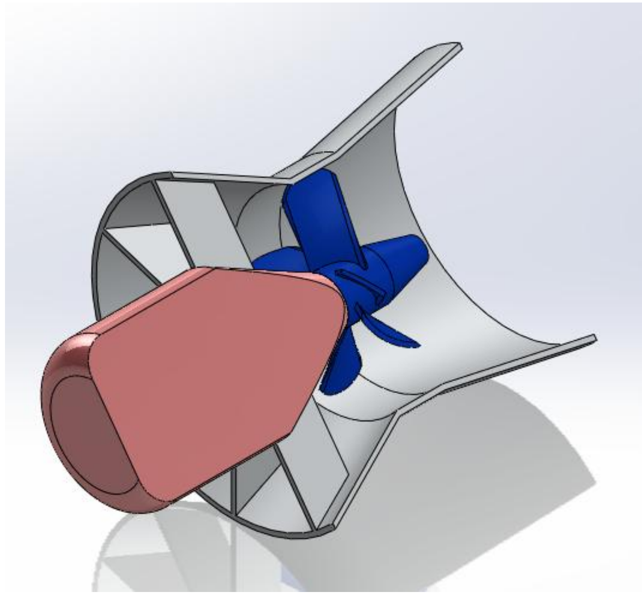


Figure 10: Bulb design for tidal turbine. Cut-away generator, shown on left, is completely submerged in fluid flow, limiting access for repairs.

Another design is the *straflow model,* where the generator is mounted radially about the port, as shown in Figure 11. In this design, only the turbine blades sit in the water, with a minor center cylinder impinging water flow in order to stabilize the turbine blades. The outer rim of the turbine blades is connected to the generator. This sleek design allows for better water flow, but requires an unusual generator design to operate, which can increase cost and complexity, culminating in expensive repairs. While harnessing power from this design is easy, modulating the system performance is difficult (i.e. rotational speed) and therefore, this design is a poor choice for forced pumping systems [33].

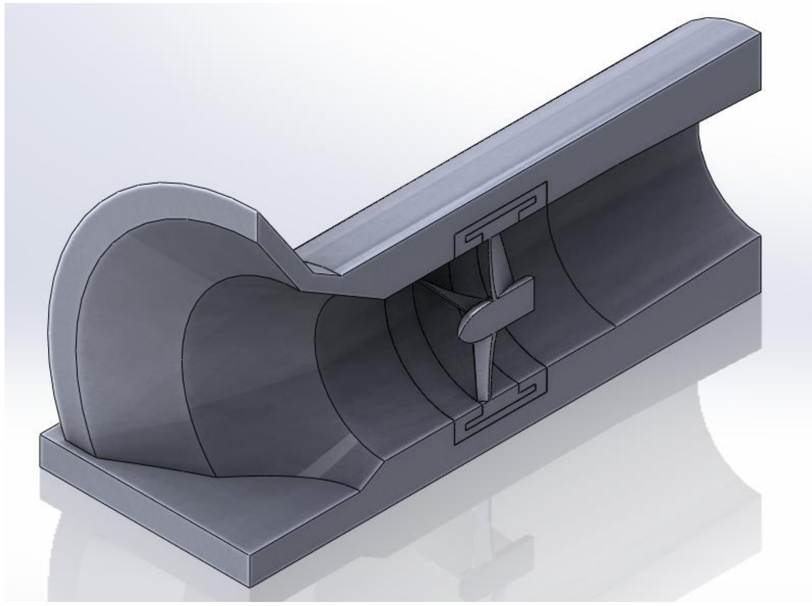


Figure 11: Straflow design attaches fins to outer rim, where generator is also attached. Flow through center shaft experiences less restriction than in other designs.

The final major design is the *tubular turbine model*, a design which shows the greatest promise for its compromise of simplicity and accessibility. This model actually adjusts the port space by installing a shallow mound on the lower portion of the port flow path. The turbine blade unit, also known as a runner, sits on one side of the mound, itself then offset from the horizontal, usually at a shallow angle of twenty to thirty degrees with respect to the horizontal (Figure 12). The runner's shaft, sleek in design and attached at one end to the turbine itself, then emerges at a slant from the slanted flow path onto the top of the barrage (or at a dry location within the barrage). At this location, a standard service generator is rotated by the shaft, providing power of the turbine *remotely.* While seemingly complicated, the overall design actually has little impact on flow if the offset is minor (less than 20 degrees). Additionally, serviceability is maximized, and choice of generator is greatly improved. This model allows use of common componentry which has greater access and serviceability, significantly driving down development, construction, and maintenance costs [33].

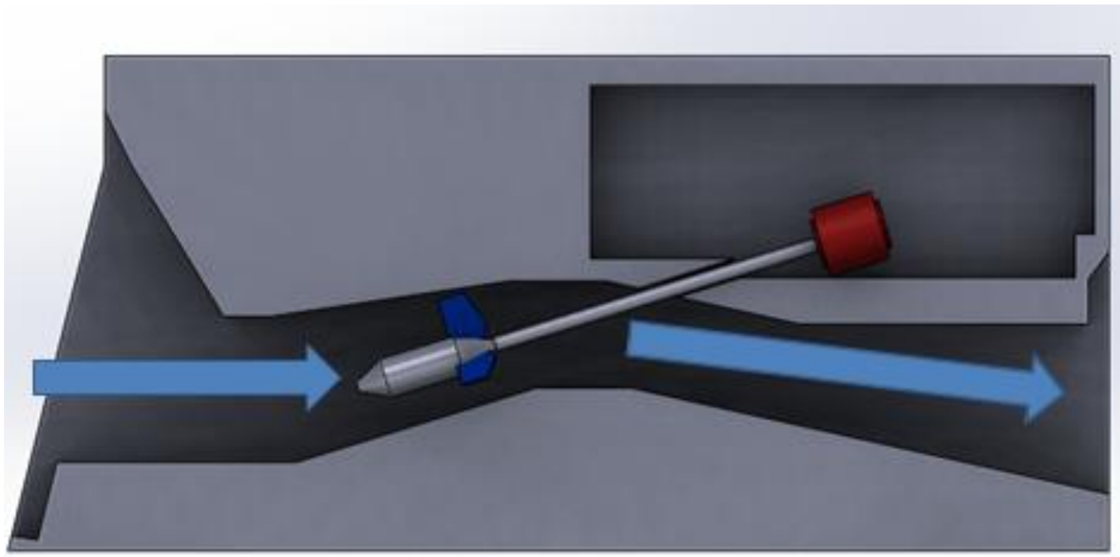


Figure 12: Generator unit sits away from flow path, reducing failure rates, but increasing initial build complexity.

1.8.3. Basin Creation and Ecological Considerations

The major concern of tidal range systems is the impact of the basin upon the local ecology. Before barrages are installed, the bay of interest often has excellent drainage and strong local fish populations, depending on the size and location of the bay. When barrages are erected, local fish and other aquatic populations often have difficulty entering or exiting the area. In the case of the birds, the change in local ecology often disrupts their migratory patterns. The altered ecosystem may no longer be viable for them [34].

High and low tides also offer an opportunity for parts of the local land to be exposed or flooded regularly. In the case of very shallow areas, like the Severn Estuary in Great Britain, exposure of large mud flats becomes prominent during low tide periods.

While not visually pleasing, exposed mud flats allow excellent feeding options for local bird populations, something which would be lost if the area remains regularly flooded [34].

Some benefits do exist to barrage creation, however. Calmer waters are clearer waters. Those zones which experience high and low tides are often constantly moving, making turbidity an issue. High turbidity causes waters which are difficult to see in, as the silt and sedimentary layers are constantly being churned. With calmer waters, vegetative growth is more likely at lower levels (due to sunlight penetration) and clearer waters become aesthetically pleasing for tourists and locals alike. Additionally, calm waters allow for water sports, including water skiing, boating, and other pleasantries where previously, rough waters did not allow such activities. Boosted tourism may result, helping to generate a stronger economy. Finally, barrages help reduce the likelihood of damage from storm surge or the like, as it acts as a natural barrier to the effects of severe oceanic storm systems [34].

1.8.4. Tidal Range Power Estimates

Tidal range systems see variations in power from a wide variety of sources, the majority of which is aforementioned. One can anticipate, however, that the system in question will have a net amount of power it can harness; the efficiency is a matter best left to another day. The available power for the turbines to harness is a product of the water height and area in the basin, a value which is highly dependent on the tidal force at that period. We consider the average power available for a *lunar month*, a value which will be maximum at the period of spring tide, and minimum at the period of neap tide

[35].

The basic energy equation for fluid fall in a column is:

$$E = mgh \tag{1}$$

where m is the mass of the fluid, g is the gravity constant, and h is the height of the fluid. To calculate the average power of one tide we need to know the average height of the fluid. Presuming a height of R at high tide and a height of 0 (ground level) at low tide to exist, we average them for a final average fluid height value of R/2. Also, we can use the standard mass value of a fluid as ρAR, where ρ is the density of the fluid, A is the surface area, and R is the height of the fluid column [35]. Therefore, the energy equation becomes:

$$E = \rho A R g \frac{R}{2} \tag{2}$$

for the energy of a single tide.

The power of the tides over a lunar month period can be further approximated as:

$$P = \frac{\rho A R^2 g}{2\tau} \tag{3}$$

if we state that the spring and neap tides together give an average tidal maximum height of R over the lunar period of τ [35].

An excellent example of tidal range power generation in action is the La Rance

Tidal Power Station in northern France, near Brittany. La Rance Tidal Power Station produces 240 MW of power through twenty-four 10 MW turbines. The barrage used, equipped with a ship lock for maritime traffic, is 740 meters long and dams a 22 kilometer long basin. While it originally produced power in ebb and flow directions, complexities with turbine technology available at the time forced a change to power production in one direction only [36].

1.9. Tidal Current Power

Whereas tidal range power utilizes a basin of water as its innate source of power, the tidal current system utilizes the flow of the tide, as experienced on the coast naturally, to rotate free standing, fully submerged turbines. It forgoes much of the complexity and ecological impact of the tidal range system for a sleeker, lower impact design. While a basin is not necessary for a successful tidal current system, a large range between high and low tides, R, is necessary. Those areas which experience larger ranges have longer and stronger ebb periods than areas with small ranges. This culminates in ultimately faster fluid velocity rates, a characteristic which strongly determines the successfulness of a tidal current system [35].

1.9.1. Maximum Output Considerations

Calculation of the height range between high and low tides requires consideration of the neap and spring tides for maximum accuracy. If we consider the neap tide, R_n, to be a portion α of the spring tide, R_s, such that:

$$R_n = \alpha R_s \tag{4}$$

then we can further include the realization that the variation between spring and neap tides is sinusoidal. The sinusoidal property gives us a complicated, but accurate realization of the range as:

$$R = \frac{R_S}{2} \quad 1 + \alpha \;\; + \;\; 1 - \alpha \;\; \sin \;\; \frac{4\pi t}{T} \tag{5}$$

where T is the lunar month period of about 27 days, and t is our period of consideration [35].

Since tidal current systems use the flow of the body of water for rotation of turbines, power output is directly related to the velocity of the river. The average power density expected from a tidal current system is only about 40% of the total power available due to logistical issues (i.e. that turbines cannot be located in every cubic meter of the tidal zone.) The average power density of the water source is calculated by:

$$Q = \frac{\rho U^3}{2} \tag{6}$$

where Q is the power density measured in kWm^{-2}, ρ is the water density measured in kgm^{-3}, and U is the water velocity measured in ms^{-1} [35]. The tide's sinusoidal property comes into question when calculating the precise tide velocity at any one period. While the tidal *range* is indeed sinusoidal, its *velocity* is also sinusoidal due to the interconnected relationship between tidal height and velocity. Therefore, we can model the tidal velocity at any time t as:

$$U = U_0 \sin \frac{2\pi t}{\tau} \tag{7}$$

knowing that the period of the tide is τ, which is about 12 hours 25 minutes, and U_0 is the maximum tidal velocity for that period [35].

1.9.2. Regulating and Controlling Power Output of Tidal Current Systems

Control and optimization of tidal current systems is critical to the long term success of this technology. Due to their variable power delivery, tidal current systems tend to disrupt, rather than aid, the national power grid. Methods which provide smoother, more predicable power delivery will ensure a better fit to societal power needs. J.A. Clarke et al. suggest that smaller generators be used at each turbine. The assembly would produce peak power earlier and longer during the harnessing period; this would in turn contribute to a more stable power production during the generation cycle [37]. Clarke et al. also recommend use of hydraulics in lieu of on-site generators; the produced high pressure fluid could be stored and released as needed. By metering out the high pressure fluid during high peak demand periods, tidal current systems would no longer be functional during tidal flows only, flows which often do not mirror normal electrical demand periods [37].

Other researchers have been trying to extend the usefulness and practicality of tidal energy through use of computer modeling programs. One such program, Harmonic Analysis of Least Square, or HAMLS for short, is a robust program that accurately models tidal current flow over extended periods [38]. With such accurate predictions, power utilities can integrate tidal power seamlessly into the national grid. Instead of having surges of power in the grid, utilities can reduce output of neighboring plants to offset the gains from the tidal system during peak tidal flows [38].

When tidal systems with directly attached generators operate (Figure 13) consideration must be made for the influence of the generator upon the rotation of the turbine. Generators with significant loading cause opposing torques upon the turbine drive shaft; this in turn can make the system much less efficient at harnessing available power [39]. Programs which can modulate load levels depending on tidal current flow and turbine rotational speeds, such as Seif Eddine Ben Elghali et al.'s high-order sliding mode control program, greatly increase productivity of the turbines. Elghali et al.'s program relies on numerous sensors to determine the current operating conditions of the turbine, as well as the current tidal flow being experienced; the result is a program which modulates load levels so accurately that chatter and vibration effects upon the turbine blades are completely eliminated [39] [40].

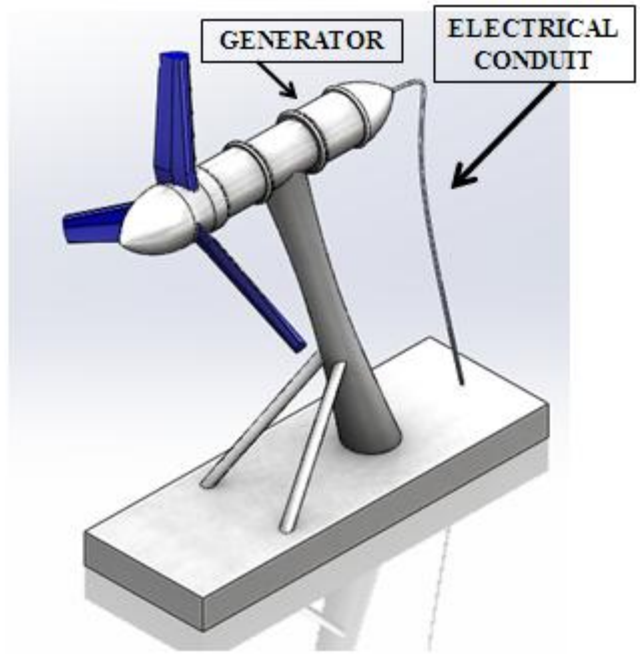


Figure 13: Basic tidal turbine assembly with attached generator.

Efficiency gains are also sought on the more macro level of turbine design and placement. Designing turbine blades which capture the maximum amount of kinetic energy from the water is key to a successful tidal current assembly. Dr. Alexander Gorban et al. have determined through mathematical modeling that the actual maximum efficiency of a two-dimensional turbine assembly (akin to a propeller) is only about 30% [41]. The markedly different helical turbine assembly has an increased efficiency of 35% and represents a better choice for harnessing kinetic energy of tidal flows [41].

Understanding the fluid flow phenomena, as it relates to the interaction of tidal currents and turbines, is an area where greater efficiency gains can be had. Elghali et al also produced a Matlab-Simulink computer model which accurately simulates the hydrokinetic turbine, the tidal flow, and the interaction between both [42]. Turbine blade pitch, speed, and diameter are all explored, as well as the flow rates, turbulence, and other disturbances that tidal flow can experience. The culmination of these variables helps to produce a more accurate representation of tidal power output for a unique scenario, and gives planners an opportunity to make a system as efficient as possible [42].

Placement of turbines also plays a significant role in how efficient a tidal power plant can be. Turbines which are placed too close together and too far across a channel, as L. Myers et al discovered, can act as a fluid block, thereby reducing flow rates and reducing the kinetic energy potential of the flow [43]. Through computer modeling, L. Myers et al discovered an optimal turbine array (Figure 14) orientation for the Race of Alderney in France [43]. Similarly, Chris Garrett et al. discovered through mathematical modeling that drag on supporting structures of turbines provides one of the most significant and underestimated drags on kinetic energy potential for a tidal fluid flow. It appears, therefore, that proper placement and sleek design is key to efficiently harnessing the kinetic energy from tidal flows [44].

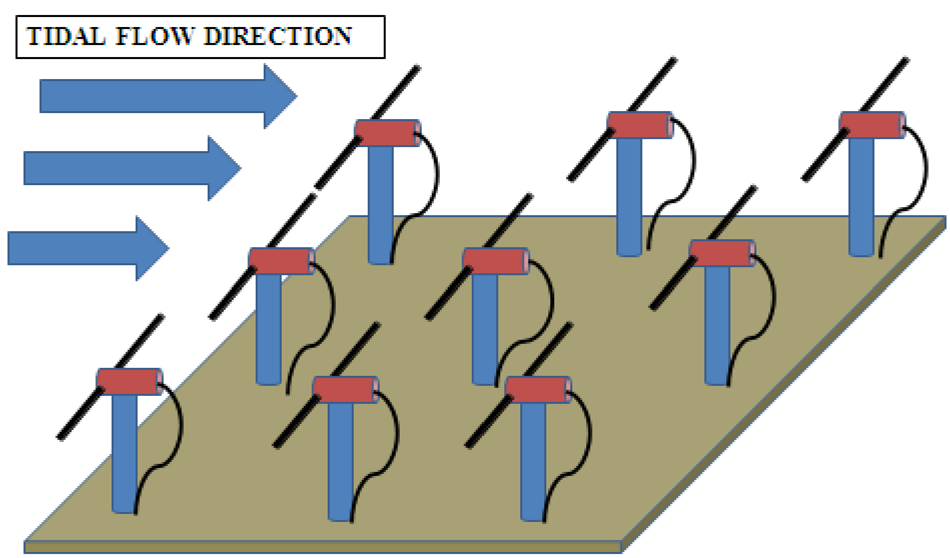


Figure 14: Basic tidal turbine array on ocean floor. Optimization of tidal turbine layout encourages

maximum kinetic energy gain by each turbine.

1.10. An Overview of Current Tidal Technologies

Tidal current systems all contain some form of kinetic energy capture system. From here, the design differences vary greatly. Some, like the Atlantis AR-Series unit (Figure 16) and the Hammerfest-Strom HS300/HS1000 units, contain a horizontal-axis, seafloor mounted turbine blade with a generator directly attached [45] [46]. As the turbine rotates due to tidal flow, electricity is generated. Tidal Energy's DeltaStream unit (Figure 15) modifies this concept by connecting three such turbine-generator units to a common platform in a triangular form. This arrangement allows for quicker installation in an ideal arrangement [47]. Most of these basic units are constructed such that the turbine-generator unit can swing 360 degrees, allowing the blade unit to face the tidal flow directly depending on direction and seasonal conditions [46].

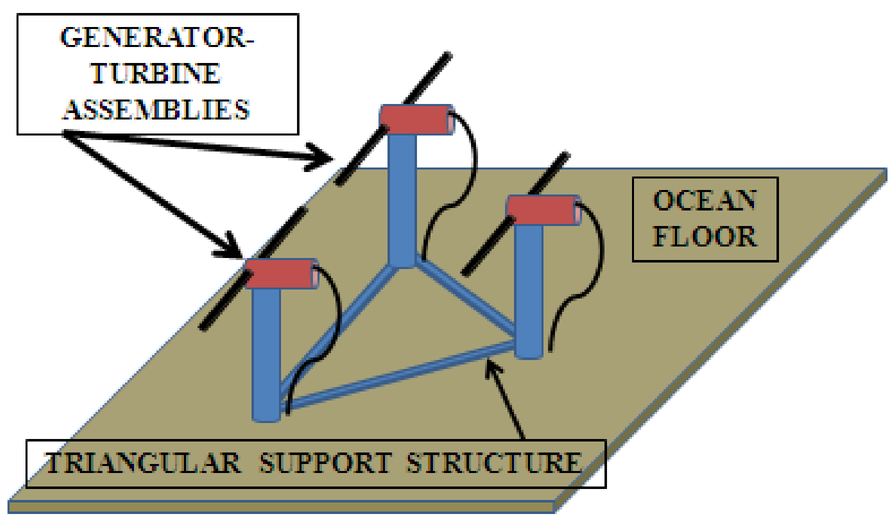


Figure 15: DeltaStream design orients three tidal turbines in a triangular form.

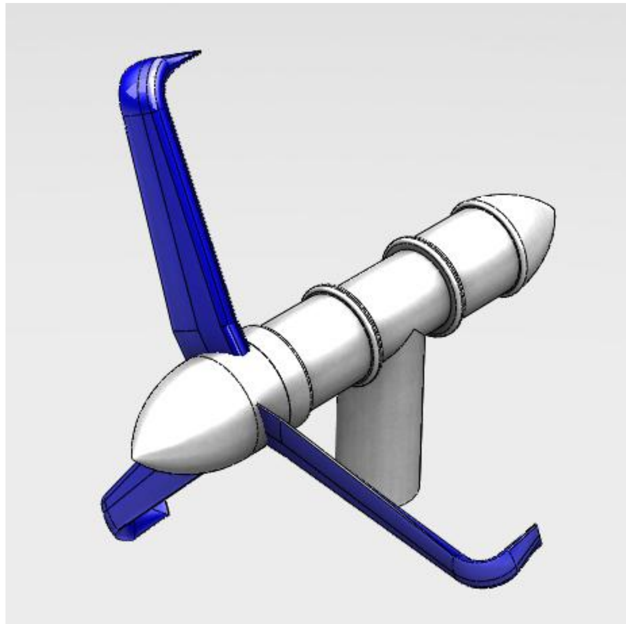


Figure 16: Atlantis AR-Series tidal turbine.

An additional variation on the basic turbine-generator design is the Marine Current Turbine's SeaGen system (Figure 17). This unit utilizes contra-rotational twin rotors (with a horizontal axis of rotation) that are mounted upon a cross beam, which is then attached to a vertical post [48]. The cross beam helps to maintain a common distance and orientation between the turbines, and keeps the individual turbine wake from negatively influencing the other turbine [48]. In order to adjust for flow in either direction, the turbines can rotate the pitch of their blades up to 180 degrees. In this manner, the unit is bi-directional [48].

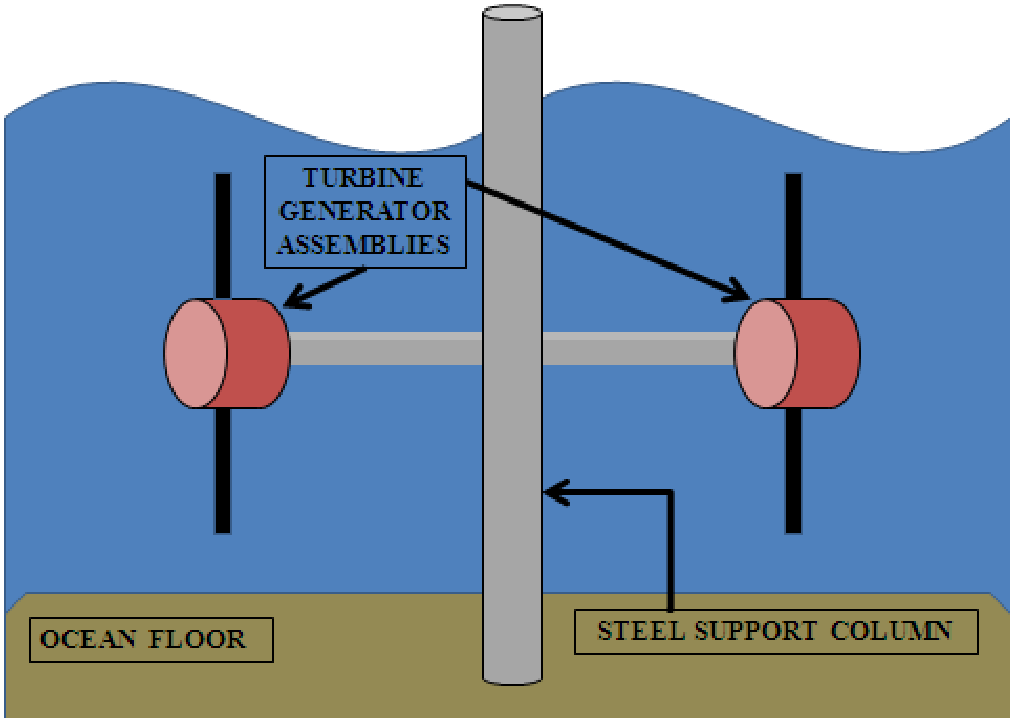


Figure 17: Marine Current Turbine's SeaGen system employs a steel support column for extra rigidity and ease of repair, as needed.

A further twist on this concept is SMD's dual turbine unit (Figure 18). The turbines contra-rotate (with respect to each other) upon a horizontal axis, and are mounted on a cross beam. The cross beam is buoyant and moored to the sea floor. This buoyancy allows the generator units to align themselves as needed to the direction of tidal flow without exterior assistance [49]. In general, moored and buoyant units are easier to lift from the water if repair or maintenance is required [49]. Additionally, these units are cheaper to install than the SeaGen unit. Instead of developing deep trenches in the ocean, shallower holes meeting a variety of seabed constructions can be used. Additionally, the columns of the SeaGen unit, while robust and stable, negatively impact kinetic energy and flow more than thinner mooring cables [49] [48]. Problems with mooring, including stability in compromised flows, can be overcome with the Ocean Flow Energy's Evopod, which uses vertical stability blades and a streamlined design to avoid interference (Figure

19) [50].

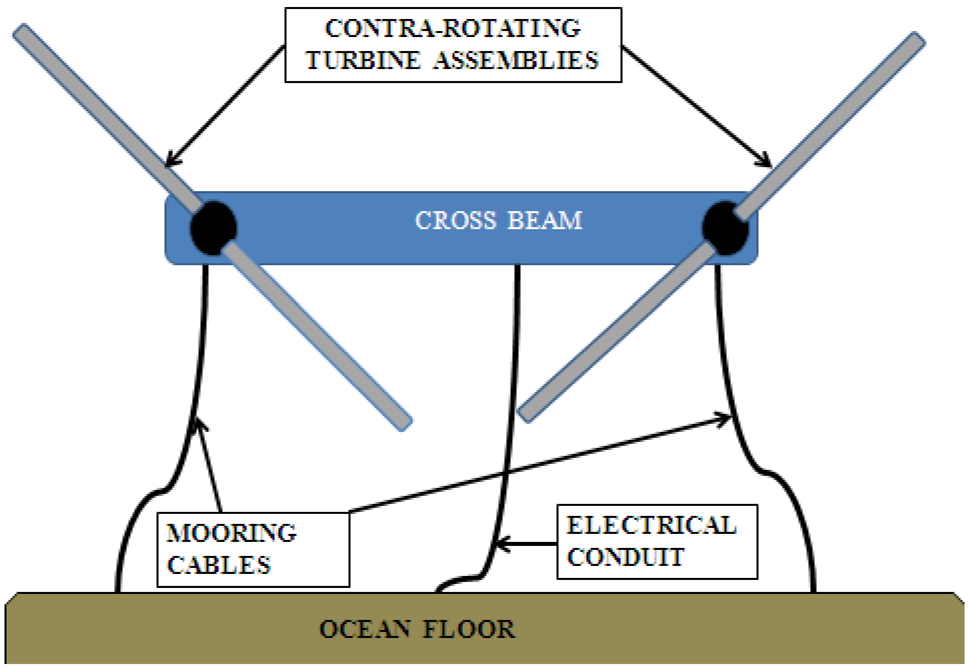


Figure 18: SMD's tandem turbine assembly uses mooring cables and a buoyant structure for flexibility in active oceanic environments.

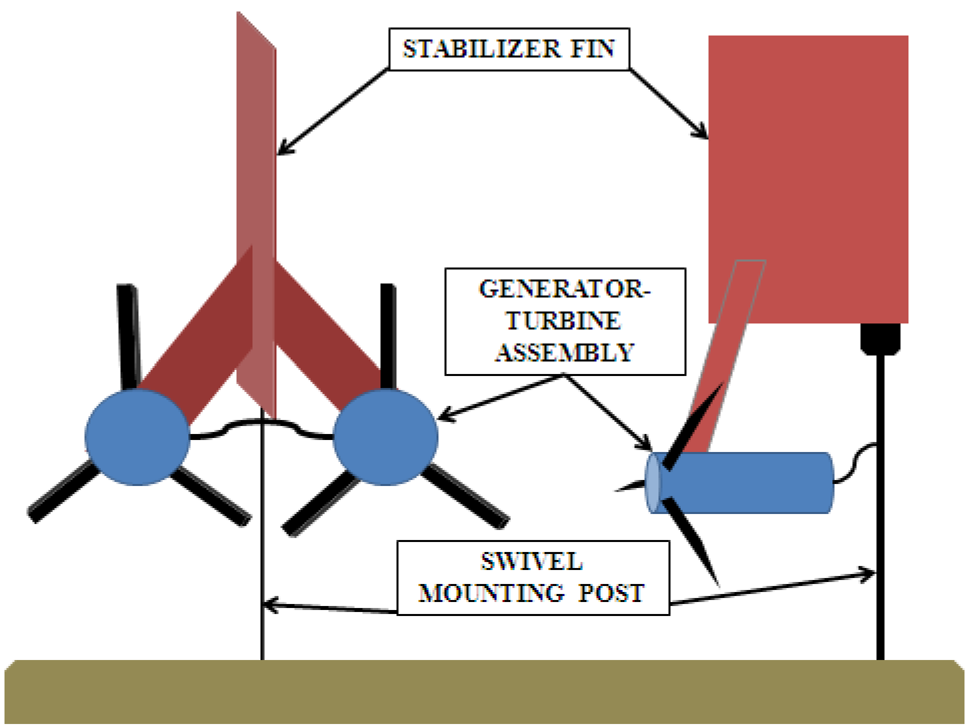


Figure 19: Ocean Flow Energy's EvoPod design includes a stabilizer fin to help keep the turbine

assembly oriented directly parallel to the ocean flows.

Fernando Ponta et al. in their paper "An Improved Vertical-Axis Water-Current Turbine Incorporating a Channeling Device" discovered that a water channeling device just before the face of a turbine helps to increase flow rates at the turbine by concentrating flow and increasing fluid velocity in turn [51]. The Atlantis AS-Series and Lunar Energy's turbines utilize a standard horizontal axis turbine with a large venturi duct surrounding the turbine (Figure 20); as fluid enters the duct, its velocity increases and more kinetic energy can be captured [46] [52].

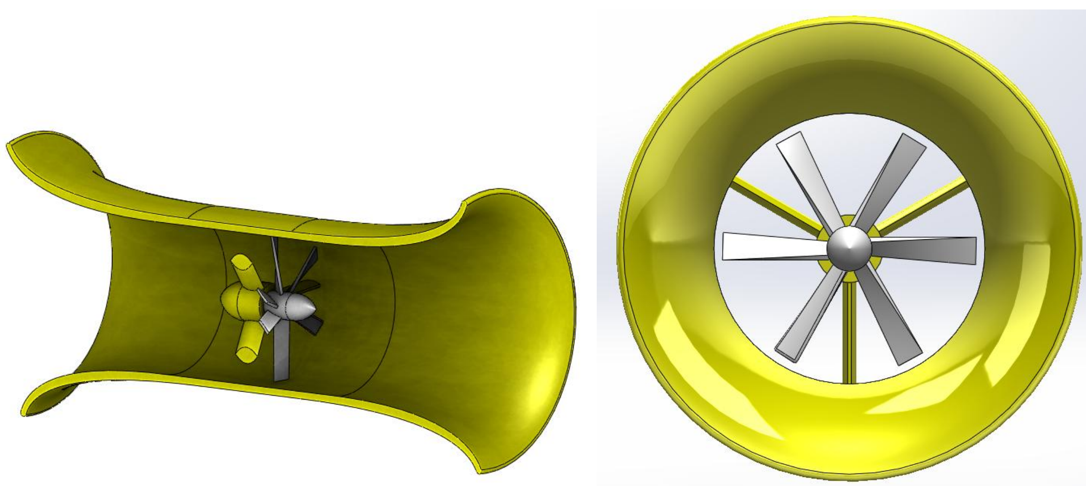


Figure 20: Venturi-duct tidal turbine channels flow. Shown here is the Atlantis AS-Series tidal turbine.

To date, vertical-axis turbines (Figure 21) have been rare in this field. Vertically oriented turbines allow for generation equipment to be out of the tidal flow path, increasing the harnessing capability of the unit [53]. Additionally, with generators mounted on top of the unit, repairs to the generators are easier to undertake [53]. Ye Li et al in their paper "Modelling of Twin-Turbine Systems with Vertical-Axis Tidal Current

Turbines: Part 1-Power Output" discovered that vertically oriented turbines can be bidirectional without complicated components to reverse turbine blades rotation [53]. Additionally, immediately side-by-side vertically oriented turbines can be up to two and a quarter times as efficient as a single horizontal axis turbine. Unfortunately, the side-by-side location means that double the turbulent action is experienced near the unit; this turbulence has a dramatically negative impact on efficiency [53].

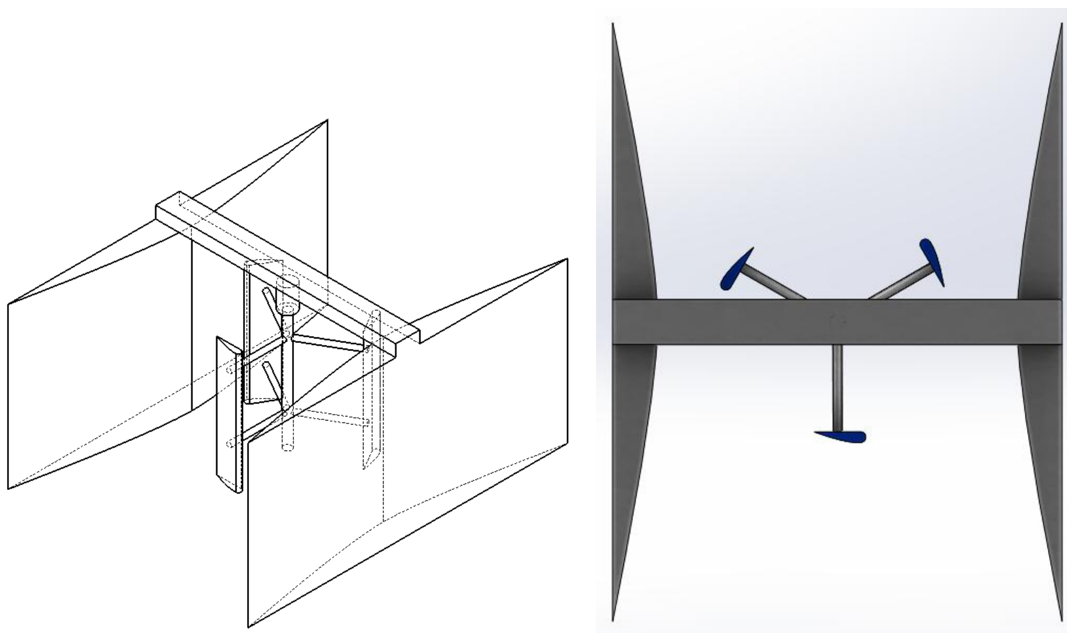


Figure 21: Vertical axis turbine sketch (left) with accompanying top-down view (right).

The paper also reviews horizontally oriented turbines, and observes that for *canard* orientations, or those wherein two turbines are mounted side by side at a distance of 1.5 times the diameter of the rotors, the wake effects upon the turbines was tremendous (Figure 22) [53]. In orientations where the blades rotated in the same direction, wakes from the individual turbines would build and negatively influence rotor speeds. When the blades contra-rotated, the wakes would cancel out and reduce negative rotor effects [53]. For orientations where the blades are stacked one behind the other and which rotate in the same direction (the rearward turbine being partially in the wake of the upward) the vortices of the co-axial blades act to build upon each other (Figure 22). This in turn causes the forward blade to discharge its fluid into a virtual vacuum effect; its blades spin considerably faster as a result. In the end, however, the contra-rotational canard system is more efficient than the co-axial design [53].

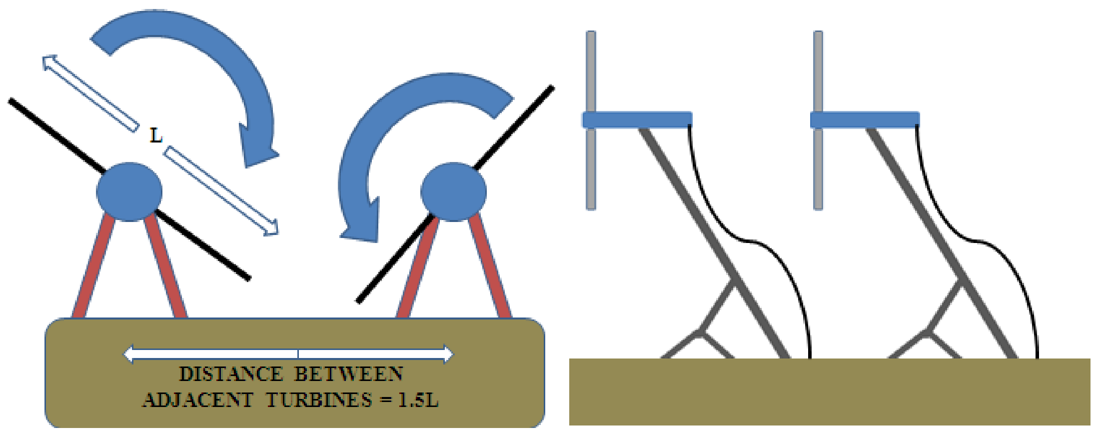


Figure 22: Canard orientation (left) is only more effective than coaxial (right) if blades contra-rotate, as shown.

The Darrieus turbine represents another type of blade orientation for harnessing kinetic energy. The Darrieus turbine is a cylinder-form with teardrop-shaped blades attached to its outer circumference (Figure 23). The cylinder is then mounted, long ways, onto a driveshaft connected to a generator. This orientation is only more efficient than plane turbines in certain configurations [54]. For example, starting torque is significantly reduced in a Darrieus turbine when four or more blades are used, making it initially more efficient at harnessing available kinetic energy; the high count of blades, however, also greatly increases internal turbulence [54]. The stronger internal turbulence slows down the turbine, and therefore greatly reduces efficiency. Although the most efficient Darrieus turbines (three blade orientations) only have a competitive advantage of 5% over plane turbines, the true advantage lies in their compact form [54]. Since a smaller Darrieus turbine produces the same power as a larger plane turbine, more Darrieus units can fit in a similar space, thereby making the macro system much more efficient [54].

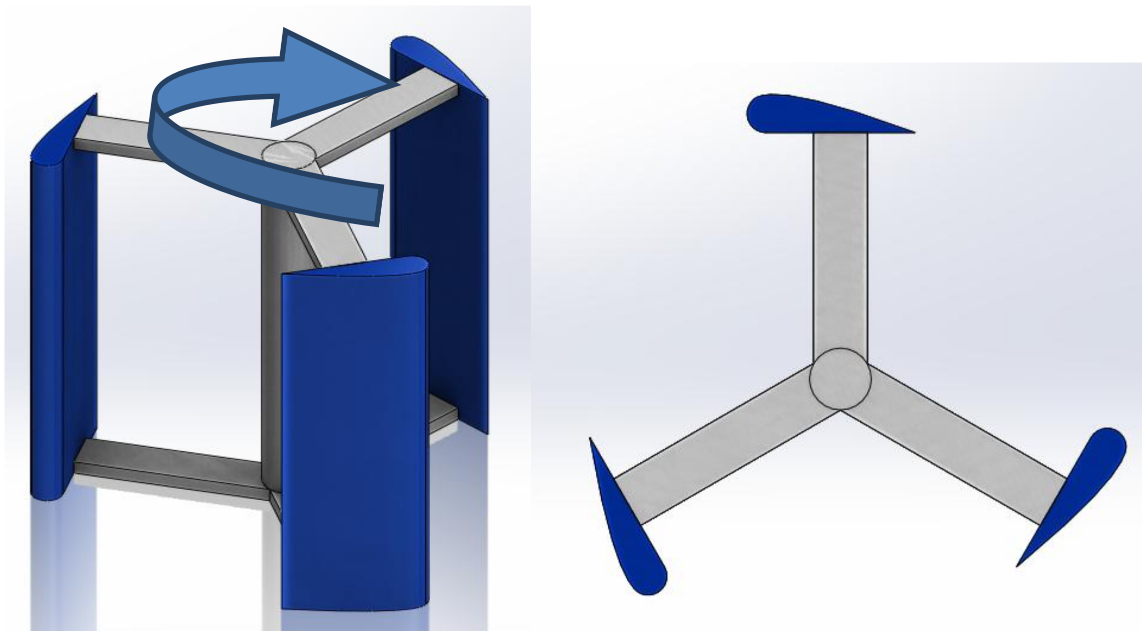


Figure 23: Darrieus turbine with three blades. Internal turbulence is the downfall of the Darrieus turbine.

Gorlov Helical Turbine group modified the Darrieus model for their hydrokinetic device. The design is such that three equally spaced blades, instead of being teardrop in form, are simply curved, flat metal (Figure 24) [55]. The curve follows the natural form of the cylinder, and provides a unit whose profile matches that of a cylinder. The main benefits of this design include rotation in one direction only regardless of fluid flow. The turbine also has low starting torque requirements; fluid need only be moving at 2 feet per second in order for the unit to rotate [55]. Akin to the Gorlov model is the Atlantisstrom turbine, which while based off the Darrieus model, deviates in its blade design (Figure 24)

[56]. Hinged but free to rotate 360 degrees, these five blades adjust position as the turbine rotates to not only capture kinetic energy, but allow as little encumbrance to fluid flow as possible [56]. Unlike the Gorlov model, which can spin quite briskly, the Atlantisstrom model does not rotate much faster than 7 rpm [56]. The Atlantisstrom model also reduces environmental impact; such slow rotational speeds do not pose a threat to sea life [56].

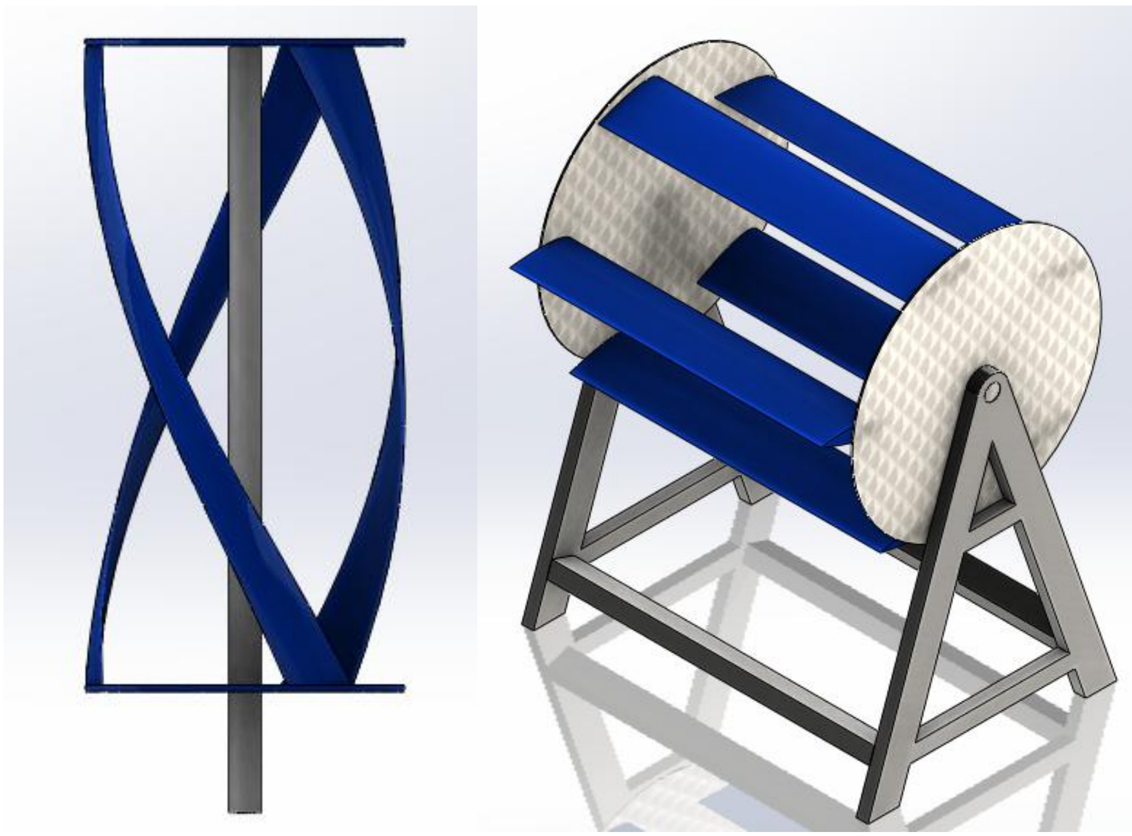


Figure 24: Gorlov helical turbine (left). Atlantisstrom turbine (right) spins much slower than the Gorlov style turbine.

Hydrofoils are the final major design group, a group which has no rotating components and poses little harm to sea life. PulseTidal makes buoyant units out of reinforced sheet metal (Figure 25) [57]. Their canted design, arranged parallel to the ocean floor, allows them to rise and fall as fluid passes over them [57]. Turbulent flows and locations near the ocean surface tend to be ideal for these hydrofoils, which depend on irregular downward forces to operate properly. As the foils rise and fall in these irregular flows, attached generators produce power. While these hydrofoils can produce power in shallower environments than plane rotors, their design requires not only turbulent flows but a bigger footprint than plane turbine models [57].

The Atlantis AN-Series turbines utilizes hydrofoils as well, but mounts them perpendicular to the ocean floor and attaches them to a chain guide (Figure 25) [46]. As the water passes over the hydrofoils, which are angled to ensure capture of kinetic energy, they cause the chain to turn. The chain is attached via sprockets to two generators, which produce electricity when turned and deliver it to land [46]. This model as well suffers from a large footprint, but instead of a large horizontal footprint, this model stands tall and wide, and lacks flexibility of design. With plane turbines, as many or as few turbines can be installed on the ocean floor as are needed. With the Atlantis AN-Series, a predetermined width is necessary for operation, a width which may impede ship traffic in certain narrow areas [46].

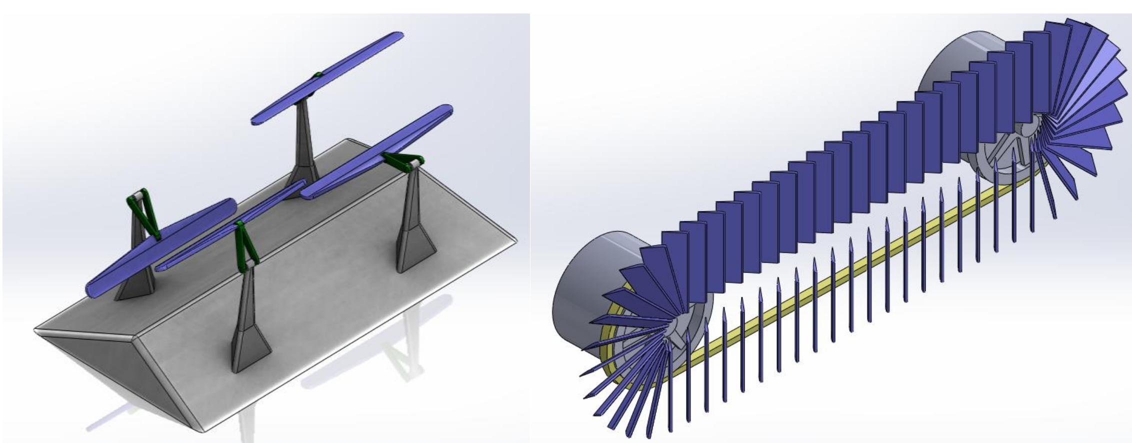


Figure 25: PulseTidal hydrofoil design bobs in the water, producing energy from the up-and-down motion. The AN-Series hydrofoil arrangement, conversely, rotates due to the kinetic energy in the

water flowing past its paddles.

1.10.1. The RITE Project

New York City, one of the most densely populated locations in the world, has begun to shift the way we think about urban living. Unlike cities such as Atlanta, Georgia, wherein almost 95% of all transportation is achieved via automobile, New York City provides the majority of its transportation via train, subway, and bus [58]. In a move to further solidify its “green” image, New York has just unveiled a tidal current power station at Roosevelt Island. Constructed by Verdant Power group, the Roosevelt Island Tidal Energy (RITE) project provides a discussion opportunity for locals and government officials alike as to the benefits of tidal current power. The system currently installed went live in 2006 on the East River near the coast of Roosevelt Island. Currently, only six turbines are actively generating power at a maximum of 175 kW. Five turbines output at 35 kW each, while the sixth turbine acts as a dynometer, providing up to the minute data on river conditions [59].

While meager by most accounts, the system capacity has the potential to expand to a hearty 10 MW if all 3,000 intended turbines are installed and brought on-line, as Verdant Power group, the head engineering and installation company, would like to see happen. At full capacity, the system has the potential to provide power to over 8,000 New York City homes. As of now, the system provides power to two locales: 1) The Roosevelt Island Operating Committee’s parking deck, and 2) Gristede’s supermarket, also on Roosevelt Island [59].

Chapter 2.

The Model Hydrokinetic Prototype

2.1. Initial Government Involvement

In response to concerns about future energy security, September 2010 saw the United States Department of Energy launch a series of investments totaling $37 million in Marine and Hydrokinetic technologies. Entitled *the DOE's Wind and Water Program,* this broad investment involved funding for twenty-seven unique corporations or institutions, each pursuing a different marine or hydrokinetic concept. Energy secretary Henry Chu proclaimed that "these innovative projects will help grow water power's contribution to America's clean energy economy" [60].

Benefits of further research in this area include better performance from hydrokinetic technologies, reduced cost of installation and maintenance, and accelerated full scale deployment of new marine and hydrokinetic power sources. The United States boasts abundant access to two oceans, one large gulf, and several strong rivers. Further utilization of these sources can help reduce reliance on carbon-intensive non-renewable resources. Additionally, increased energy security and stabilized energy costs are achievable benefits [61].

2.2. The Roles of Sunlight Photonics and Rutgers University

Sunlight Photonics is a private, venture-based company specializing in alternative energy research. Primarily focused in photovoltaics, Sunlight Photonics is also involved in marine and hydrokinetic energy technology [62]. Sunlight Photonics' research in tidal current systems proved a good match to the DOE's Wind and Water program, and herein a partnership was formed. Following DOE research guidelines, Sunlight Photonics sought out the assistance of Rutgers University's Dr. Kimberly Cook-Chennault for her expertise in hybrid and alternative energy.

The team developed to work on this project involves the cooperation and involvement of three main groups: Sunlight photonics, Rutgers University, and NASA's Jet Propulsion Laboratory. In Sunlight Photonics, three key individuals are involved: 1) Dr. Allan Bruce, Vice President of Sunlight Photonics and project manager, 2) Steven Lim, head project engineer, and 3) Jodi Maria Ciongoli, director of financial and intellectual property management for the project.

Rutgers University's role in this project involves assistance in conceptual development as well as deployment and operation of prototype. The key personnel are: 1) Dr. Kimberly Cook-Chennault, professor and lead research advisor for Rutgers University, overseeing all aspects of Rutgers' contribution to the project; 2) Sean DeGennaro, graduate student responsible for product research and physical project assembly and testing; 3) Jason Torres and David Specca, undergraduate Rutgers mechanical engineers who provided assistance to Sean DeGennaro and Dr. Cook-Chennault in preparation of and testing of the experiment as needed.

2.3. The Initial Concept

The initial concept was the brainchild of Jack Jones and Yi Chao, an engineer and scientist respectively at NASA's Jet Propulsion Laboratory. These two individuals were originally tasked with finding a new way to power underwater robotic vehicles. The main goal was to find a way that didn't use batteries (which require pulling the robots out of water to recharge them) but instead utilized thermal differences in the ocean to power them [5].

A key component in their project, a system which utilizes temperature differences to make high pressure fluids, gave Jones and Chao the idea that could power their robots. The high pressure fluid could rotate a generator, making electricity for the robot. Jones and Chao were inspired by this revelation, realizing the potentially broad scope of this project. It was readily apparent to the researchers that temperature differences in the ocean were not a very strong source of power for the world's energy needs, but the tides were. At this moment, the concept was born [5].

Jones and Yi proposed a concept wherein large turbines slowly rotate in the ocean from tidal flows, both during the transition to high tide, and the transition to low tide. These turbines would turn pumps which are attached to the turbine's main axle (Figure 26). As the pumps' driveshafts rotate, they pressurize fluid, thereby transferring the kinetic energy of tidal flow into kinetic energy of hydraulic fluid. The hydraulic fluid exists in a closed loop system which then flows to a main "high pressure" line. The high pressure fluid quickly moves to land, where a nearby electrical facility harnesses the kinetic energy of the pressurized fluid to generate electricity. At this point, the low pressure fluid would then be collected and returned via a “low pressure” hydraulic line to the individual pumps, which would again re-pressurize the fluid [5]. In this manner, the kinetic energy of the tides is transferred to the consumer through hydraulic fluid. The system design forgoes any electrical componentry below water like most current systems have. Additionally, high pressure holding tanks can delay or extend the electrical production cycle as desired.

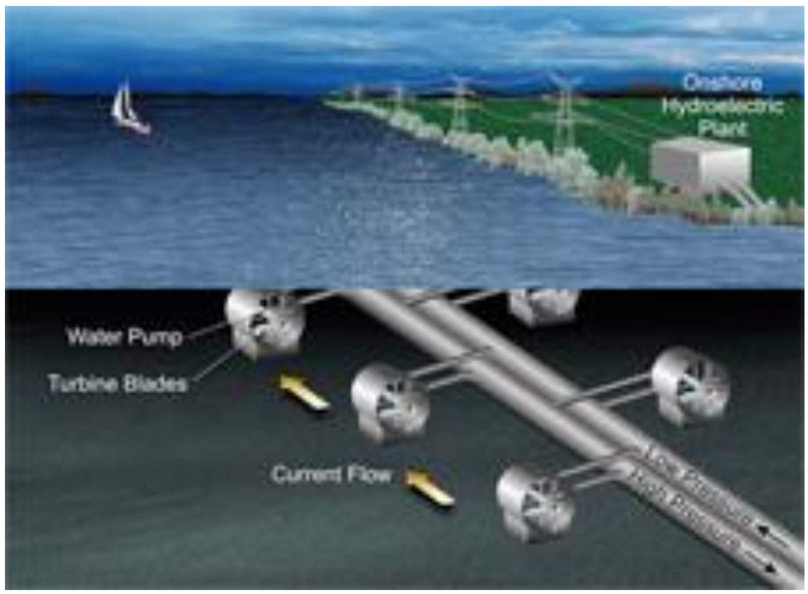


Figure 26: Artist’s rendering of full-scale system concept. Provided with permission by Sunlight Photonics Inc.

The artist’s rendering fails to highlight some of the more important aspects of the design, including piping construction, pumps used, and fluid compressed. These topics are discussed at length later in the thesis.

2.3.1. Comparison of Sunlights’ Design with Existing Systems

Current tidal hydrokinetic systems generate electricity at the site of the turbine, requiring generators and electrical cabling to be installed underwater. This individual attribute of new tidal-current systems makes them cost prohibitive, as electrical cabling must be buried below ground and extend long distances, increasing power losses along the way and posing a severe and deadly shock hazard should wires become damaged. Additionally, the highly corrosive and otherwise inhospitable maritime environment leads to short lifetimes for the electrical components, and “opens the door” to system failure down the line. These system failures have proven to be expensive in existing systems and difficult to repair due to the deep depths these systems work at. Trained SCUBA divers are required for any repairs or adjustments necessary, only causing costs to balloon further [5].

Jones and Chao also realized another excellent attribute to their design. Most tidal current and tidal range systems provide power only when the tides are rising or falling. This creates an almost sinusoidal pattern of power development, with distinct peaks and valleys. Normally, tidal current and tidal range systems can only be an addition to a national power grid system, as the tides do not naturally occur when peak power demands exist. The system that Jones and Chao developed calls for the potential addition of a high pressure holding tank. If desired, the system can produce an abundance of high pressure fluid (more than on-site motors can handle during the turbine rotation period) and hold it in a reservoir on-land for use when the turbines stop rotating. In this way, the intermittent effect of this system can be eliminated [5].

The major attributes of Sunlight Photonics’ design and the competition are summarized in Table 2. Properties like construction style, electricity generation and management, failure rates, maintenance costs, and environmental impact are explored. It becomes clear through comparison that Sunlight is superior to its competition in its lower failure rates (attributed to reduced underwater complexity) and significantly lower maintenance costs in the long term.

Table 2: The key differences between Sunlight Photonics' design and the major competition.

Attribute	Sunlight Photonics	Industry Standard
Construction Style	Pump and turbine assembly underwater; on-shore generation station.	Turbine and generator assembly underwater. No on-shore components.
Electricity Generation	On-shore generation station produces electricity from pressurized hydraulic fluid.	Immediate; electricity produced at turbine from turbine rotation.
Electricity Management	High pressure tanks can delay electricity production until later time. On-shore generation station can manually adjust electricity generation with human involvement, if necessary.	Computer programs can ensure efficient loading of generator. Otherwise, immediate production only.
Failure Rates (short term)	Low. Pump and hydraulic assemblies ideal for aquatic environment.	Low. Equipment initially sustains well in aquatic environment.
Failure Rates (long term)	Low. Hydraulic equipment is often used in subsea environments, and is designed to be corrosion	High. Corrosive aquatic environment causes failure of generator and electrical cabling; electrical

	resistant.	production was never intended to be done underwater.
Maintenance Costs	Low. Hydraulic equipment is long-life and commonly used subsea.	High. Expensive generators fail frequently in long term; require electricians who are SCUBA certified to initiate repairs at site, or arrange for equipment to be removed and reinstalled for repair.
Environmental Impact	Low. Biodegradable hydraulic fluid poses no danger to subsea ecosystems.	Low-Moderate. Exposed or damaged electrical equipment could pose shock hazard.

2.3.2. Design Drivers

In the battle to make renewables economically, Jones and Chao quickly realized that tough maritime environments call for simple, robust systems for maximum effectiveness, not overly sophisticated ones. In this section, the engineering characteristics that drive the system design are described.

The system calls for a pump which can operate at low RPMs. Currently, estimates place rotational speeds of underwater tidal turbines at about 10-100 RPM; a pump attached to this tidal turbine's driveshaft would be required to pressurize fluid at these rotational speeds. Initial estimates call for fluid to be pressurized to 500 PSI and then return on the low pressure side at about 15 PSI [63].

An environmentally friendly system is desired by DOE, and by all members of the team. Hence, use of water or another biodegradable hydraulic fluid is critical. In the unlikely event that the system should experience a pipe failure or be damaged, any fluid which leaks must not damage the delicate maritime ecosystem [63].

Additionally, the choice of pipes is a unique one. The pipes must be sustainable in a deep underwater environment, where they shall not fail due to high pressure (500 PSI), nor a corrosive environment, nor cold, nor due to vibrations or rotations from the system. A flexible, non-metal piping is preferred, one which can absorb shifts and jolts to the system that may occur during high-pressure fluid transfer [63].

The use of check-valves is a strong recommendation. Since an array of turbines will be used in the full-scale system, all individually contributing to one main high pressure and low-pressure line, the effects of each pump must be *isolated* [63]. A check valve would be required on the high pressure and low pressure lines that would effectively "shut off" the pump in case its pressure is lower than optimum. This would allow the individual pump to underperform without affecting the line pressure of the whole system.

The system has two other less obvious but very important considerations which must be made. Since tides cause water flow in two directions, the system should be designed such that flow can be harnessed in *both* directions. Of course, due to the small-scale nature of the prototype, ideal turbine design will not be explored at the prototype level. Instead, the turbine will be replaced with an electric motor and gearbox combination unit to simulate the effects of an operational tidal turbine unit. Finally, the system needs to be durable, requiring low wear characteristics. Thus, a long-life gear or centrifugal pump is a requirement, as well as avoidance of a gearbox or any other peripheral underwater equipment wherever possible [63].

Chapter 3.

The Prototype: Research and Construction

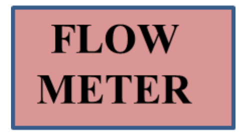
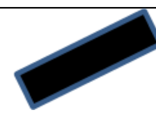
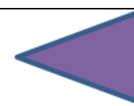
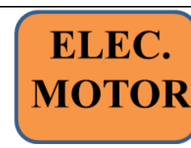
3.1. System Overview

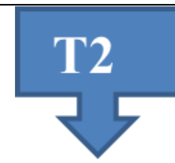
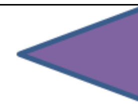
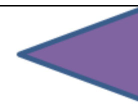
The hydrokinetic system is comprised of four distinct systems: 1) the high pressure hydraulic circuit; 2) the low pressure hydraulic circuit; 3) the mechanical to electrical power conversion and dissipation system; 4) the working hydraulic fluid. The hydraulic circuit (high and low pressure circuits combined) consists of all equipment (e.g. pumps, generators, motors, check valves, sensors, and hosing vessels) used to pressurize a working fluid and return the low pressure fluid to be pressurized again. The mechanical energy from the pressurized fluid is converted to electrical energy via a generator. The electrical power is then dissipated in a resistive load cell. The methodology for the selection of hydraulic fluid is provided, where efficiency, system durability, and life cycle are considered.

Figure 27 that includes both high and low pressure circuits is presented. In order to better understand the figure, Table 3 which precedes it provides a list of the components shown in Figure 27 and their labels and symbols on the graph.

Table 3: Figure key for comprehensive hydraulic circuit.

Component Name	Label in Figure 27	Symbol

Hagglunds Radial Piston Pump	HAGG. PUMP	
Axial Piston Motor	AXIAL MOTOR	
Hedlands Flow Meter	FLOW METER	
Pressure Sensor	Pj, where $1 \leq j \leq 4$	
Hose	Hj, where $1 \leq j \leq 4$	
One Way Check Valve	ONE-WAY C.V.	
20 Horsepower Electric Motor	ELEC. MOTOR	
Reduction Gearbox	GEARBOX	
Resistive Load Bank	RESISTIVE LOAD BANK	

Tee-Junction (hose fitting)	Tj, where 1≤ j ≤ 10	
Generator	GENERATOR	
55 Bar Check Valve	55 BAR C.V.	
10 Bar Check Valve	10 BAR C.V.	
Exhaust Hot Air	HOT AIR	

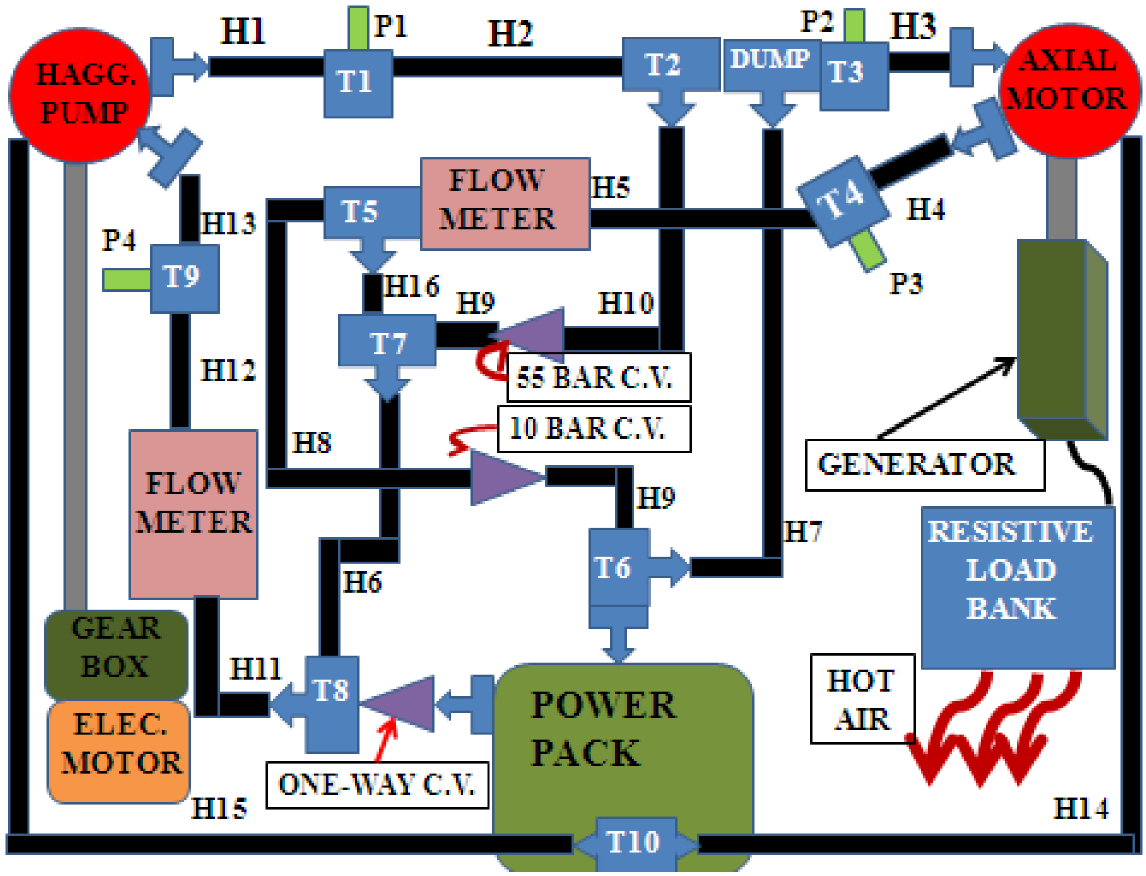


Figure 27: Comprehensive hydraulic circuit for the experimental prototype.

3.1.1. The High Pressure Circuit

In a full scale system, tidal flow is used to produce mechanical motion of a turbine, which in turn powers a pump that pressurizes the working fluid of the system (hydraulic fluid). Specifically, as the turbine rotates, it causes a pump (attached to the turbine's driveshaft) to pressurize hydraulic fluid, pushing it through hydraulic piping to an on-land generation station equipped with generators that harvest the energy from the pressurized fluid. In the prototype system, tidal flow is replaced with a Baldor-Reliance 20 horsepower electric motor, as shown in Figure 28.

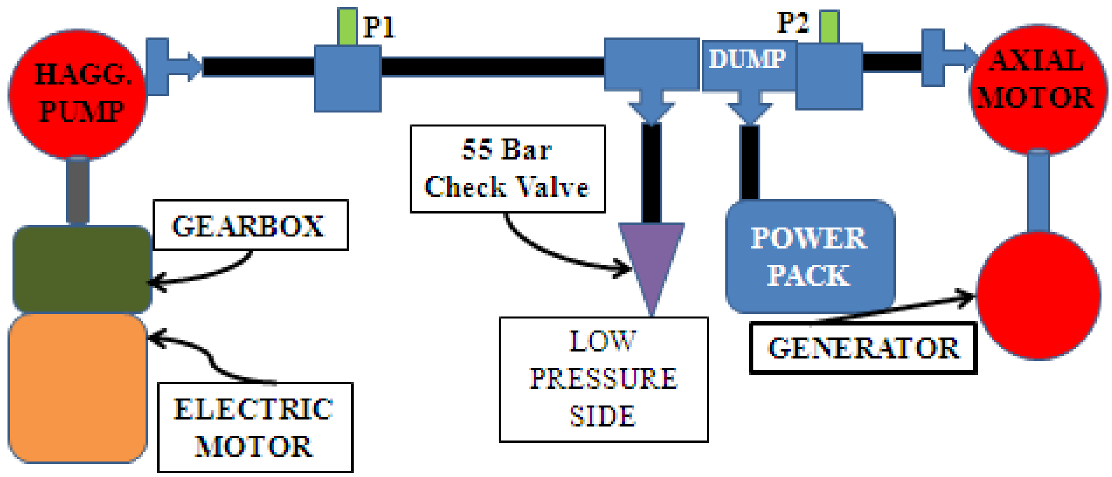


Figure 28: The high pressure hydraulic circuit.

This electric motor, when paired with our Quantis 14.63:1 gearbox, produces a maximum of 120 RPM at top output, as shown in Figure 28 [64] [65]. This RPM range closely matches the anticipated underwater turbine output power. The electric motor speed can be adjusted to model the beginning, middle, and end of a tidal current flow.

The output shaft of the gearbox is connected directly to a Hagglunds CA 50-32 radial piston pump. This pump, operating at 120 RPM, is capable of 64 gallons per minute of flow and near 900 PSI of pressure [66]. This pump was chosen for its scalability, as were the other major components of this system. Should the prototype function well, a similar but larger motor (preferably from the same company) would be used on the full-scale system. In the case of our experiment, the Hagglunds CA-50-32 model can be scaled up to a much larger unit, a unit whose pressures and flow rates would better match the power density of tidal flows. Matching the tidal flow power density on the prototype level would have required much larger componentry than would feasibly fit in a laboratory. An equivalent scale down of all components allows for an equivalent scale up on the full scale system. Therefore, if the prototype was scaled down to 1/10th of full capacity, the full-scale system would require a Hagglunds motor which is ten times larger.

At the beginning of the high pressure circuit (depicted in Figure 28) fluid is pressurized by the Hagglunds pump. This high pressure fluid then flows past two pressure sensors, P1 and P2. In addition, the circuit allows for diversion of fluid to the low pressure side through a 55 bar check valve, which will be discussed in detail later. A dump valve allows the high pressure circuit to be rapidly depressurized by allowing excess pressure to be released to the power pack reservoir. Provided the circuit is operating normally (dump valve closed and 55 bar check valve closed) the hydraulic fluid is sent directly to the axial piston motor to have the kinetic energy in the fluid harnessed. The motor's output shaft is turned by the high pressure fluid, which in turn drives a driveshaft to the generator, causing the generator to spin and electricity to be produced.

3.1.2. The Low Pressure Circuit

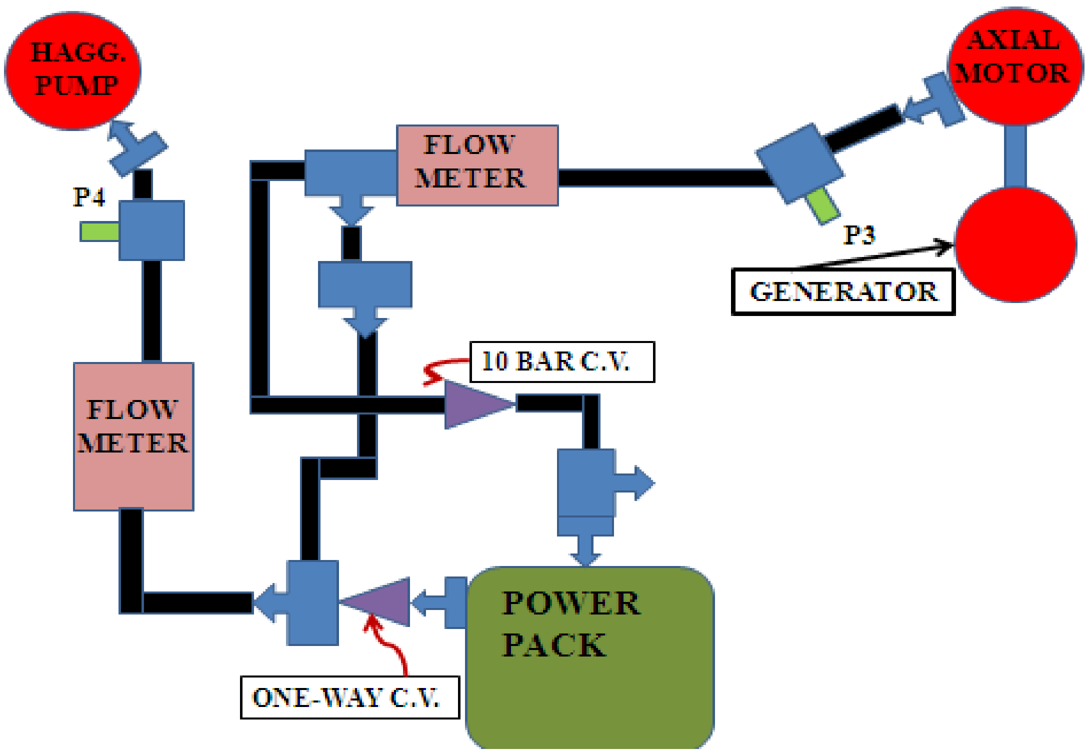


Figure 29: The low pressure circuit includes both the discharge of the axial piston motor and the return of the Hagglunds pump, as well as the power pack and several safety bypasses.

Operation of the low pressure circuit is more complicated than the high pressure circuit, namely because many of the pressure check valves are built into the low pressure side. Reduced pressure fluid starts the circuit at the discharge of the axial piston motor, as shown in Figure 29. Fluid flows past pressure sensor P3 and through the flow meter, where two possible flow paths are created. If fluid flows down, it flows directly to the output of the power pack, and then back to the Hagglunds pump inlet. This is the main flow direction. However, if the fluid pressure in the circuit exceeds 10 bar, it will flow to the left, circumventing into the power pack, where dumping into the reservoir helps to dissipate some of the kinetic energy and pressure. This maneuver represents the second safety valve in the system.

The final attribute to the low pressure side is the power pack booster. In addition to acting as a kinetic energy dump on the low pressure side during emergencies, the power pack has a ¾ horsepower motor attached to it, giving it the ability to physically boost fluid pressures on the output side of the pack. This boost helps to lift return pressures to the Hagglunds pump to the minimum *charge pressure*, or inlet pressure, mandated by Hagglunds of 29 psi. By nature of the design, a vacuum can develop in the inlet of the Hagglunds pump. The power pack helps to reduce that effect [67].

3.1.3. Energy Generation and Dissipation Circuit

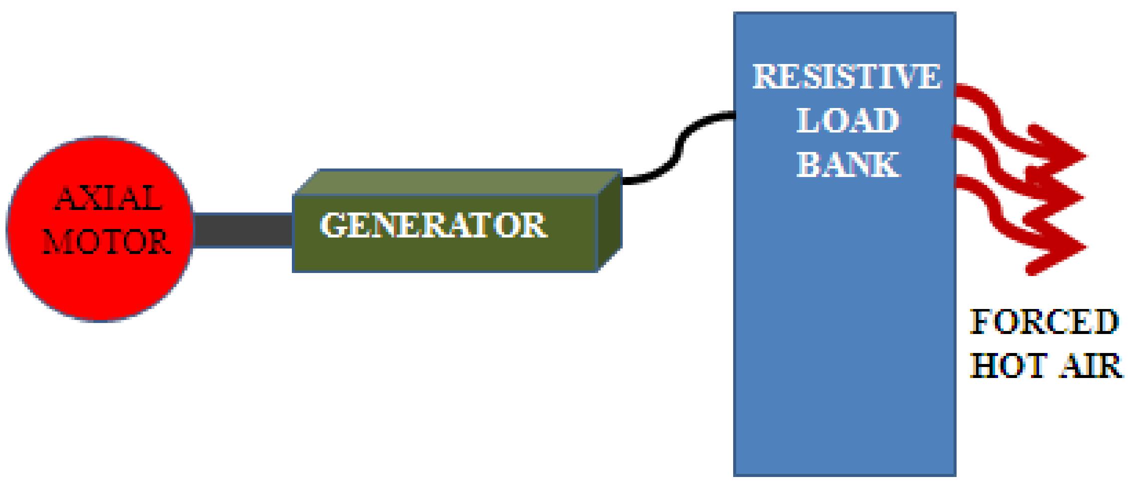


Figure 30: Electrical Generation Circuit

As shown in Figure 30, the electrical generation circuit is a relatively simple addition to the system on the surface, using the motion of the axial motor driveshaft to spin a generator, which in turn has its electricity dissipated by the load bank as hot air. In reality, however, the generator helps one to understand the impact of load on a hydraulic system and gives concrete values to the efficiency of the system.

On the macro level, the efficiency of this prototype depends on two variables:

electricity drawn from the grid to power the electric motor verses the electricity generated by the generator. Equation (8) shows the direct relationship:

$$Efficiency = \frac{Power\ Produced}{Power\ Consumed} \times 100 \tag{8}$$

From a fiscal standpoint, any hydrokinetic system needs to have very high efficiencies (greater than 70%) in order to be considered a worthwhile alternative to non-renewable energy sources. Low efficiency units (less than 70%) reduce the validity of the system as a whole.

Additionally, any functional system with a generator attached will experience resistance and "influence" from the generator itself depending on the load subject to the generator, yet another area of concern. This phenomenon occurs due to the relationship:

$$HP \propto S\,T \tag{9}$$

where HP is the horsepower (or load) of the motor in question, S is its shaft speed, and T is the torque of said shaft. As either torque or shaft speed rise, horsepower (or acceptable maximum load) will rise. Conversely, for a given load, reduced shaft speed requires each revolution to have greater torque per revolution and vice versa. However, if we consider that horsepower (load) is a form of power, then:

$$HP \propto I\,V \tag{10}$$

where I is the current produced and V is the voltage produced. This gives additional insight into the load dynamic. In equation (10) we see that if the load rises (either through increased voltage, current, or both) the revolutions on the shaft, the torque or both must rise to meet the higher load [68]. This resistance to motion of the generator may cause untoward effects upon the axial piston motor, and consequently the system itself. Therefore, it becomes imperative to assess the impact of a loaded electrical generator upon the system.

The electrical generator chosen, a Baldor-Reliance Inverter Duty AC unit, is a three phase, 460 volt, 25 horsepower generator capable of 26.6 amps at 1800 rpms [69]. Since the axial piston motor, which is directly attached to the generator, will be operating significantly slower than 1800 rpms, fewer amps will be delivered.

In order to properly operate the generator, however, a circuit must be attached to the output of the generator which will consume the electricity generated. The circuit attached is an Avtron K595 resistive load bank. This unit dissipates the electricity sent to it by heating coils and blowing off the heat [70]. Essentially a large space heater, the load bank provides the ability to run the circuit as needed without fear of damage to the generator.

3.2. Simulating Tidal Forces: The Baldor-Reliance Electric Motor

The model hydrokinetic system employs a variety of mechanisms to accurately represent the harnessing of power from the tides. From a radial piston pump to an axial piston motor, several components harmonize to create an effective hydraulic application of power from the tides. The system gets its initial power, not from an actual turbine rotating in a maritime tidal environment nor from a laboratory sized water tank, but rather from an electric motor.

The electric motor chosen for this project provides the torque and overall power deemed appropriate for a tidal system, based off of currently operational hydrokinetic systems operating in a maritime environment which are subject to tidal currents. It was chosen for its size, a size that accurately reflects the power capabilities of one turbine rotating under the influence of the tides. The motor chosen was a Baldor Reliance Super-E electric motor (Figure 31) capable of 20 horsepower at 1765 rpms. The motor draws 48 amps at 208 volts in three phase AC when starting. Operational amperage varies depending on desired output of the motor, but will never exceed the initial 48 amp draw [64].

Figure 31: Baldor-Reliance 20 HP electric motor.

The electric motor was chosen for its relative simplicity. The alternative design involved construction of a large-scale water tank, with a turbine mounted in the water and an artificial tidal environment generated. In this environment, the turbine would rotate, providing power and torque to the Hagglunds radial piston pump. The system would then more accurately represent a maritime environment but at a severely increased cost and complexity, a level which need not be approached at this early stage of development. Rather, the electric motor, under control of a sophisticated computer control module, provides accurate representation of the behavior of the turbine under tidal influence.

Control of the electric motor, to accurately simulate tidal conditions, is achieved through a Mitsubishi variable frequency inverter (Figure 32) a device which modulates and conditions incoming 208 volt three phase AC to generate the appropriate amperage for desired conditions. The inverter contains significant calibration and alteration abilities, including the ability to adjust the severity and duration of a cycle, the frequency at which the motor operates (0-120 Hz) and several other behaviors. The inverter is also capable of being controlled remotely. One can wire the inverter to a computer such that a program could control and guide the electric motor through a variety of operating conditions [71]. The end goal of the inverter is to provide a variety of operational scenarios for the motor to simulate. In this way, we can test the durability and behavior of the hydraulic system under a variety of conditions.

Figure 32: Variable frequency drive by Mitsubishi.

3.3. Hydraulic Pumps

3.3.1. Introduction

Over the course of prototype preparation and design, it became evident that a valid choice for pump and motor would need to satisfy several important system characteristics in order to be a good choice for the project. The ideal pump for the system needs to connect directly to the underwater turbine, harnessing tidal flows without use of a gearbox or additional efficiency stealing equipment; said pump therefore would need to operate in the range of 0-100 rpm, and produce at least 500 psi of fluid pressure in this revolution range. The accompanying motor, at the on-land power generation station, would need to be a smaller displacement unit which, when impacted with the high pressure fluid, would spin in excess of 1500 rpm without use of a gearbox or other modifications. The motor would be directly connected to a generator, and would therefore need to be optimized to the generator's rpm and torque requirements.

Both the pump and motor would need to fulfill these requirements while operating in their best efficiency point range, or BEP for short. A pump chosen which can meet the desired parameters, but is operating out of the BEP range is a poor choice for both the lifetime of the pump and the overall efficiency of the project. Finally, both pump and motor must be robust, long-lifetime units; frequent repairs or replacement of either unit will quickly erode the feasibility of hydraulic tidal current technology.

3.3.2. Centrifugal Pumps

The hallmark behavior of a centrifugal pump is the dramatic increase in flow rate of a fluid, with a much less dramatic increase in pressure. Centrifugal pumps utilize a spinning *impeller*, or rotational component which pulls fluid in and ejects it during the spinning cycle (see Figure 33). Typically, the impeller is shaped like a pedestal, with fluid pulled into the center and ejected out the sides. This sort of action only occurs when the impeller rotates. The faster the impeller rotates, the greater the *centrifugal* response upon the fluid, thereby creating a higher velocity. This centrifugal response has little impact on pressures, making it a poor choice for our prototype.

3.3.2.1. Centrifugal Pump Types

Centrifugal pumps, while having a very basic overall design, can be subcategorized into three major areas: standard centrifugal, multistage centrifugal, and regenerative turbine. The standard centrifugal pump contains a rounded body, housing an impeller (Figure 33). The impeller pulls fluid through one flange and pushes it out the other. In order to make the pump active, an electric motor or other device to provide rotational motion must be applied to the driveshaft that the impeller is attached to. Depending on the size of the application, this motor could be quite large [72].

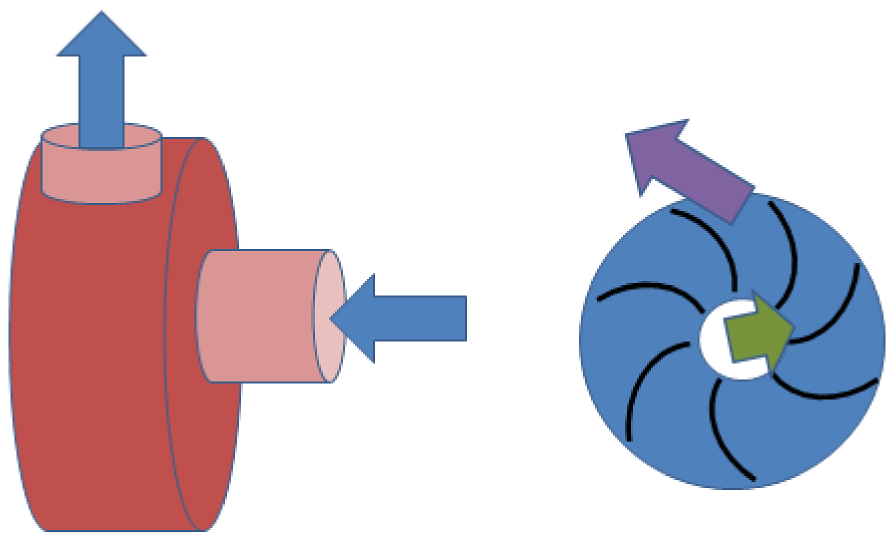


Figure 33: Standard centrifugal pump body (left) has a simple disk impeller inside (right) which pushes fluid from center to outer edge, thereby providing the centrifugal effect.

Multistage centrifugal pumps are conceptually similar to the standard centrifugal pump, except in the overall impeller design. Instead of one pedestal style impeller, multiple impellers are used, literally stacked upon each other on a single axis, as shown in Figure 34. This design orients the impellers along a common drive shaft and multiplies the centrifugal effect. The discharge from one impeller becomes the input for the next impeller. As the fluid travels through each impeller, it becomes faster and more highly pressurized than previously [73].

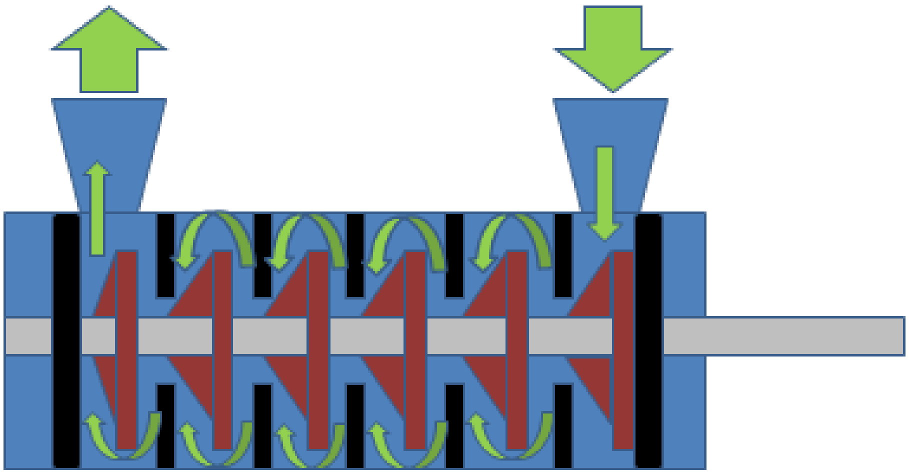


Figure 34: Multistage centrifugal pump forces fluid to sequentially pass through each impeller (red) before exiting pump body.

The regenerative turbine pump is marked by a sophisticated, somewhat delicate design. The regenerative turbine contains a reinforced outer shell encasing a single impeller, as shown in Figure 35. Unlike the standard centrifugal pump's impeller which has flanges on the center of the body and along the perimeter, this pump has both inlet and outlet flanges along the outer perimeter [74]. In fact, the core difference between the former two pumps and this model is that fluid is *doubly cycled* through this pump. Not only does the fluid circulate around the perimeter of the pump body, but it also circulates around the top of the impeller itself. Helical channels cut into the top of the impeller cause fluid to circulate in loops through the *inside* of the impeller [74]. This double cycling produces pressures that are the highest among similarly sized centrifugal pumps

[75].

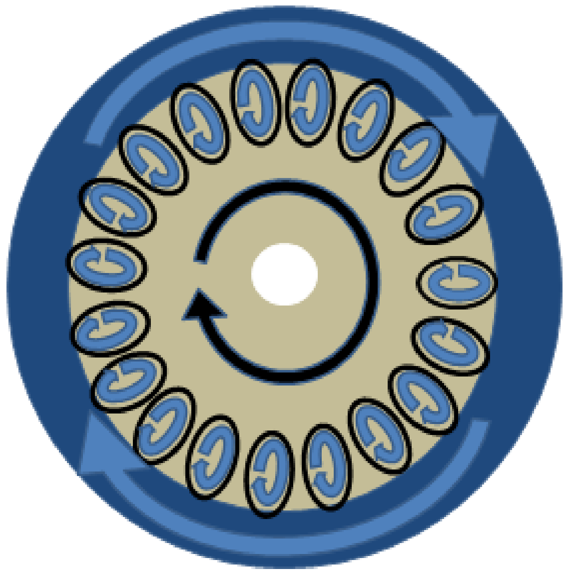


Figure 35: Regenerative turbine pump cycles fluid around entire pump body perimeter due to motion of impeller body (large blue arrows). Fluid also rotates through narrow helical cavities in impeller body giving the doubly cycled characteristic.

3.3.2.2. Centrifugal Pumps: Performance Analysis

As Table 4 shows, these pumps are operating in a revolutions range which does not meet the needed criterion for this project. In order for centrifugal pumps to be an integral part of this project, a gearbox would need to attach onto the turbine axle which would increase rotational speeds at a minimum ratio of 1:10. Besides this parasitic loss, centrifugal pumps, especially regenerative turbine units, create much lower pressures at similar sizes to the next class of pump, the positive displacement pump. Finally, standard and multistage centrifugal pumps need to be primed, a phenomenon wherein fluid must always exist inside the pump body in order for flow and pressure to be developed; air pockets would completely destroy the effectiveness of the pump and disrupt flow rates.

Table 4: Summarized operational ranges for centrifugal pumps.

Pump Name	Max Press. (PSI)	Flow Capacity (GPM)	Speed (RPM)	Efficiency	Maximum Solid Size	Sources
Centrifugal	3200	200,000	1000-6,000+	45%-80%	0.15”	[76] [77] [78] [79] [80] [81] [82]
Regenerative Turbine	520	200	1000-3500	30%-35%	None	[83] [78] [79] [84] [81]
Multi-stage Centrifugal	3200	10,000	3500-20,000	75%-85%	0.03”	[85] [86] [78] [87] [81]

3.3.3. Positive Displacement Pumps

Positive displacement pumps represent the second major category of pump technology currently available to consumers. Positive displacement pumps, through the use of reinforced steels and metal alloys, are more rigid and durable than centrifugal pumps, which often use plastics and thinner metals in their construction [75].

The core concept behind a positive displacement pump is its fixed displacement. At any given revolution, a certain amount of fluid will be expelled, no matter what the back pressure or restriction on the system [88]. Centrifugal pumps see a reduction in flow rate (culminating in a stall of flow) as back pressure rises, but the positive displacement pump sees no such reduction. Provided back pressure is strong enough, the positive displacement pump will simply stop rotating.

Positive displacement pumps typically use close tolerance metals to achieve fixed displacement behaviors and high pressures. Stainless and standard steels, metal alloys, and other rigid metals, joined with tolerances on the order of tenths of millimeters or less are typical in positive displacement construction, culminating in sturdy, precise and complicated designs. These metals have specific lubrication and temperature requirements to operate effectively [88]. In a positive displacement pump, it is common to find a *crankcase*, or body wherein mechanics separate from fluid flow operate, and the *pumping body*¸ the place where fluid is pressurized and circulated, as shown in Figure 36. The crankcase usually requires circulation of filtered, low pressure oil to maintain proper lubrication and to sufficiently cool the crankcase's internal components [89] [66].

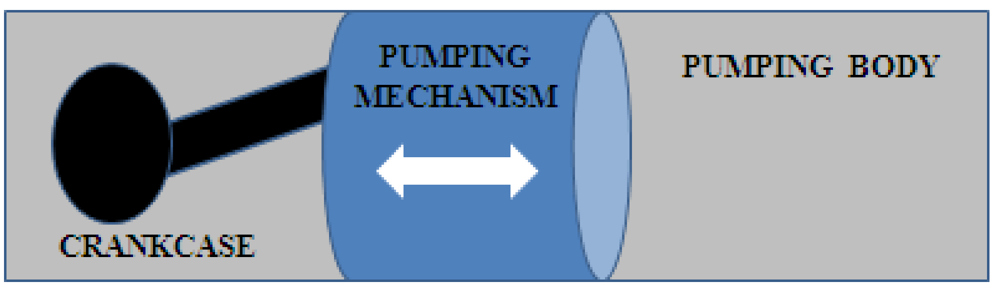


Figure 36: A typical piston pump, where the crankcase (left) is separated by the pumping body (right) via the piston itself.

The pumping body itself is also subject to the same heat gain and lubrication needs. However, since the pumping body's internal components are subject to fluid flow from the main hydraulic circuit, no additional cooling or lubrication circuitry may be necessary. The fluid used, and the quantity applied, is critical, however. Like with the crankcase, the pump body will have certain requirements for viscosity and filtration. The main hydraulic circuit will need to adhere to these conditions strictly, or severe damage to the positive displacement pump may result [90].

Filtration and cooling may be necessary for both the crankcase and the pump body under extreme or long term operation schemes. In both cases, sediment and metal debris accumulate over time. A proper filter helps to extract such contaminants without negatively affecting flow rates [90]. As oil heats up, its viscosity and composition may also be compromised. In order to prevent reduction of viscosity grade below acceptable limits, oil coolers may need to be added to the circuit in order to maintain a stable oil temperature during vigorous operation [89].

Despite these system complications, the positive displacement pump is considered to be far more capable than a centrifugal pump at equivalent revolutions. Where centrifugal pumps often require shaft rotations of *thousands* of revolutions for excellent flow rates and high pressures, positive displacement pumps needs only hundreds of revolutions to operate similarly [89]. Additionally, positive displacement pumps, due to their robust nature, are much more able to maintain system pressure than centrifugal pumps during periods of increased back pressure or restriction. In fact, safety check-valves must be installed to dump out excess system pressure should a blockage occur; positive displacement pumps have the potential to cause severe system damage if safety check-valves are not installed [89]. Also, because positive displacement pumps do not need to be primed, they can effectively manage variable flow rates as needed without a loss in continuity [89].

The remaining sections of this subsection, *positive displacement pumps*, are dedicated to the variety of pumps that were explored during initial research into positive displacement pumps. It is not exhaustive, but rather serves to show a variety of options for positive displacement pumps.

3.3.3.1. Progressive Cavity Pumps

The progressive cavity pump allows circulation and pressurization of fluids through use of a helical rotor which is mated to a helical stator. This stator is designed such that the rotor and stator do not fit together properly, as shown in Figure 37. Rather, gaps exist between both units, gaps in which fluid can travel. As the rotor turns, fluid shifts up the stator in a regular fashion [91].

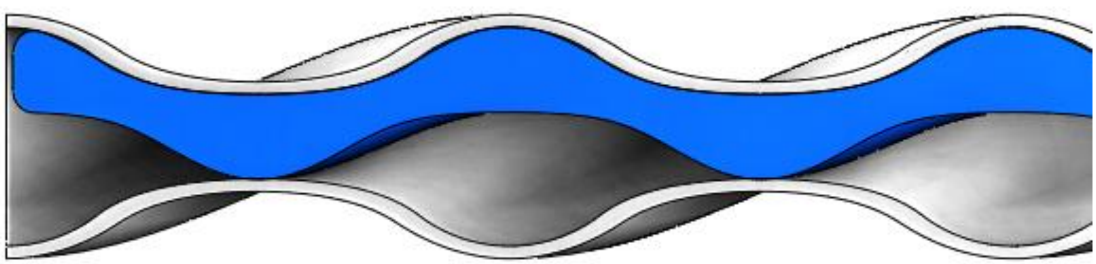


Figure 37: Progressive cavity pump. Fluid travels transversely as stator (blue) rotates.

Since these cavities are rather large, the progressive cavity pump can provide one key attribute that its competitors cannot: the ability to pump suspensions and solids with ease, a rare attribute among positive displacement pumps [75]. It achieves this though use of a strong outer metal casing which is significantly longer than it is wide. The pump is capable of pressures up to 2100 PSI, and flow rates in excess of 2200 GPM. Unlike many other positive displacement pumps, the progressive cavity pump often operates in excess of 1000 RPMs, and can be difficult to start due to its design [91].

Often called a “sticky start” pump, these pumps tend to bind or lock up when stopped for extended periods of time. During these periods, the circuit’s hydraulic oil falls away from the critical lubrication points, making the pump difficult to move or adjust. Weaker electric motors and engines may struggle to start this kind of pump, and consequently, need to be sized for this high-torque starting condition for effective use [91].

3.3.3.2. External Gear Pump

External gear pumps are the most common pump type seen in engines and other hydraulic lubrication environments. Utilizing a leader and a follower gear that rotate together, these pumps create flow and pressure by traversing the fluid around but not between the gears (Figure 38). The leader gear obtains momentum from an attached motor, which is transferred in kind to the follower gear. Both gears exhibit tight tolerances with respect to each other. The leader and follower gear each have bearings attached to their rotating shaft; this bearing is submerged in the circulated fluid [92].

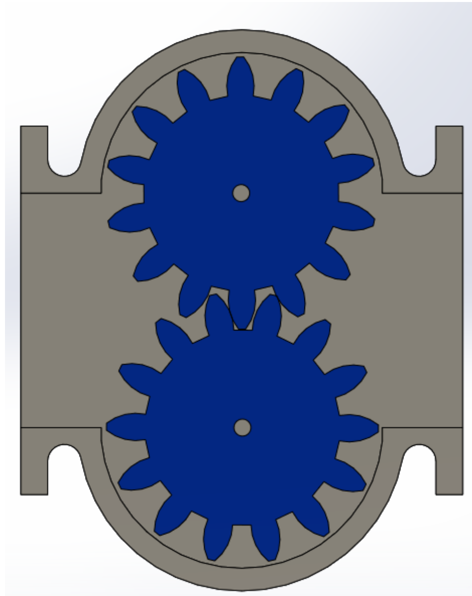


Figure 38: External gear pump. Fluid is compressed when it revolves around outer perimeter of gears.

As the fluid enters the pump body, it is pushed around the exterior of the gears by the rotation and pull of the gear teeth. The fluid is likewise compressed as it flows through the tight clearance between the gear teeth and the pump body. The meshed gears finally force the fluid out through the outlet port. Due to the tight gear tolerances, no fluid can pass between the gear teeth as they recombine in the middle of the pump body. Without these tight tolerances, pressure loss occurs because fluid can pass through the center mesh point of the gear teeth, thereby losing forward momentum and circulating continuously in the pump body [92].

Unlike some positive displacement pumps, no separate crankcase exists for the pump. This pump requires use of highly filtered and good quality fluid to prevent excess wear to the gears or bearings; without proper fluid, all mechanicals for this pump will effectively degrade. The benefit to a tight tolerance gear pump is the ability to precisely monitor and adjust fluid flow rates. Since the gears are interconnected (in the center of the pump body) there is no overhanging mass; the bearings, in turn, experience very little wear. The close gear tolerances reduce slack and noise, leading to a quiet, low vibration pump unit. The simple, robust design allows for high pressure and higher speed applications than for most positive displacement pumps [92]. Top rotational speeds meet or slightly exceed 3,000 RPM. Larger units revolve at a slower 500-1000 RPM [75].

In addition to lubrication work, external gear pumps find use in laboratory settings with various chemical mixing and blending, as well as chemical metering projects. Care must be taken to avoid caustic fluids or suspensions, as these can severely damage the pump. Longest life is achieved in hydraulic oil applications due to the continuous lubrication that occurs; external gear pumps often find use in industrial hydraulic applications, such as fork-lifts, log splitters, and other areas where high pressures are needed [92].

3.3.3.3. Internal Gear Pump

Internal gear pumps utilize technology similar to external gear pumps, but are oriented differently. Utilizing a follower and a leader gear, the internal gear pump is arranged such that the leader gear is the larger exterior gear while the follower gear is a smaller internal gear, as shown in Figure 39. The follower gear is “offset” and does not reside in the direct center of the pump body. Rather, the follower gear is arranged such that it sits near the entry port, and meshes its external teeth with the leader gear’s *internal grooves* [93]. In this respect, the internal gear is markedly different from the external gear pump, wherein two identical gears are used.

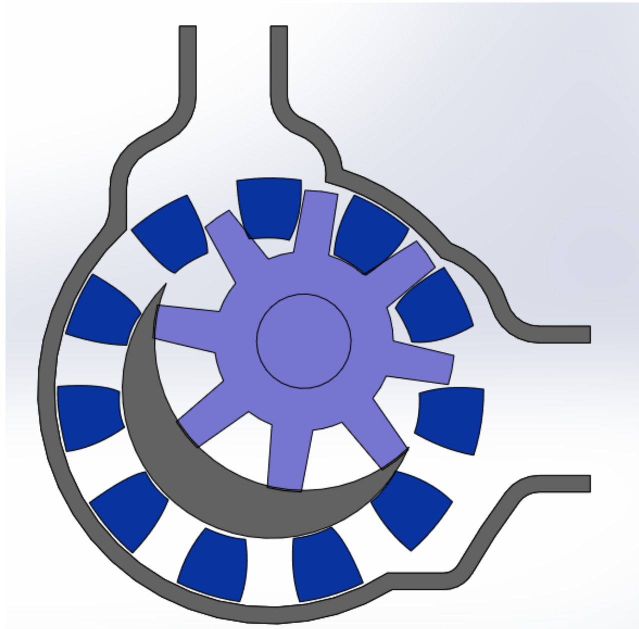


Figure 39: Internal gear pump. Note how the follower gear (light blue) is offset from the direct center of the pump body.

Unlike the external gear pump which forces fluid around the exterior of the intermeshing gears, the internal gear pump forces fluid in the zone *between* the two gears. The area between the exterior of the rotor (large, “exterior” gear) and the inside of the pump body accepts little fluid, since the tolerance in this zone is so tight. Due to the offset nature of the follower gear, ample space in the form of a crescent shape exists inside the rotor, surrounding the follower gear, wherein fluid can collect once it enters the pump body. As the leader gear and follower gear disengage (near the entrance port) fluid is pulled into the crescent. The fluid circulates under the influence of the rotating gears, working towards the exit port. At this zone, the gears re-mesh, causing compression of the fluid; this compression provides momentum and pressure to the fluid as it exits [93].

When the direction of the gears is reversed, the pump can provide the same experience *backwards.* Therefore, the pump is bi-directional and can serve a variety of purposes several other pumps cannot [93]. Also, larger internal gear pumps can accept a wide variety of fluids, including suspensions and solids. It is this unique quality which makes the internal gear pump perfect for pumping liquid foodstuffs (peanut butter, ketchup, corn syrup), macadam, paints and corrosives, soaps, and of course hydraulic oils. Since intermeshing teeth are combining in distinct stages (and not “all at once”) internal gear pumps tend to provide a non-pulsating discharge [75] [93].

Disadvantages of the internal gear pump design are significant, however, and need consideration. These pumps are not capable of the same high speeds as external gear pumps due to the constant engagement and disengagement of the different size gears, so only low speeds are allowed. Additionally, due to the offset nature, moderate to low pressures are required. The offset nature also means that the gears are *overhung* on their bearings, resulting in additional wear to the bearing set and as a result, a reduced lifespan. Additionally, due to design, usually one bearing (on the idler gear) is submerged in the fluid being pressurized; this results in additional wear and reduced lifetimes [93].

3.3.3.4. Multi-stage Screw Type Pump

The multistage screw pump, often called the multiphase screw pump, is one in which multiple screws are aligned in the pump body and rotate in concert to produce high pressure fluid. The screws are really long steel shafts with helical raised threads, as shown in Figure 40. Each shaft has threads oriented in a different direction, and when one lead screw turns, the other (or two) follower screws also turn, but in the opposite direction. It is this opposing motion which pulls in and pressurizes fluid, thereby causing the increase in pressure and velocity upon the fluid [94].

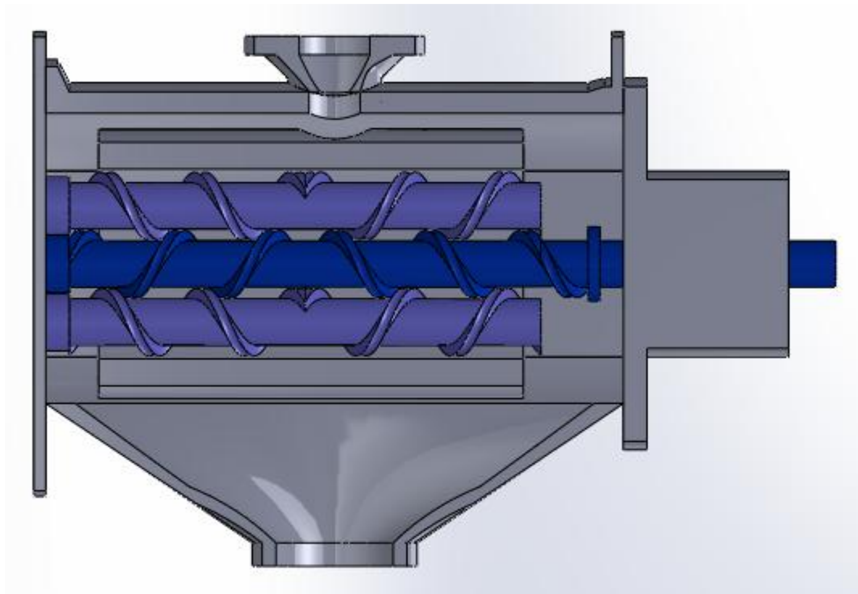


Figure 40: Multistage screw pump.

Since all the threads are constantly meshing and un-meshing, it becomes critical that the proper fluid is chosen for circulation. Suspensions are not allowed, nor are caustic or very thick fluids. Rather, lubricating and otherwise thin fluids are preferred, making this pump ideal for the hydraulics industry. Flow rates can be quite high, as can pressures, due to the very tight tolerances of the unit. Typically, pressures of over 300 bar and flow rates in excess of 3300 gallons per minute make this one of the most powerful positive displacement pumps [75]. While compromise must be made on circulated fluid choice, the benefits extend to the relative quietness, long life, and high reliability of the unit. Since the screws are always interconnected and supportive, pressure on bearings is low and overhung weight is nonexistent [94].

Multistage screw pumps are often used in cooling services, where high pressure coolant or lubricant is needed and reliability is a must. Multistage units find use also in crude oil pipeline services, as well as hydraulic applications for marine environments, such as for lifting and moving luggage and equipment on board ships [94].

3.3.3.5. Radial Piston Pump

One of the final pumps reviewed for this project was the radial piston pump. The pump which ultimately would drive our entire system, the radial piston pump proved to be an excellent choice for our scenario. With rotational speeds typically reaching a maximum of only 1200 rpm, the radial piston pump thrives in a slow speed environment. The radial piston pump has several variations of design, but all of which utilize pistons arranged radially around a center driveshaft. As the pistons harness high pressure fluid by pushing pistons down, they cause the shaft to rotate. Conversely, if the driveshaft is spun, the pistons compress fluid, providing greater pressure to the hydraulic circuit [95].

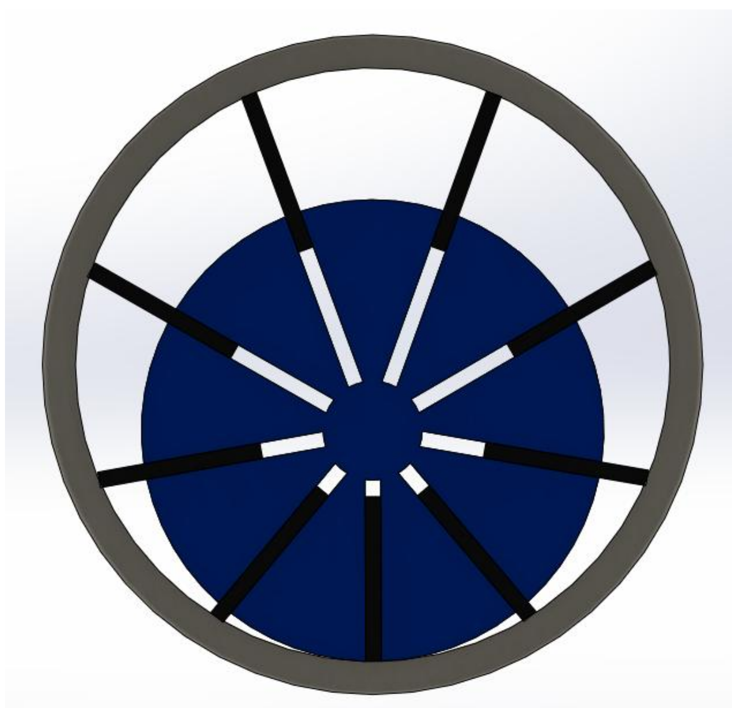


Figure 41: Radial piston pump with offset center shaft.

In one set-up, the pistons are arranged such that the inlet and outlet ports are near the outer ring of the pump body. The pistons accept fluid in this zone, and are acted upon near the center of the pump body, at the rotating shaft. The rotating shaft is offset from center (Figure 41) or has a cam lobe affixed to it such that one piston at a time is acted upon as the shaft rotates through a 360° full rotation. As the shaft comes in contact with the pistons, it pushes them up, causing them to expel fluid. As the shaft moves away, the piston falls toward center due to lack of pressure from the center shaft, thereby pulling in a quantity of fluid [95].

The second set up reverses the idea, having influence of the pistons upon the outermost area of the pump body, and fluid flow near the center. In this manner, pistons are connected to the center drive shaft in one large assembly, but the influenced "ends" of the pistons are exposed to the outer casing of the pump body. This outer casing can be irregularly shaped, helping to push on pistons individually [95].

3.3.3.6. Axial Piston Pump

The axial piston pump is one of several positive displacement pumps which has both a crankcase and a pump body. The pump has several key components: the pistons, swash plate, rotating shaft, piston shoes, barrel and barrel bores [96]. In the main pump body, a barrel or canister of six barrel bores exists, each barrel bore containing a piston (Figure 42). This barrel is directly connected to the drive shaft and rotates in tandem with it. The barrel is exposed to an inlet and outlet port of equal size, since the axial piston pump is a bi-directional unit and can be reversed as desired. The inlet port finds fluid pulled in by pistons which are pulling away from the top of the barrel, thus creating a vacuum. The outlet port finds fluid being pushed out because the pistons are pushing toward the top of the canister [97].

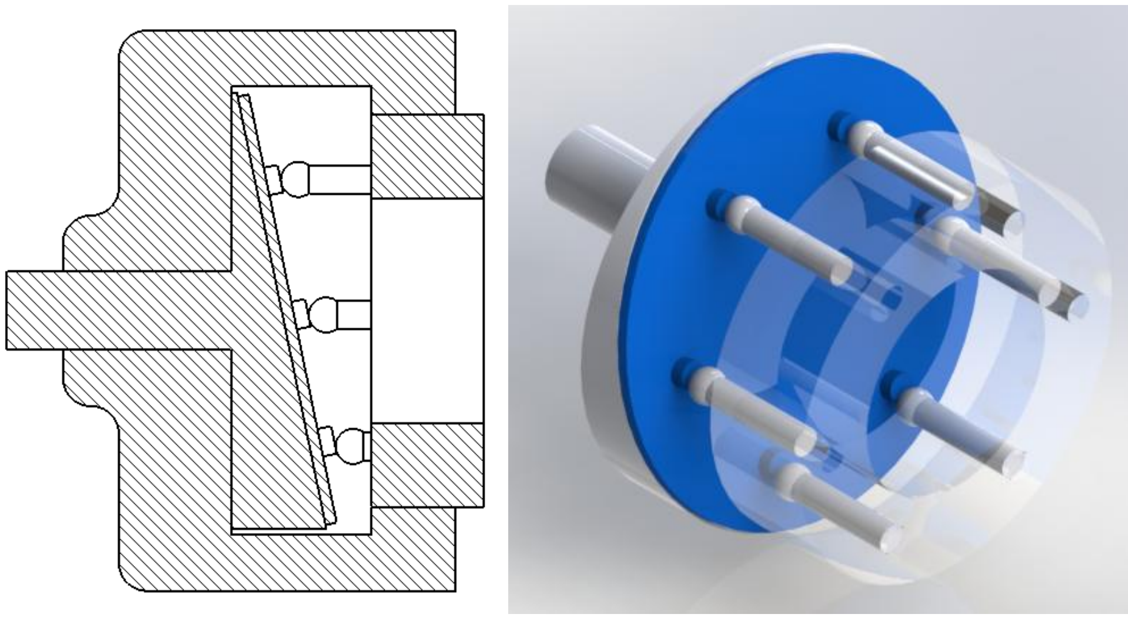


Figure 42: Axial piston motor has offset swash plate, causing pistons to pull in and out of bore as plate rotates.

The rise and fall motion of the pistons is made possible by the *swash plate*. This is a slanted plate to which the pistons are affixed via *piston shoes*, or rotating cuffs. The piston shoes allow the pistons to rotate at angles to the swash plate with ease. The swash plate itself is connected to the outer driveshaft via a rotatable cuff, similar to the piston shoes. This cuff allows the plate to maintain a fixed, slanted angle, while still rotating. This fixed slant is oriented such that the pistons become fully extended on the inlet side (allowing for near complete fill of the bore) while the outlet side has fully retracted pistons (allowing for near complete discharge of the bore). This arrangement means that as the pistons fill and the canister shifts from the input fluid force, rotation is achieved on the driveshaft [96].

The axial piston pump achieves these ends in a very compact form, but it has its difficulties. Achieving proper compression and smooth operation requires manufacture and design of a system with extremely tight tolerances and excellent sealing, all while operating at speeds in excess of 3000 rpm [97]. The pistons must maintain a good seal with the barrel bore. If too much *blow by* occurs, the pistons will lose hydraulic fluid before they can expel it. The barrel itself must maintain an excellent seal with the outer pump body; should leaks occur copious amounts of fluid would be lost to the crankcase, thereby reducing system pressure and causing unnecessary contamination. Like in other positive displacement pumps, the proper fluid is critical: use of caustic fluids or suspensions would be particularly harmful to the longevity of the axial piston pump. Well filtered, proper viscosity fluids are tantamount to a long life unit [97].

3.3.3.7. The Submersible Pump

In an effort to reduce complexity of the final system, and to maximize use of available off-the-shelf products, submersible pumps were considered as an alternative to the electric motor-pump assembly currently used on the prototype system. The prototype system, the thought went, would have a submersible pump sitting in a tank of hydraulic oil. The pump would provide high pressure fluid for use in the system circuit, to be harnessed as needed, and then would be dumped as low pressure fluid back in the tank. In this manner, the project would be simple, efficient, and compact in nature.

Submersible pumps do not produce high pressure fluid (greater than 500 PSI) as a hallmark behavior; therefore, these pumps would serve as a poor choice for our needs [75]. Submersible pumps are mainly used not to pressurize systems, but to provide dumping and draining capabilities. In these areas, submersible pumps excel, providing well over 7000 gallons per minute of flow rate among the largest units [75].

Submersible pumps are simply constructed and conceived. Utilizing a centrifugal pump mated to an electric motor, the pump pulls in fluid around its outer circumference and ejects it through the impeller center, usually due north. In this manner, the pump can pull fluid horizontally but eject it vertically, ideal for emptying large basins in an effective manner. Mesh cages usually surround the inlet of the pump body, helping to keep debris inflow to a minimum. The entire assembly is usually modeled after a column, and is significantly longer than wider. This narrow design makes these pumps significantly less efficient than standard centrifugal pumps [98].

The impeller in the pump body is propelled via an electric motor; if the pump is to lift fluid northwards, the electric motor must sit below the pump body to avoid restricting or interfering with fluid flow. Electrical wires wind down the outer casing of the entire submersible unit, posing a shock hazard or source of failure should the wires be damaged. The electric motor can be hermetically sealed, making it a dry motor. If the motor is exposed to the water, especially its bearings, the motor is called a wet unit. In either orientation, adequate circulation of cooler fluid near the pump and motor is necessary to prevent overheating [98].

Due to the nature of the pump, priming is necessary to keep the pump functional. By keeping the pump body fully submerged in fluid at all times, suction and discharge will continue unabated. Interruptions of available fluid will effectively terminate operation due to the design of the centrifugal pump. Owing to the ability to handle fluid suspensions, submersible pumps make excellent sump pumps and sewage pumps, finding use in a variety of household and sewage treatment applications [75].

3.3.3.8. The Concern about Cavitations

Cavitations are a phenomenon which must be fully understood when working with pumps. By their nature, cavitations can cause significant damage to pumps and hydraulic systems when allowed to continue for extended periods of time unabated [99].

A cavitation is a vacuum cavity which occurs as fluid molecules are pulled apart. This cavity only occurs until pressure raises enough for the fluid molecules to merge again. Upon merging, shockwaves occur, potentially causing damage to any surrounding solid materials. Cavitations occur near or at zones where dramatic pressure changes occur, typically at a pump body or other significant pressure changing location. When cavitations occur at a pump, a loud banging noise and heavy vibrations can result. Impellers, seals, and drive shafts can all be damaged. So extensive is the damage potential that industries have been devoted to the elimination of cavitations through proper sizing and scaling of hydraulic systems [99].

Cavitations typically occur when pumps are misaligned to the actual needs of a hydraulic system. In the case of centrifugal pumps, cavitations typical occur directly at the impeller due to blockages or reduced discharge rates. In this scenario, the pump outputs more fluid than the discharge route can accept. Therefore, fluid beings to “back up” and the impeller is unable to discharge as much fluid as it would like, given the current speed of its impeller. As the fluid begins to stall in the pump body (failing to discharge quickly enough) it heats up as a result of the increased friction. The more precisely the impeller is fashioned, and the tighter its tolerances to the pump wall, the greater the friction. As heat begins to build in the fluid, foaming potential increases. Foaming occurs as the fluid heats to its boiling point, and becomes gaseous. These gaseous bubbles can collapse suddenly if the fluid moves out of the hotter zone. As they collapse, shockwaves are sent out, and damage can occur [99].

The design of the positive displacement pump is such that the previous scenario is unlikely to ever occur. Because positive displacement pumps always output the same amount of fluid, they will simply blow through any system blockages that may occur or stall as a direct result of the blockages. However, an insufficient return pressure can indeed be a cause of cavitations for positive displacement pumps. Owing to the same displacement property, positive displacement pumps are likely to cause vacuums in the return line if insufficient fluid is available for the pump to utilize. The pump will actively collapse pipes when insufficient fluid is available, such is the strength of the positive displacement pump. During these low flow periods, the vacuum cavities created collapse when compressed by the positive displacement pump. This sudden collapse can cause surging in the pump or banging, knocking and other damage depending on the design of the pump [99].

In both scenarios, and for both pumps, the first and easiest remedy is to reduce pump speed until the banging or other symptoms of cavitations subside. For the centrifugal pumps, this allows discharge capacity of the pump to come back in line with maximum hydraulic line capacity. For the positive displacement pump, desired inlet flow rates will match available supply. In the long term, if insufficient flow rates or pressures are achievable *below* the cavitation threshold, a new hydraulic circuit may need to be designed. In many cases, cavitations signal an upper limit to system performance. To increase performance doesn't always mean increasing pump size; rather, reducing system restrictions and increasing hose diameter can have a dramatic effect on system capacity [99].

3.3.3.9. Positive Displacement Pump Overview

In Table 5, the important parameters considered in choosing the final pump and motor for the project are outlined. The maximum pressure characteristics of the pump, flow capacity, speed at which the pump optimally operates, and the maximum solid size, or size of suspended particles in the pumped fluid, are all considered in the decision making process.

Table 5: Performance statistics for various positive displacement pumps.

Name	Max. Press. (PSI)	Max. Flow Capacity (GPM)	Speed (rpm)	Sources
Progressive Cavity Pump	2100	2200	1000	[100] [101] [102]
External Gear Pump	4000	200	1750	[103] [104] [105]
Internal Gear Pump	600	240	400-1700	[106] [107] [108]
Multi-stage Screw Pump	5000	900	3000	[109] [110] [111]
Submersible Pump	200	2000	3450	[112]
Axial Piston Pump	5800	2100	8000	[113]
Radial Piston Pump	5000	1000	500	[114]

Additionally, Table 6 summarizes appropriate fluids for each type of pump:

Table 6: Viscosity limitations of various positive displacement pumps.

Name	Acceptable Fluids	Sources
Progressive Cavity Pump	No viscosity limitations; solids OK	[100] [101] [102]
External Gear Pump	Fluids as thick as 250,000 cSt. No solids permitted.	[103] [104] [105]
Internal Gear Pump	Fluids as thick as 2500 cSt. No solids permitted.	[106] [107] [108]
Multi-stage Screw Pump	Fluids as thick as 650 cSt. No solids permitted.	[109] [110] [111]
Submersible Pump	Fluids as thick as 100 cSt. Solids OK.	[75]
Axial Piston Pump	Fluids as thick as 1600 cSt. Solids not permitted.	[113] [75]
Radial Piston Pump	Fluids as thick as 2000 cSt; solids not permitted.	[114] [75]

3.3.3.10. Final Choice: Axial Piston Pump

The axial piston pump was chosen as one of the pumps in this system for its highest speed rating of all the fixed, positive displacement pumps, as shown in Table 5. The axial piston pump will actually be used in reverse, thereby classifying it as a motor. As the high pressure fluid in the high pressure circuit impinges upon the pistons of this motor, the output shaft will spin. Since the output shaft is directly connected to the generator, high rpms to satisfy efficient generator operation will be possible. Like the radial piston pump, this unit is capable of handling very high flow rates and pressures. These characteristics make it an excellent candidate not only for the prototype (where a small unit is used) but also for the full-scale system, wherein a very large unit capable of high flow rates can be used.

The final motor chosen for the prototype was a Bosch-Rexroth axial piston unit (Figure 43) displacing 0.042 gallons per revolution. This small displacement means that the axial piston motor is capable of very high rpms (over 1500) for our circuit.

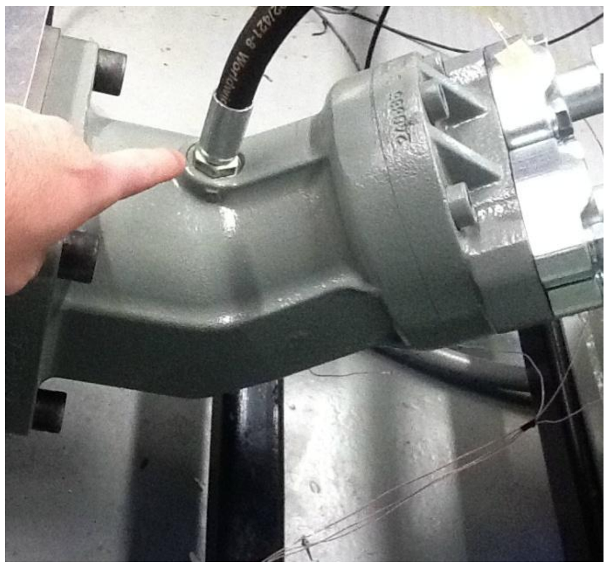


Figure 43: Bosch-rexroth axial piston motor.

3.3.3.11. Final Choice: Radial Piston Pump

The radial piston pump was chosen as the main pressure and flow producer in the high pressure hydraulic circuit in the prototype system. The radial piston pump is one of the slowest moving positive displacement pumps, as seen in Table 5, making it ideal for mating to an undersea turbine which only reaches maximum rotations of 100 rpm or less. Furthermore, the radial piston pump is capable of the highest pressures and is mid-pack on flow rates for positive displacement pumps surveyed. While both the axial piston motor and the radial piston pump are not capable of moving fluids with solids, they are capable of moving significantly viscous fluids, on the order of 1000 cSt. Since the hydraulic fluid used will be considerably less viscous (less than 100 cSt), this range is more than adequate.

Here, a Hagglunds CA-50 32 radial piston pump (Figure 44) was used for this project. It can output 63 gallons per minute at 120 rpm. This very large displacement at a low rotational speed is unique to the radial piston pump, and makes it a superior choice over several of the other kinds of positive displacement pumps in our application.

Figure 44: Hagglunds CA50-32 radial piston pump, top down view, installed in a sheet metal tank. Inlet and outlet ports are exposed on the top.

3.4. Gearboxes

3.4.1. Introduction

Gearboxes represent an efficiency loss for any system to which they are affixed;

the benefits of their use must therefore greatly outweigh this detriment. In the design of the prototype, builds with and without a gearbox were considered. Knowing that electric motors spin at speeds of thousands of revolutions per minute, the use of no gearbox meant choosing a pump which could safely operate in high rpm ranges. However, this concept was in conflict with the initial design proposal, which called for a slow moving pump, on the order of 10-100 rpm. Therefore, it was decided that a gearbox would be a necessary efficiency loss only for the prototype system. It would translate the power of the electric motor, which simulates the tidal force, to a rotational speed appropriate in tidal scenarios, 10-100 rpm.

The appropriate gearbox needed to:

- Be over 90% efficient, to avoid significant energy loss in the system.
- Offer a gear reduction appropriate for our electric motor
- Provide a compact, coaxial solution that keeps the electric motor and pump in-line.

3.4.2. The Different Types of Gears

In choosing the proper gearbox for this project, understanding which gear best suits the needs of the system was important. Outlined in Table 7 are the benefits and detriments of several types of gears considered for this project. The first, the spur gear, is a straight toothed gear, the most basic of designs, as shown in Figure 45 [115]. The second, the helical gear, also has teeth which protrude directly perpendicular to the face of the gear, but instead of being straight, are curved, as seen in Figure 46 [116]. The third type is the bevel gear, in which the tooth face is not perpendicular to the secondary face, but rather set at an angle [117]. This allows for perpendicular or otherwise non-coaxial shafts to be mated. The fourth kind is the worm gear, which is a spiral gear set onto a helical or spur gear such that neither axes shafts are perpendicular or coplanar [118].

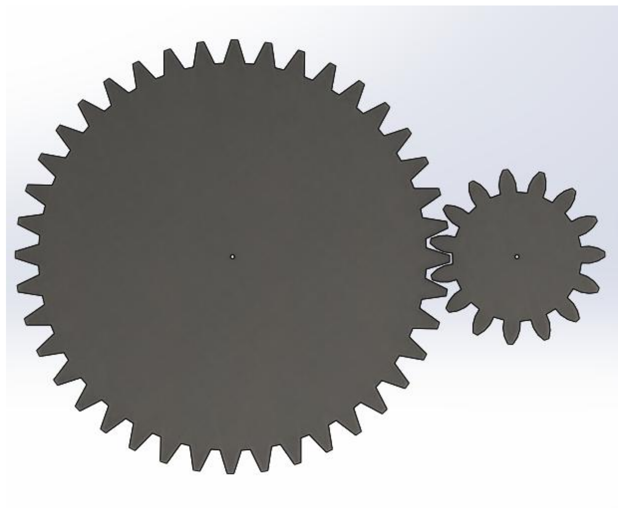


Figure 45: Spur gears utilize straight teeth, causing rough engagement and disengagement.

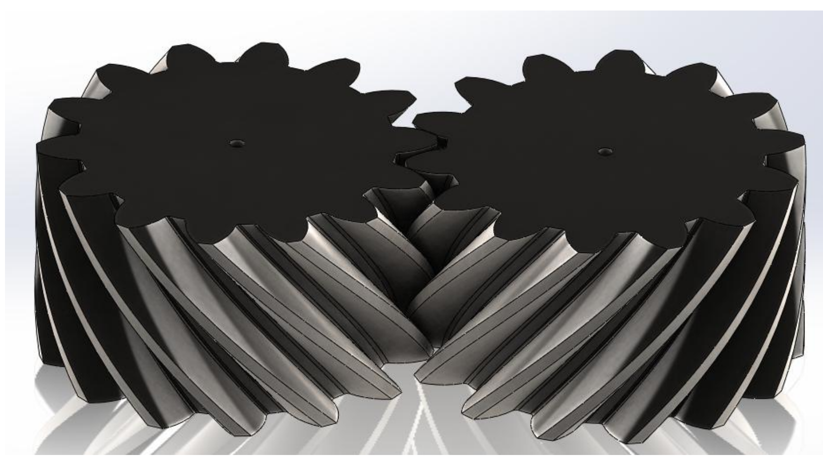


Figure 46: Helical gears use curved teeth for a smoother engagement.

The final major type of gearbox is the planetary set, in which three different groups of gears, called the sun, planet, and ring gears are all combined. By locking one group of gears (sun, ring or planet) while allowing the other two groups to rotate freely, different output effects can be had. For instance, locking the planetary gears allows the input and output shafts to rotate in reverse to each other [119]. This design may be advantageous to a full scale system looking to harness fluid flow both fore and aft without switching rotational direction of the pump.

The benefits and detriments of the various applicable gear types are outlined in Table 7 for direct comparison. It becomes clear, after review, that the helical gear, while costly, avoids much of the unnecessary complexity of the bevel and planetary gear units, yet also avoids the rough nature of the spur gear.

Table 7: Advantages and disadvantages for different gear classes acceptable for this prototype.

Gear Name	Advantages	Disadvantages
Spur Gears [115]	Very basic and commonplace. Good for parallel drive shafts. Good for low speed applications.	Gears abruptly engage, causing whining and vibration. Fastest wearing gear due to rough engagement.
Helical Gear [116]	Good for parallel and perpendicular shafts. Curved tooth makes for smooth, quiet engagement. Excellent for high speed applications.	Very high cost compared to spur gears, necessitating cost evaluation before implementation.

Bevel Gear [117]	Allows for coplanar driveshaft axes. Can set driveshaft axes at a variety of angles depending on bevel angle (0 to 90 degrees). Can employ helical design method to quiet gear engagement.	Expensive to manufacture. Drive shafts must be coplanar. Non-coplanar drive shafts excessively wear with bevel gear engagement.
Worm Gear [118]	Only the worm gear can move the larger spur gear; therefore, can act as braking mechanism.	High gear reductions only (20:1, 300:1, etc.). Non-coplanar, perpendicular drive shafts only.
Planetary Gear [119]	Multi-ratio gearing available. Reverse gear available.	Extremely complex. Often requires exterior control and command center to operate properly. Expensive.

3.4.3. Final Choice: The Quantis Helical Gearbox

Our project utilizes a gearbox with a 14.63:1 reduction to allow the rotational speed of the electric motor to come in line with real-world rotational expectations during operation (Figure 47). This reduction provides that the driveshaft of the Hagglunds pump will never exceed 120 RPM during normal electric motor operation (0 to 1765 rpm at its output). The Quantis unit uses helical gears arranged in an ellipsoid fashion; this design creates a gear which “eliminates tooth wearing and assures meshing in the strongest tooth area." [65] Such a design keeps efficiency high as well; losses are only expected to be 2-3% from this design [65].

Figure 47: Quantis helical 14.63:1 reduction gearbox.

Losses from the gearbox were a major concern from the beginning of development. Initial designs sought to eliminate the gearbox, not only for the efficiency losses, but for the potential inertial gain added to the system. Any addition to the circuit which makes it harder for the full-scale turbines to harness tidal energy would result in less power generated. A gearbox has the potential to make it harder for the turbine to spin; in such instances, only dramatic fluid velocities would cause rotational motion.

The Quantis gearbox used does not contain reverse gear capacities [65]. From a cost stand-point, the need for a reverse equipped gearbox is, in some senses, unclear. Many highly successful tidal systems have relied upon singular direction operation. In fact, having a reversible system often introduces complexities and losses that mitigate the advantage initially gained from harnessing in both flow directions. In the case of the hydraulic system, a gearbox with a dedicated reverse gear greatly increases costs, weight, complexity, and the potential for failure. In addition, perfectly timed “shifts” must occur, and be monitored. Gearboxes which engage a reverse gear often are unsynchronized, meaning that engagement can only occur when the gearbox comes to a complete stop. Engagement at any other point causes grinding and premature failure of internal components. The actuators and electronic components needed to operate such a system may significantly reduce reliability of the system in the long-term.

3.5. Hydraulic Fluids

3.5.1. Introduction

Hydraulic fluid represents, next to the physical equipment, the single most important component to analyze and understand during this project. Hydraulic fluid has the ability to affect and be affected by nearly every component of this project. The fluid itself affects lifetimes of components since in one role it is a lubricant. It affects potential system pressures, since some oils are subject to foaming, boiling, and compression in the pressure zone we wish to operate in. In turn, the fluid itself is affected by system temperatures and efficiencies; the most ideal fluid, under the worst efficiencies, will quickly become the poorest choice as it begins to degrade.

The first step in choosing a hydraulic fluid was understanding the role the fluid would play in our experiment. The critical criteria for the hydraulic fluid include:

- Biodegradability, with no appreciable impact on the environment should a leak occur
- High density, to efficiently transfer energy during pressurization
- Stable performance in the operational range of 35-200°F, both for laboratory and maritime temperatures, wherein the fluid should:
    - Experience no difference in compressibility
    - Not oxidize, degrade, or fail prematurely
- ISO 68 equivalent-grade viscosity, as per pump and motor requirements.
- Long operational lifetime (greater than 1000 hours between changes)

The major classification of fluids acceptable for this project, outlined in Table 8, include petroleum-based, biodegradable, and water-based hydraulic fluids. The table outlines various major characteristics of the fluids, including compressibility, temperature based behavior, cost and lifecycle assessment. Due to the first requirement, petroleum based fluids must be stricken from the list of acceptable fluids.

Table 8: Performance data on several general hydraulic fluid types.

Fluid type	Compressibility (Decrease in volume /psi)	Temperature Based Behavior	Cost ($/gal)	Life Cycle Assessment
Petroleum based composition [120] [121] [122]	~ $67\text{x}10^{-6}$ psi^{-1}	Thins considerably at temperatures greater than 200°F	30.00-50.00	Average of 4500 hrs.

Biodegradable [121] [123] [124]	~ $65x10^{-6}$ psi^{-1}	Comparable to petroleum based composition	45.00 +	~ 2500 hrs. Severe oxidation and degradation risk for some
Water-based hydraulic fluids [125] [126] [121] [127]	~ $91x10^{-6}$ psi^{-1} Changes with temperature	Narrow operating temperature range (33-212°F)	5.00-10.00	~ 1000 hrs. Corrosive; increased wear compared to former oils; can become acidic

3.5.2. Biodegradable Oils

Oil-based biodegradable fluids, known as biodegradable oils, are the first major environmentally conscious choice of hydraulic fluid available for this project. Biodegradable oils are subdivided into several key areas, shown in Table 9. Among them are synthetic oils, including hydrocarbon and polyester based fluids, as well as naturally occurring fluids like glycols and triglyceride based fluids. Major factors in deciding among these fluids includes:

- Pour point, which is the lowest temperature at which the fluid will still behave as a liquid;
- Density, which when higher, indicates a less compressible fluid;
- Any reactive qualities;
- Operational behaviors, including performance when extremely hot or cold.

Table 9: Behavior characteristics of biodegradable oil varieties.

Biodegradable Oil Name	Characteristics	Oil Density (g/ml)	Pour Point (°F)
Synthetic Polyester Blends (SPBs) [128]	Constructed of esters, the basis to most petroleum oils. Acceptable to replace mineral-based petroleum oils. Higher density than petroleum oils. Only fluorocarbon rubber seals permitted, as reaction may occur otherwise.	.92	-50
Glycols [128]	Constructed from ethylene glycol. Slightly oily and toxicity concerns in high concentrations. Excellent resistance to aging and excellent viscosity-temperature behavior. Heats up more than equivalent viscosity grade mineral oil counterpart. Higher density than petroleum oils. Cooling systems and reduced maximum pump speeds are needed for equivalent swap of petroleum oils.	Greater than 1.0	-50

Synthetic Hydrocarbon Blends [128] [125]	Preferred over glycols or SPBs in severe duty scenarios. Like SPBs, only use fluorocarbon seals. Good temperature-viscosity relationship. Higher viscosities do not degrade as readily as lower viscosities. Lower density makes poor choice for energy transfer scenarios.	.86	-104
Triglycerides Vegetable Oil Blends [125]	Fluid degrades above 140°F, lowest of group. Solidifies/gels after extended storage. Gumming and heavy varnish development under otherwise standard operating conditions.	.92	-50

3.5.3. Water-Based Hydraulic Fluids (WBHs)

Water-based hydraulic fluids typically are sold as a bottled concentrate added to conventional water as needed to meet a variety of concentrations. The additives combine biodegradability with needed lubrication and corrosion-resistance properties.

WBHs tend to be the cheapest hydraulic choices available, running as little as five to ten dollars a gallon for the additive, whereas many biodegradable oils cost in excess of forty dollars a gallon. Since water is inflammable, insurance costs from fire dangers are much lower than for oils, which burn above the flash point. Provided biodegradable additives are used, WBHs are environmentally friendly.

WBHs, however, tend to have the lowest viscosity of all the biodegradable fluid choices available. This not only increases the potential for leaks, but may increase metal to metal wear in components which rely on high viscosity fluids to provide “cushion” against metals rubbing together. Water is also a highly corrosive agent when coupled with metallic components [128]. Provided the additives used allow for anti-corrosion properties, this problem can be easily overcome.

One of the biggest issues with WBHs involves temperature behavior. While freezing can be overcome with biodegradable antifreeze agents, boiling behavior is harder to control. Since hydraulic systems can readily reach temperatures in excess of 200°F, flirting with the boiling point of water, density and performance can be affected sooner than for oils [125]. Boiling can also cause foaming, cavitations, and pressure loss. Additionally, lubrication and bacterial growth can occur more readily than for oil-based fluids [125].

3.5.4. Synthetic Polyester and Vegetable Oil Blend

Choosing the appropriate fluid for this project came down to meeting three major conditions: 1) meeting viscosity demands of the pumps; 2) biodegradability; and 3) maintaining good density and performance as temperatures rise. Petroleum based hydraulic oils, while ideally suited for the pumps and performing well across extreme temperature ranges, are not biodegradable and thus were immediately eliminated.

Biodegradable glycols, due to their mild toxicity, run counter to the environmentally conscious attitude of this project and thus were eliminated. Additionally, the extreme temperature swings and highly demanding environment of the pumps demands a robust fluid, something the vegetable based fluids couldn't provide adequately above 140°F. In comparing the effectiveness of synthetic polyester blends to synthetic hydrocarbon blends, the final choice comes down to density. Since synthetic polyester blends are denser than hydrocarbon blends, they tend to transfer more power and compress less [128]. This in turn helps to make a system more efficient, as less energy is being spent compressing fluid, and more is being spent transferring energy.

Schaeffer's specialized lubricants provided our hydraulic oil, an ISO 68 compliant synthetic ester *and* vegetable oil blend specially formulated for hydraulic uses [129]. Many of the detriments of vegetable oil have been mitigated in this product through the high concentration of oleic acid. Oleic acid is an emulsifying agent that helps to keep oils from solidifying and gelling [130].

3.6. Piping Vessels

3.6.1. Introduction and Vessel Terminology

Throughout the explanation of this system, the words hose, pipe, and tube may be used. Let it be known that these three words, while colloquially similar, are by no means the same in the hydraulics industry. All three products are sized by a diameter reading. So, a vessel which is 1¼" "thick" is measured across the diameter. The points of measurement for this diameter reading vary per variety of vessel.

A pipe's size, for example, is not reflective of the outer or inner diameters.

Rather, the value listed for the size (say ¼”) represents a *trade size*. The outer diameter of the pipe will remain constant, but not at the trade size. The outer diameter is usually larger, measured at .540” for a ¼” pipe. Depending on the *schedule* of the pipe (resistance to burst, with higher schedules being stronger) the inner diameter may change [131].

In this experiment, only *hose* will be used. The size of the hose is important for proper flow through the system. If the hose is too large, higher flow rates will be possible. However, higher flow rates can cause excessive pressure drops, a condition which may cause instability in the hydraulic system and reduce predictability. When the hose is too small, the system cannot provide the flow initially intended. Sluggish performance and cavitations are possible when oriented in this way [132].

Choosing the proper flow rate is easy, but requires prior knowledge of the optimum system flow rates. Using a *nomogram*, a straight edge can be placed between the flow rate and the maximum velocity range. The proper inner diameter, which lies between the two previously determined values, is then revealed. In the case of our experiment, the maximum system flow rate would be 60 gallons per minute. Considering a maximum pressure line velocity of 20 feet per second, the nomogram states that the inner diameter should be 1¼”, or -20 [133]. The nomogram determines this value from the formula (as per Parker Hydraulics group):

$$D = \sqrt{\frac{0.4081Q}{V}} \qquad \textbf{(11)}$$

where D is the hose inside diameter (in inches), Q is the flow in gallons per minute, and V is the velocity in feet per second.

3.6.2. Hose Construction and Pressure Ratings

Hoses are a multilayered construct that provide stability and strength to a hydraulic system. Depending on their construction, they can be rated for a wide variety of pressures. The construction starts with an inner vessel, whose inner diameter is the rated diameter of the overall hydraulic hose itself. This vessel may be made of rubber or a variety of composites to suit the particular fluid being transported [134].

The second layer is the reinforcement layer, and dictates the pressure to which the hose can be subject. Reinforcements can be multi-layered, and are made from a variety of materials, including natural fibers, steel wire, and synthetic materials. The reinforcement falls into two main categories: braids and spirals. Braiding is the traditional choice, but spirals are chosen when the highest pressures are demanded. Both braiding and spiral's strength is dictated by how many layers they have. A "two-wire" braid has two layers of wire braiding, for example [134].

The final layer is the outer covering, usually made of a cloth or textile material, and tightly stretched over the reinforcement layer. This covering, while seemingly unimportant, is actually the first line of defense against detrimental damage that the hose may experience. Many hoses are subject to chafing during normal operation; a robust cover can help prevent degradation and premature failure. Additionally, high or cold temperature environments can degrade hose covers; a well-designed cover will withstand these extremes [134].

3.6.3. Final Hose Choice

The hose chosen for our experiment was a Parker Hydraulics 3000 PSI, -20 (1 1/4” inner diameter) constant working pressure hose. Even though we will only be working with a designed maximum of 900 PSI, the extra 2100 PSI allows for improperly functioning components that may cause temporarily higher pressures in singular sections of the circuit. The hose chosen was model 451TC-20, where “451” refers to the model, a 3000 PSI hose, TC refers to the “tough cover” addition to the exterior of the hose, and -20 refers to the size [135]. The tough cover, as described by Parker, provides an additional 80 times the resistance to abrasion as the standard model. This will allow hoses to maintain integrity in the event that they rub together or in other fashion experience an abrasive environment [135].

Additionally, the 451 model hose that Parker offers has an inner diameter of 1 ¼” but an outer diameter of 1.85”, over a half inch bigger. The minimum bend radius of the hose is 8 ¼”, which means that when the hose makes a half circle of minimum diameter, 16 ½” will exist between the two ends of the hose [135]. The 451 model hose has an inner tube of synthetic rubber, capable of transporting petroleum based hydraulic fluids and lubricating oils. The reinforcement for this model is a four layer metal spiral, and a synthetic rubber abrasion resistant material makes the final cover. The hose can be safely operated in the temperature range of -40 degrees Fahrenheit to as high as 212 degrees Fahrenheit, or the boiling point of water [135].

3.6.4. Additional Hose Requirements

The main hydraulic circuit utilizes 451TC-20 hose, but additional hoses are needed for the circuit to operate properly. The Hagglunds pump and the Bosch-Rexroth axial piston pump both require pressure relief for their crankcases. The crankcases allow for two methods of achieving this pressure relief: 1) A separate hydraulic circuit pumps fresh fluid through the crankcase, utilizing both the fill and drain port for this purpose; 2) A hose is attached to the fill port (always the vertically higher port) and is attached to a hydraulic tank. If the system is left sealed at both fill and drain ports, positive or negative pressure can develop in the pump, causing unnecessary system strain [66]. By utilizing option two, we were able to provide a pressure relief without overcomplicating the circuit. The power pack provides a port on the top of the tank to which we attached a tee fitting; the tank receives a hydraulic hose, ½" (or -8) in size from both the Hagglunds pump and the Bosch-Rexroth axial piston pump [67].

The hose chosen was conceived around the general operating conditions for which these pumps would be subject. The crankcases will not operate in temperatures exceeding that of the hydraulic fluids, as the work expected of the fluid is considerably less than the main circuit fluid. Additionally, the pressure in this portion of the circuit is very low, at only 10-20 PSI above ambient. Parker recommended the 422 series hydraulic hose, as it is a readily available, low pressure hydraulic hose [135]. Parker makes several other lower pressure hoses, but they are special order hoses for a variety of non-hydraulic tasks, such as suction and transportation purposes (cars, tractor-trailers, etc.) The 422/421-8 "Worldwide" hose was used, which provides working pressure of as much as 2325 PSI, a value far above our needs. The 422 series hose achieves this robust pressure rating through use of single braid steel wire reinforcement. The nitrile inner tube and synthetic rubber outer covering complete the package. The hose has a minimum bend diameter of 14 inches, and can withstand temperatures in the range of -40 to 212 degrees Fahrenheit [135].

3.7. Fitting and Thread Styles

3.7.1. Introduction

Hydraulic systems are very complicated due to the wide and extensive variety of fittings and thread styles available to join different pipe connections. Each fitting *and* thread style provides a certain element of security against leaks, and must be considered carefully when planning which to use. Additionally, thread styles are generally not compatible; adapters must be used to convert from one style to the next. This section serves as a guide to the variety of thread styles used in this project. It does not cover every style and type of threading available for hydraulics equipment.

3.7.2. Fittings Used

The dash number system is a convention which allows for same sized hydraulic hoses to be mated properly. It multiples the dash number against 1/16” to get the actual diameter desired. When pairing different components, like relief valves and hoses, using the same dash value will allow the components to maintain the same flow rate throughout. For example, if 1 ¼” inner diameter is desired throughout an entire circuit, all the components should comply with the -20 sizing ideal, since 1/16” X 20 gives the value of 1 ¼” [134]. If any -16 or -24 components are used, the flow rate in the system will change; additionally, adapters will be needed to make these differently sized components mate.

The dash number convention is also used in *threading*, and is different from the dash convention for hoses, typically due to size differences. For several of the thread styles, a *larger* thread value is needed to meet a smaller hose. An example would be for the JIC flared fitting: to mate a -20 hose with the JIC fitting on its end, the JIC fitting would not be -20, but would be -26 [136].

Only some of the thread conventions are irregular in their mating, and must be watched for proper continuity. The JIC fittings, O Ring Boss, O Ring Face Seal, SAE 45°, and SAE Inverted Flare are all threading conventions which do not match their equivalent hose sizes, as outlined in Table 10 [137]. Differences will occur, and conversion charts are necessary. For the British Pipe Thread (BPT), National Pipe Thread (NPS), JIS, Metric DIN, and SAE Flanges, the sizes remain consistent [137]. Therefore, one will choose a -20 hose to go with a -20 sized British pipe thread fitting.

The fittings used in this project, as well as their key features, are listed in Table 10. If their sizing conventions are equivalent to hose sizing convention, it is so stated.

Table 10: Fitting characteristics and sizing conventions.

Fitting Name	Key Characteristics	Sizing Convention Same As Dash Hose Convention?
JIC Flare Fitting	Broad 37° flare on outer end of male	No. -20 hose mates

[138]	threads; inner 37° flare on female component. Provides excellent sealing surface for JIC to JIC connections. No cuts, chips or dents allowed in flare face; leaks could occur as a result.	to -26 JIC fitting.
British Straight Thread Parallel Pipe (BSPP) [139]	Demarcated by a G before the nominal thread size (G1/4, G1/2, etc.). No sealing capability due to straight threads; pipe dope or thread tape required to seal connections.	Yes, but usually sized by G lettering convention. G1¼” is equivalent to -20 sizing.
British Standard Pipe Thread (BSPT) [139]	Demarcated by a G before the nominal thread size (G1/4, G1/2, etc.) Threading tapers smaller closer to outer edge along a 55° slope. Taper provides adequate sealing capacity.	Yes, but usually sized by G lettering convention. G1¼” is equivalent to -20 sizing.
American National Pipe Thread (NPT) [139]	Straight threads in similar orientation to BSPP. Threads per square inch different than BSPP. No sealing capacity. Pipe dope/thread tape required.	Yes. -20 NPT is same as -20 hose.
SAE O-Ring Boss [137]	Wide collar with compressible, attached rubber O-ring near the hose end of the fitting (provides seal.)	No. -26 is equivalent to -20 hose sizing.
SAE 45°Flare [137]	Similar to JIC fitting, except with 45° flare instead of 37°. Cannot size up to 1 ¼”.	No. -26 is equivalent to -20 hose sizing.
SAE Flareless [137]	Straight thread similar to BSPP and NPT. No sealing capacity, so pipe dope/thread tape required.	No. -26 is equivalent to -20 hose sizing.

3.7.3. Hydraulic Flanges and the Code 61/62 Concept

Fittings are threaded components which allow mating of different hydraulic components. A flange, by comparison, is a flared, threadless tip which sits flat on a surface, and must be bolted in place by clamps that help seat and compress the tip. Flanges are used in our project for the connections to the Hagglunds pump and the axial piston pump. These connections are four bolt flange connections following SAE standard J518, used in tight spaces, higher pressures, and large port holes (Figure 48) [140]. The pressure rating is code 62, which for a -20 flange, allows a maximum of 5000 psi fluid pressure [140].

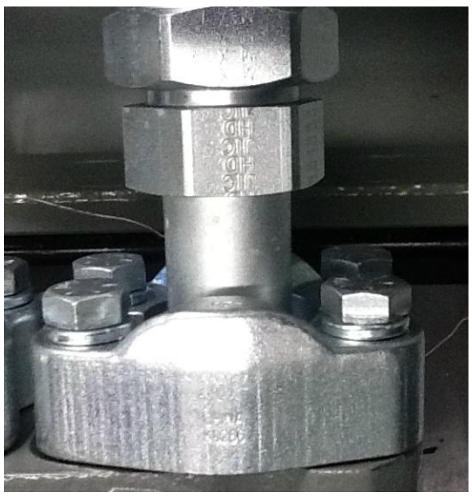


Figure 48: Code 62 SAE J518 flange connector with four-bolt attachment.

3.8. Storage Solutions

3.8.1. Introduction

Fluid which impinges upon the axial piston pump causes rotations of the pump's crankshaft, which is directly connected to the generator, thereby causing electricity generation. In a full-scale system, we can anticipate that the generator or generators on land will be directly connected to the national power grid, as shown in Figure 49.

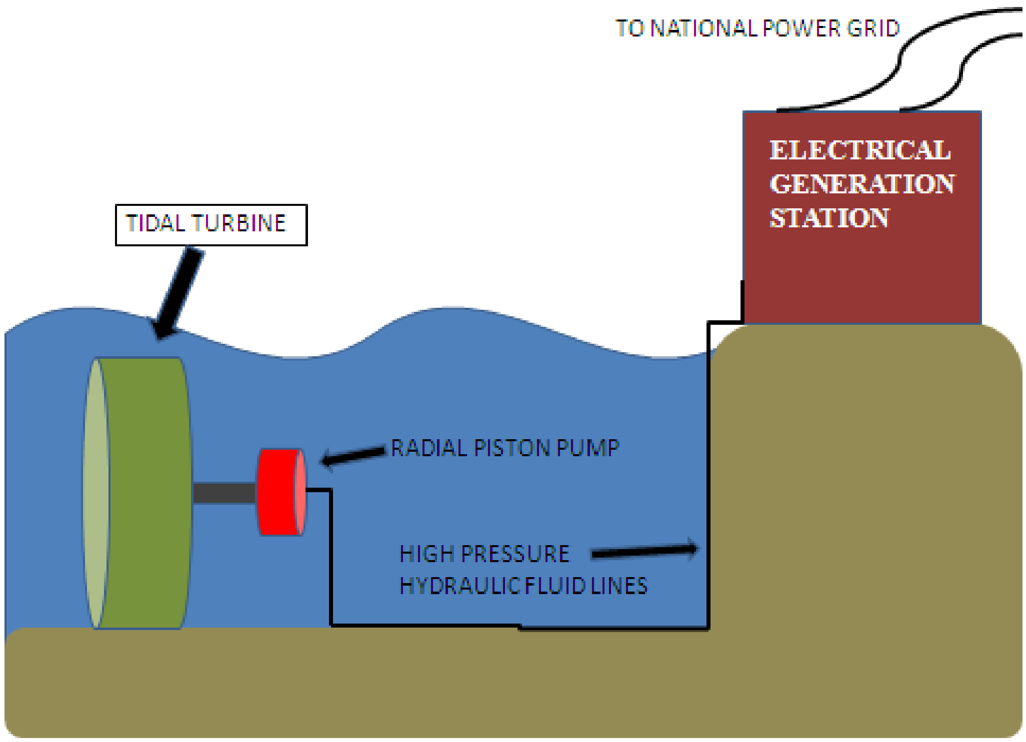


Figure 49: Full-scale system conceptualization.

In the full-scale system design, storage of the generated electricity is not considered. Since the tides are a regularly occurring phenomenon, electrical companies can anticipate the surges associated with tidal current technology. Other power plants on the national grid can be taken off-line during this time, as needed, to ensure a steady power generation to the customer.

In the model hydrokinetic system, however, all the electricity produced must be “dealt with” as shown in Figure 50. Various options include using a battery bank, a load bank, or a flywheel storage system. First, however, understanding the phenomenon behind why a generator works, and how it is affected, is critical to understanding why our final storage solution was chosen.

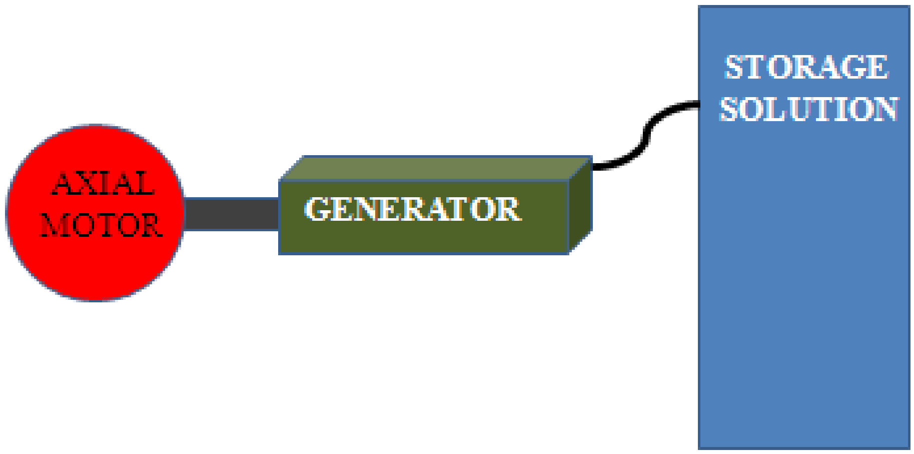


Figure 50: Conceptualization of model prototype storage solution.

3.8.2. How Generators Operate

A generator is a device which converts mechanical motion into electrical energy. In the case of the Baldor-Reliance generator considered here, the input is a rotational crankshaft connected to the Bosch-Rexroth axial piston motor. The faster the motor spins, the faster the generator spins. The input shaft of the generator is connected to a winding of copper coils mounted on an iron shaft, known as an armature [68]. The circuit and its load are connected to this armature. A load could be, for example, a battery bank, light bulb, or resistive load bank. Recall that the load is just a way for the electricity to be consumed. A high load is one which demands more electricity than is being produced by the generator. A low load is one which demands less than the generator can produce.

If we define "electricity" as the flow of electrons, then we can say that electricity is generated whenever this armature passes around a magnet. As electrons pass the negative or positive terminal on the magnet, they shift from the radiated force of the magnet. The faster the armature spins, the faster the electrons can flow, and the more productive the electrons can be, giving rise to the thought of “more electricity.” When a generator spins faster, it can light more light bulbs, for example, than it can when it spins slowly [141].

The impact of the load is significant in this scenario, and greatly impacts the operation of the generator overall. We can think of the load as a resistance to electron flow. The greater the load, the harder it is for the electrons to flow. The lower the load, the easier it is for electrons to flow. Recall that as the armature passes the negative *and* the positive ends on the magnet, the electrons flow. This behavior is a requirement. In the instances where electrons cannot flow, the armature cannot move. This occurs because the fields from the armature electrons and the magnet oppose each other *more severely than* the torque from the armature’s input shaft can overcome. This results in a condition called *stall,* or when the input shaft of the generator is forced to stop rotating [68].

With proper monitoring, such an occurrence can be avoided. Furthermore, the opposite condition can be just as problematic. If the load applied to the generator is suddenly less than the generator can produce at that moment, an electrical “surge” will be felt on the circuit [68]. If a light bulb, rated for 75 watts, is attached to a generator producing 100 watts, the bulb may burn out as a result of receiving too much electricity.

3.8.3. Approximate Power Calculations

Anticipating total power produced is critical for correctly matching load to generator output. Calculating the total power produced should consider each of the following three important concepts: 1) the electric motor will be responsible for a total maximum power rating of 20 hp in 208 volt three phase AC [64]; 2) the generator is capable of 25 hp at 460 volt three phase AC [69]; 3) hydraulic losses through the circuit may reduce pressure and flow rate seen at the axial piston pump. It is this third concept which bears the most importance in considering how much power to factor for when designing an appropriate circuit load. This factor also happens to be the hardest to properly calculate due to the various components and fittings used to manage the flow.

A safer way to calculate approximate power generated would be to consider maximum wattage generated by the electrical motor and the generator. Using formula (12), we can calculate the power generated in watts, as seen as DC output:

$$P = 1.73VIP_f \tag{12}$$

where P is the power produced in watts, 1.73 is a three phase to single phase conversion factor, V is the voltage in three phase AC, I is the current in amperes, and P_f is the performance factor [142]. Since most circuits applied to the generator demand DC voltage, all power calculations must be considered with DC voltage, not three phase AC voltage. Hence, an equation must be applied wherein conversion from AC to DC is considered automatically, as both the generator and electric motor are rated in three phase AC voltage.

In equation (12) the performance factor is a value which sums up the losses due to conversion from AC to DC power [142]. In order to reduce complications in the equation, it is generalized to "1" for the calculations provided in this comparison. In real world considerations, a reduced value would occur, as losses from the conversion to DC from AC provide significant negative impact to the efficiency of the system.

Conversion from 208 volt three phase AC and 48 amps yields roughly 19 kW of power produced by the Baldor-Reliance electric motor. In comparison, the 25 hp Baldor-reliance generator is capable of 21 kW of power from its potential of 26.6. amps at 460 volts AC. From a standpoint of building to maximum capacity, it becomes clear that one should build to 21 kW of power, as this represents the maximum output possible under all circumstances. However, building to this capacity, especially in the case of a battery bank, greatly increases costs when a lesser amount of power is actually realized. Considering that hydraulic system losses will be significant, it is highly unlikely that 21 kW of power will ever be realized in the system. Predicting the reduction before experiments are conducted is nearly impossible due to the variety of unknowns. It remains a safe bet to maintain the 21 kW power rating for calculations.

3.8.4. Managing Power Generation on the Prototype Level

Any hydrokinetic system must be designed to create electricity as the final goal. The amount of electricity produced in relation to the amount of energy harnessed via the tides directly tells us the efficiency of said system. In our prototype, a 25 HP generator is attached to the axial piston motor, as described previously. As the axial piston motor spins, electricity is generated at the generator, giving us an idea of what kind of power can be expected in real life. The full-scale system would use an axial piston motor to harness the kinetic energy from the hydraulic fluid at an on-land electricity generation facility. Our generator will give us an idea of how an on-land electricity generator station would impact system operations.

The electricity generated must be dissipated or used. Without a load to send the electricity to, a dangerous build-up of amperage and voltage will cause damage to the internal components of the generator [143]. In the full-scale system, a load will never be an issue; the generator will simply connect to the national power grid and send its electricity to consumers for use. In the prototype system, however, the electricity must be accommodated without connecting to the University's power grid. Connection to the power grid without adjustment of the electrical phase and strength would cause severe damage to the building's wiring system. Additionally, the well-orchestrated and well-timed electrical generation stations on campus would be disrupted by this surge. Additional options, explored and dismissed as appropriate, are discussed below.

3.8.5. Battery Banks

One option considered, the battery bank, often finds use in household photovoltaic systems providing continuous power to homes even at nighttime. This system combines the short-term productivity of solar panels with the long-term storage capacity of the battery bank system to provide uninterrupted power to the home, provided proper maintenance is performed and the system is sized properly. Since solar panels operate in a DC, or direct current form, and batteries charge and discharge on a DC circuit, the two are well-mated for operation [144].

The battery bank starts with the individual capacity, measured in amp-hours (Ah). Amp hours are the power of the battery measured in watts divided by the acceptable voltage in DC volts. So, for example, if the battery has a power capacity of 3000 watts over one hour, and charges at 6 volts DC, its capacity is 500 amp-hours per hour, a rather high value [145]. The equation can also be looked at from the standpoint of needed capacity. Since we base all calculations on the reality that at most, 21,000 kW would be produced by the generator (over one hour), we can presume at least this much power capacity must reside in the battery bank per one hour of operation. If the battery bank charges at 6 volts DC, we would need approximately 3500 amp-hours of capacity.

Off the shelf batteries do not come at these capacity levels, for they would be oversized for most uses and too difficult to lift and move. Instead, multiple batteries are used to achieve these system capacities. In order to achieve higher amp-hour capacities, batteries can be wired in parallel to additively increase capacity. If two 500 amp-hour batteries are wired in parallel, the combined battery bank has a capacity of 1000 amp-hours. Battery bank voltage can be increased, conversely, by wiring batteries in series. So, for example, if 12 volt DC charging voltage for the battery bank is desired but the batteries being used are 3 volt batteries, four batteries in series will need to be wired [145].

3.8.5.1. The Requirements of a Battery Bank

Unlike a photovoltaic system which operates in DC, a hydrokinetic system operates in AC, both at the energy production phase and at the energy consumption phase. In order to create a functional, efficient battery bank, the following necessary components are compiled in Table 11.

Table 11: Components needed for properly functioning battery bank.

Component Name	Function
Transformer [146]	Step voltage down from 460 volts to 48, 24, or equivalent voltage for battery bank
Rectifier [146]	Convert from three phase AC to DC
Battery State Monitor [144]	Oversees charging rates and capacity of battery bank. Should a fault be detected, charging can be stopped to avoid damage to bank.
Battery Bank [144]	Holds power generated. Can be constructed from a variety of amperages, voltages, and chemistries.
Inverter [144]	Convert from DC to three phase AC.
Transformer [144]	Step up voltage from 24, 48 (or equivalent) to 208 volts for consumption
Conditioner [144]	Adjust phase of produced electricity to meet load’s phase pattern

Varying battery chemistries allow for a variety of charging styles and storage solutions in a battery bank. Different battery chemistries currently available are listed in Table 12, as well as their merits and demerits.

Table 12: Various currently available battery technologies.

Battery Name	Advantages	Disadvantages
Lead Acid [147] [148]	Proven technology/reliable. Deep cycles well, and handles electrical irregularities well.	Largest and heaviest battery. Cannot be fast charged

	Up to 500 Ah capacity per battery. Inexpensive.	(charge at up to 1/8th rated capacity per hour). Needs periodic maintenance: venting, water added, charge equalization.
Nickel Cadmium [149]	Can charge at 10 times rated capacity. Performance not affected by severe temperatures.	Used primarily for small storage applications (1-2 Ah). Cadmium is toxic, leading to complete phase out of technology (currently on-going).
Nickel Metal Hydride [149]	Twice the energy density of lead acid batteries. Fast charging, like nickel cadmium batteries.	Only good for small storage applications (less than 50 Ah).
Lithium Ion [150] [151]	No liquid electrolyte reduces maintenance. High densities (1000 Ah). Can charge at rated capacity per hour. Can discharge at up to 40 times the rated capacity.	At slow charge/discharge rates, more expensive than lead acid batteries for equivalent storage.

The only two acceptable battery choices for this experiment are the lithium ion battery and the lead acid battery. The nickel cadmium battery is currently being phased out due to toxicity, and therefore is an unacceptable choice; its forbearer, the nickel metal hydride battery, is far superior to the lead acid battery in terms of energy density, but simply isn't available at the amp-hour sizes needed for this project. While the lead acid battery has a slower charging rate than the lithium ion battery, it is far superior in durability and cost, making it a reasonable competitor. The lithium ion battery, in comparison, will be a far more space compact solution, but runs the risk of being more expensive to build, depending on charging rates.

3.8.5.2. Choosing the Best Battery Bank

Knowing that our system produces 21 kW of power per hour of operation, choices arise as to the voltage and capacity of each battery in the bank. Outlined in Table 13 are the options, using two kinds of batteries. The first, the Rolls Surrette lead acid battery, is a very large battery with 450 amp-hour capacity at 6 volts [152]. The second battery is the Real Force lithium ion battery, which has a charge capacity of 240 amp hours at 3.2 volts [151]. The desired battery bank, if 48 volts, requires a capacity of 441 amp hours to meet the 21 kW power storage requirement.

Table 13: Battery quantity needed to satisfy voltage and capacity requirements.

Battery Type	Batteries in Series to Create 48 Volts	Batteries in Parallel to Create 441 Ah	Total Battery Count	Total Battery Cost
Rolls Surrette @ $400/battery Charge Capacity/Hour: 450/8=56.25 Ah Voltage: 6V	8	8	64	$25,600.00
Real Force @ $325/battery Charge Capacity/Hour: 240 Ah	15	2	30	$9750.00

Voltage: 3.2V				

The charging rate here was the determining factor in choosing the lithium ion battery over the lead acid battery. Since the lead acid battery cannot be charged very quickly, the cost of the bank becomes over twice that of the lithium ion bank since over double the number of batteries are needed to properly hold the power generated, as evidenced in Table 13.

Considering the rough costs of the additional components needed in addition to the costs of the bank itself, and the total for a lithium ion battery bank rises steeply as shown in Table 14.

Table 14: Parts breakdown and cost estimate for battery bank.

Component Name	Quantity Needed	Cost Per Component	Total Line Cost
Lithium Ion Battery Bank	1	9750	9750
Transformer [144].	2	750	1500
Battery State Monitor [144].	1	295	500
Rectifier [153].	10	435	4350
Inverter [154].	1	10000	10000
Line Conditioner [155].	1	8000	8000
		Total Cost	34,100.00

This represents the least expensive option to run the prototype for one hour, the minimum testing period in the creators' minds. With such a high cost and incredible complexity level, it becomes apparent that a battery bank system is not an ideal or cost effective way to manage power generator from the system.

3.8.6. Load Banks

The load bank is a device which has been in use for several decades, providing electrical load on an as-needed basis. Often, load banks are used to test the durability and performance of back-up generator systems currently installed at a facility. Instead of waiting for a power failure to determine the viability of a diesel generator, for example, one could attach a load bank and operate the generator independently of true electrical need. Load banks also provide a way to test brand-new electrical equipment without waiting for the in-service period. Problems with new equipment can be resolved before shipping to the customer [156]. The load bank comes in several varieties, but in general, takes the power produced by a generator or other electrical source and dissipates it immediately [157].

In our experiment, the load bank would need to provide a variable three phase AC load to the generator. The load should be adjustable by the experimenter, such that the effects of load upon not only the generator but the hydraulic system as a whole can be inspected and assessed. Said load bank should be quick to install and disassemble, due to the brief nature of operation, and cost less than $10,000 to operate over the period of testing.

Load banks can provide three different types of load: inductive, capacitive, or resistive. The three types of loads and their characteristics are listed in Table 15.

Table 15: Load bank operational characteristics.

Load Bank Type	Characteristics
Inductive [156].	Load wherein current waveform is delayed a few milliseconds behind voltage waveform. Ideal for testing lagging power factors on generators. Represents load from electric motors and transformers, and is ideal for generators which supply power to these sources.
Capacitive [158].	Current waveform *leads* voltage waveform by a few milliseconds. Generator actually stores electricity during part of the AC cycle then releases it as a surge during another portion of the cycle. Ideal for testing capacitive banks (power storage units) and underground electrical cables for integrity (as visual inspections can be difficult to administer).
Resistive [159] [160] [161]	Most common type of load bank available. Unlike capacitive and inductive, load is direct and immediate, with current and voltage waveforms directly overlapping. Represents load from lights, heaters, electronics, and other standard household demand.

The load bank chosen for this experiment came down to consideration of the average load a generator in a full-scale system would experience. Since the full-scale system would experience a load effect from a resistive load, the resistive load bank made the most sense to choose. Resistive load banks, rentable from ComRent LLC, cost on average $1300.00 a month for an Avtron K595 55KW resistive load bank (Figure 51). The 55kW unit from ComRent has load bank steps of 2 kW, loading up to 55 kW upon a generator [161]. This allows a precise and controlled increase of the load. The other recommended model by ComRent, the Avtron LPH100, is a 100 kW unit with load steps as small as 5kW. These large load steps represent almost 25% of total electrical capacity, making far more dramatic an influence on generator operation than with the K595 unit [160]. In the end, the K595 unit was chosen for its reasonable cost and simpler construction. At the end of testing, the load bank can be returned; battery banks, however, would require the sale and/or disposal of several batteries at a significant loss.

Figure 51: Avtron load bank.

3.9. Bosch-Rexroth Power Pack

Operation of the Hagglunds radial piston pump requires operational considerations that are unique to this particular brand and style of pump. The pistons are arranged such that no springs or additional complexity is used to motivate them in their bores during operation. Simply put, the wavy outer cam ring provides the compression stroke upon the piston. During the intake stroke, however, the pump relies on the intake fluid's pressure to push the piston down in its bore, thereby accepting a quantity of fluid. This pressure is quantified as the *charge pressure*, and has minimum values for successful operation of the pump. With a charge pressure too low, the pistons become sluggish and do not fill up properly during the intake cycle. They can disconnect from the cam ring, and experience quick, abrupt reconnection with the cam ring, causing loud banging and potentially damage to the pump [66].

Bosch hydraulic consultation group provided the initial design of the hydraulic circuit, and briefly but sternly mentioned the importance of charge pressure to this system's proper operation. From rest, the hydraulic circuit fails to provide sufficient charge pressure for the pump to operate normally. The power pack is designed to provide boost to the system during this critical start period and provide a location to store and draw fluid from. The power pack contains a ¾ horsepower, three phase AC electric motor attached to a 20 gallon oil reservoir (Figure 52). The power pack has three ports. The inlet port contains a breather valve and accepts a portion of the system fluid as needed. The outlet port provides hydraulic fluid which flows under the power of the electric motor, but whose outlet pressure is modulated by a variable strength check valve. The check valve can be adjusted to retain fluid until a sufficient pressure is reached; at this point, the fluid would then be released [67]. The final port is a direct line to the tank; it allows for easy filling or in our case, provides pressure relief and a steady oil supply as needed to the crankcases of the axial piston and radial piston pumps.

Figure 52: Bosch-Rexroth 20 gallon power pack.

The importance of the power pack is very high during the initial start of the hydraulic circuit and during shut down of the system. During normal operation, where the axial piston motor provides adequate charge pressure to the Hagglunds pump, it is believed that the power pack becomes unnecessary. The hydraulic circuit, outlined later, is designed such that a total bypass of the power pack can be achieved during spirited operation.

During system shut down, however, use of the power pack becomes as critical as during start up. The power pack provides a collection location for the hydraulic fluid used in the circuit. In this way, it serves as an air bleeding mechanism and helps to maintain proper fluid volume in the pump and motor. As the system heats, it can cause the fluid to volumetrically expand and contract, as per the temperature at that point; the power pack provides relief to these changes in fluid volume. Finally, the power pack, during the final slow speed rotations of the Hagglunds pump, provides charge pressure once again to avoid piston slamming and potential system damage.

3.10. Bosch-Rexroth Check Valves

3.10.1. Introduction

The hydraulic circuit designed by the hydraulic specialists at Bosch requires several safeguards for proper use. Since positive displacement pumps fail to reduce volumetric displacement during blockages or other low flow conditions, the system must contain intelligence that can stop or divert hydraulic pressure during these scenarios. Two types of check valves are employed in this circuit, the DBDS series check valves and the one-way check valve.

3.10.2. The Bosch-Rexroth One-Way Check Valve

The one-way check valve serves as a basis for the more complicated DBDS series check valves. This one way check valve is employed at the exit port of the power pack to ensure that even when system pressure exceeds power pack outlet pressure, hydraulic fluid will not flow backwards into the power pack, thereby diminishing system pressure *and* subjecting the power pack to excessively high pressures for which it is not designed (Figure 53) [162].

Figure 53: Bosch-Rexroth one-way check valve (center, black).

The one way check valve employs a ball and spring arrangement to achieve its desired operation. The spring presses against the ball, which in turn presses into a cone shaped port. During normal flow direction, the hydraulic fluid presses against the steel ball under compression, pushing it away from the cone and allowing fluid to escape. The check valve can be designed for a variety of system pressures, but in this case, the spring tension is actually quite low. The check valve is designed to prevent backwards flow, not modulate pressure [162].

If fluid flow should ever reverse, it will flow past the spring and against the steel ball, thereby pushing the steel ball further into the cone and stopping the circuit. This simple yet robust design is an incredible effective method of modulating fluid flow direction [162].

3.10.3. The DBDS Series Check Valves

The Bosch-Rexroth group recommended use of the DBDS series check valves as an additional system safety. Mounted on the auxiliary portion of the hydraulic path, these check valves only open when system pressure is high enough, thereby providing fluid bypass as needed. Two check valves are used: a 10-bar check valve and a 55 bar check valve (Figure 54). Should system pressure exceed either of these values, the valve will open and fluid will flow, thereby providing pressure relief off of the main circuit [163].

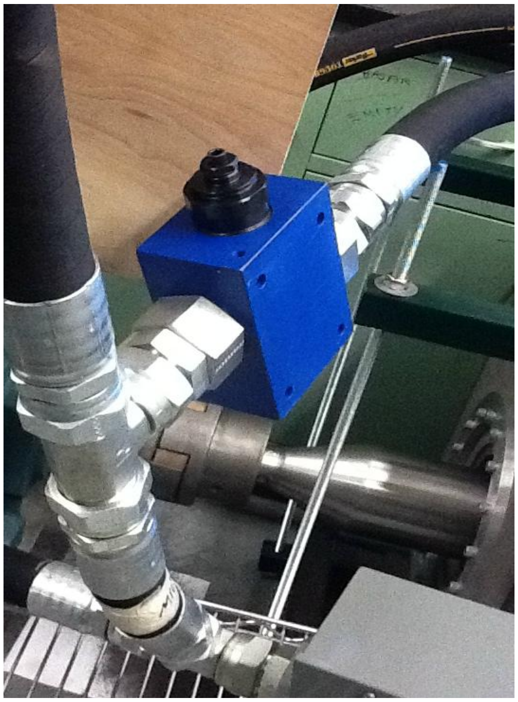


Figure 54: The 55 bar DBDS series check valve (center, blue) used in the hydraulic circuit.

The check valves have several components. The first component is the main inlet flow path, P, of which there is a female port on the body of the check valve to accommodate. The second portion is the adjustable spring and ball valve assembly, which have a rotating cuff on the exterior of the check valve to adjust spring tension. The spring presses the ball into a cone shape. The more tension on the ball, the harder it is to lift the ball from the cone face and allow fluid to flow. The less tension, the easier it is to lift the ball from the cone face and permit fluid flow. In this system, fluid comes into the check valve through path P and presses against the ball from the underside of the cone. When fluid pressure is great enough, the ball lifts from the cone and fluid flows past the cone and out through the outlet port path, path T. The system can be modulated to provide a range of pressures (near the rated pressure) via the adjustable cuff [163].

This check valve design has an operating range of 5 to 300 degrees Fahrenheit and requires use of specific hydraulic oils to function properly, of which our vegetable oil and synthetic ester combination is an acceptable type [163].

3.11. Sensor and Metering Technology

Monitoring of system performance is critical for safe and efficient operation. Robust electrically operated sensors and meters allows for monitoring of system performance from a computer or other remote device. The hydraulic circuit has two types of monitoring equipment being used during operation: pressure transducers and flow meters.

The first class of monitoring equipment, the pressure transducer, allows the operator to read pressure at a specific system point through use of an electronic device the size of a spark plug (Figure 55). This device houses a rubber membrane attached to an electronic circuit. The electronic circuit inputs from 19 to 36 volts DC and 4 to 20 milliamps of power to the sensor, depending on demand. The electronic circuit, in turn, is connected to a computer controller which reads and interprets the sensor requests, especially the amperage, and converts that into a readable pressure signal. Since the pressure transducer has a pressure range of 0 to 100 bar, we can reasonably deduce that 4 milliamps represents 0 bar pressure, while 20 milliamps of draw is 100 bar of pressure [164].

Figure 55: Pressure transducer (center) is connected to the data acquisition system via electrical cabling, which attaches at the top of the unit.

The pressure sensor uses a 4 pin port to connect the sensor to an electronic circuit and a G1/4 threaded port on the opposite end to connect to the hydraulic circuit. Inside the G1/4 port housing is the rubber membrane, which is in turn connected to the electronic circuit. The higher the system pressure, the greater the membrane is deflected. The greater the deflection, the greater the amperage draw, up to the maximum of 20 milliamps [164].

The hydraulic circuit also employs two Hedlands brand flow meters to measure approximate system flow on the low pressure return side (Figure 56). The Hedlands flow meter is a bi-directional unit, and therefore can be used to measure fluid flow in either direction. It employs an electronic circuit similar to the pressure transducer. Unlike the transducer, however, it does not output a signal when zero flow occurs. Current draw occurs only under fluid flow [165].

The unit has a cone and port assembly. The cone is a sealed cavity centered in the flow meter box. A spring-loaded, mobile port surrounds the cone. The port is capable of shifting up and down the length of the cone, but has an opening diameter which is less than that of the widest diameter of the cone. During the no pressure, no flow scenario, the port presses against the cone body and provides no passage of oil. As system pressure and flow increase, the port is pushed away from the cone body, moving against the tension of the port spring. The port body moves as far away from the cone's widest end as needed to allow adequate fluid flow. The position of the port body is recorded as the approximate current flow rate. Since the port body only shifts as much as is needed to achieve nearly unrestricted flow, it accurately represents current flow rates [165].

Figure 56: Hedlands flow meter.

3.12. Electrical Circuitry

3.12.1. Introduction

Operation of this hydraulic system requires use of commercial grade professionally installed electrical circuits. High gage wiring is used to provide the voltage and amperage needed for the circuit to be operated safely. Circuit breaker boxes, emergency shut off buttons, and dedicated switches help complete the safe design.

3.12.2. Main Electrical Circuit

The majority of the circuit is modulated and controlled via one 60 amp, 208 volt three-phase AC service line from the main university grid. This high voltage, high amperage circuit is first controlled via a power box with a throw lever. In order to initially energize the circuit, the throw lever must be shifted to the ON position. Pulling the lever to the OFF position causes the circuit to de-energize immediately and completely. The second step involves the circuit breaker box, which is mounted on a wood panel over the electrical motor, directly next to the variable frequency drive. This box receives all 208 volts and 60 amps of service. It is immediately disabled via two emergency shut off buttons (EMOs) located in two distinct, distant positions in the room. The first EMO is mounted next to the front door of the room, and immediately de-energizes the circuit by depressing it fully. The second EMO is located on the circuit breaker box itself, and accomplishes the same task independently of the first. Both EMOs are colored bright red for easy identification.

The circuit employs an additional safeguard in the circuit throw lever, a lever which is mounted on the face of the circuit breaker box (Figure 57). When placed in the OFF position, this lever does not allow electricity to flow to the power pack nor the electric motor. When placed in the ON position, electricity may flow. If the circuit is overloaded, the circuit breaker will de-energize the circuit. In order to re-energize the circuit, the lever must be turned to OFF and then rotated back to the ON position.

Once the system has been energized, the electric motor and power pack can be controlled via two switches. The power pack is turned on and off via the left switch, whereas the right switch allows flow to the electric motor via the Mitsubishi variable frequency drive. During an emergency, the system is designed to be shut down quickly via the emergency shut offs, the rotating lever, or the switches located on the circuit breaker face.

Figure 57: Electrical circuit breaker box (left) and Mitsubishi VFD (right). The circuit breaker box has emergency shut off (upper left), circuit throw level (middle right) and power switches for VFD (lower left) and power pack (lower right).

3.13. The Model Hydrokinetic System in Full Detail

For this section, Figure 58 will serve as the reference diagram. When referring to H1 through H15, the text is referring to hoses on the diagram. P1 through P4 refers to different pressure transducers. Finally, T1 through T10 refers to the many tee fittings used to join hoses together.

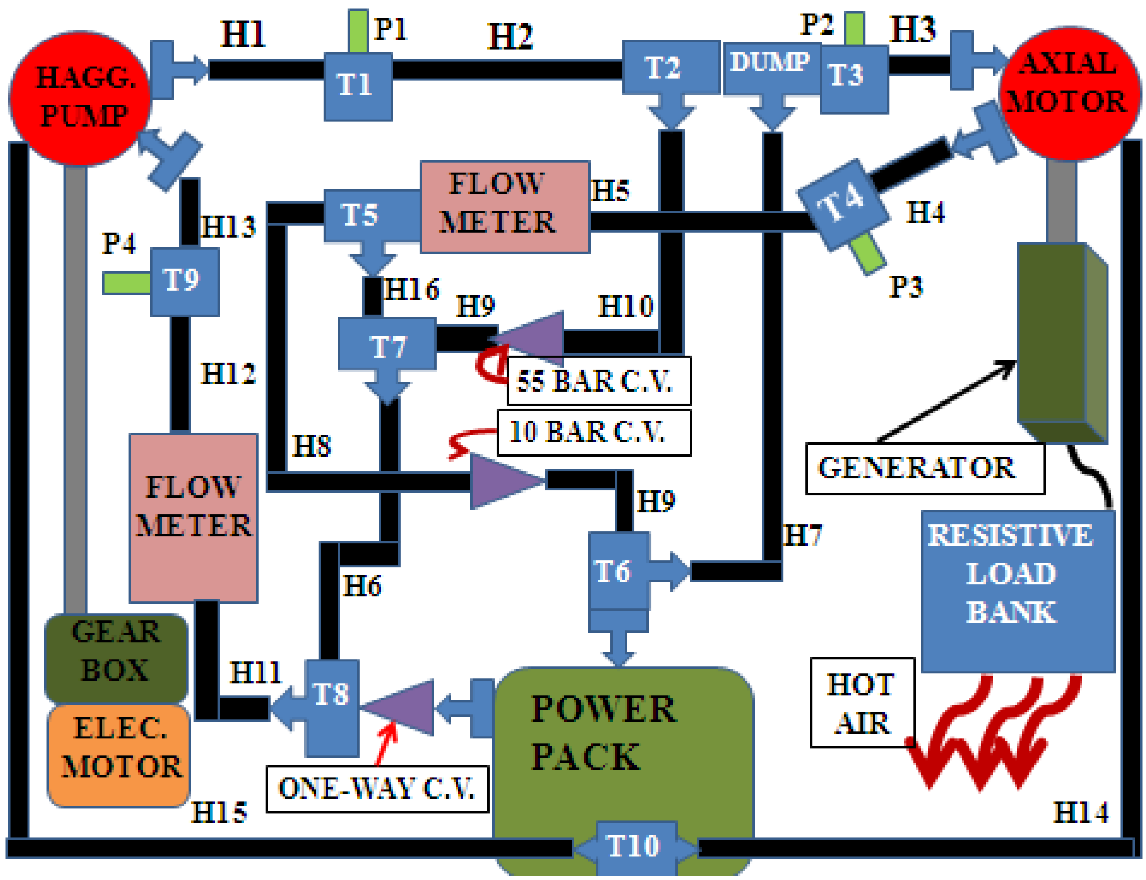


Figure 58: Full detail hydraulic circuit diagram.

3.13.1. The High Pressure Path

The high pressure circuit begins at the electric motor, as shown in Figure 58. The Baldor-Reliance 20 horsepower electric motor spins from 0 to 1765 revolutions per minute, being fully controlled by the Mitsubishi variable frequency drive. The Quantis gearbox attached to the electric motor (Figure 59) reduces output shaft speeds to a range of 0 to 120 revolutions per minute for the respective electric motor range. The gearbox output shaft is then connected to the Hagglunds radial piston pump via the first set of LoveJoy couplers.

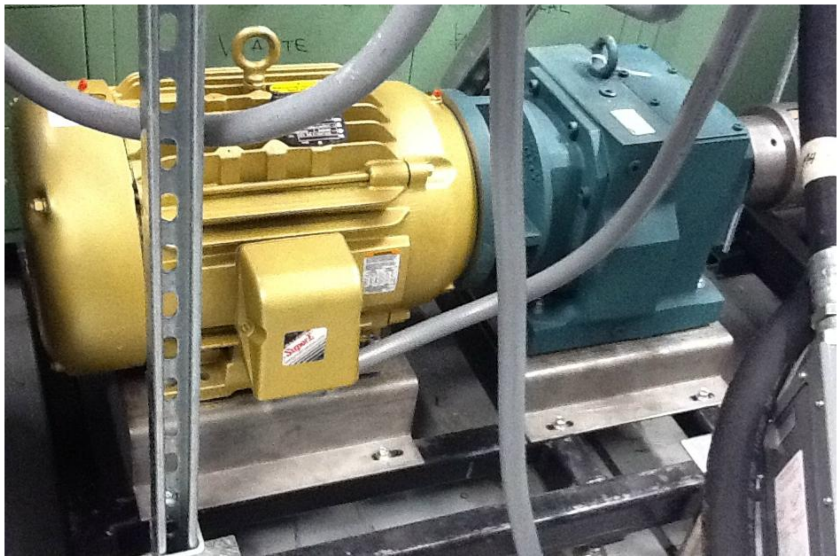


Figure 59: Electric motor (left) and gearbox (right) combination.

When the driveshaft of the radial piston pump is rotated, pressurized fluid is produced. The faster the Hagglunds pump's driveshaft spins, the greater the pressure and the greater the flow rate of the fluid. Internally, the radial piston pump has a wavy cam-ring attached to the driveshaft of the pump. As this cam-ring spins, it impinges upon pistons, pistons which compress and pressurize fluid.

Upon exiting the Hagglunds pump, the fluid flows in one of three different directions. They are either:

1) The main circuit, a direct path to the axial piston motor, or;

2) The first safety circuit, employing a 55 bar check valve, or;

3) The dump valve circuit, also known as the by-pass circuit.

During start-up, or when the high pressure lines need to be immediately depressurized, the dump valve circuit is employed. When the high pressure circuit reaches pressures greater than 55 bar (approximately 800 PSI) the first safety circuit will be used. Under all other operating conditions, the main circuit is used.

During operation of the main circuit, fluid flows from the radial piston motor through H1, T1, H2, T2, DUMP, T3, and H3 before flowing into the axial piston motor, as per Figure 58. H1 is a steel pipe of 18", 1 ¼" on the inner diameter (Figure 60) connected to the radial piston pump via a SAE J518 flange. The other end of the pipe is connected to T1, which also holds the first pressure transducer, P1. P1 accurately relays the pressure of the Hagglunds discharge port. The pressure transducer produces an electrical signal which is relayed to the data acquisition system via copper cabling for recording and interpretation. H2 is a Parker Hydraulics 451TC-20 hose (detailed previously) and connects to T2. At T2, the fluid can flow straight through to continue the main hydraulic circuit path, or divert 90° to the first main safety circuit, described in detail later.

Figure 60: 18" pipes, center, allow both outlet and inlet ports of the Hagglunds pump to clear the sheet metal enclosure tank.

From this point, T2 ends and is directly screwed into DUMP. The pressurized hydraulic fluid then flows through the DUMP ball-valve, a three way valve that allows diversion of fluid to the dump circuit via a lever (Figure 61). Under normal circuit operation (dump valve “off”) the fluid will make a 90 degree turn to the left inside the ball-valve and exit towards the axial piston motor. If the lever is thrown such that the dump valve is now in the “on” position, the fluid will turn 90 degrees to the right and flow through the dump circuit.

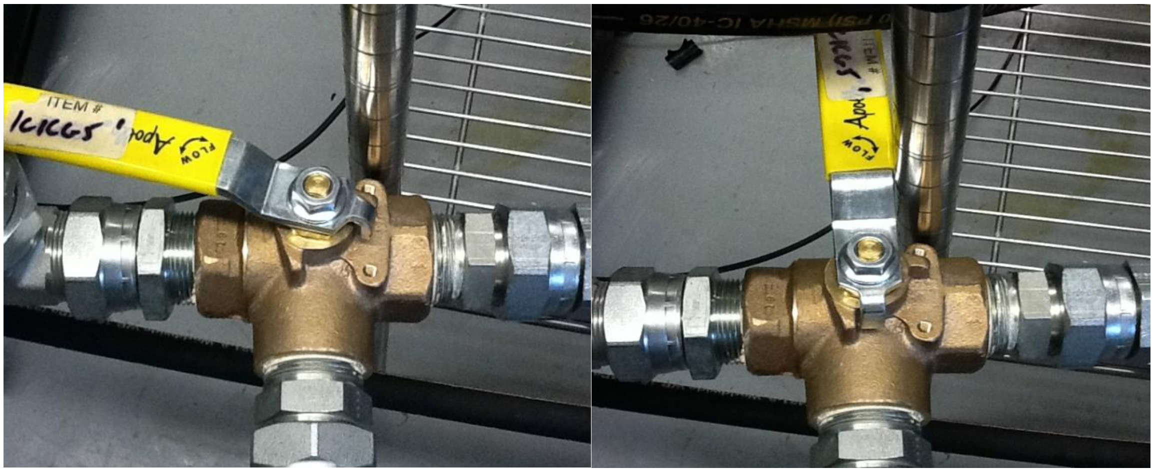


Figure 61: DUMP circuit ball valve. Dump valve is closed (left) when lever is as shown. However, when rotated 90 degrees (right) dump valve is active, bypassing axial piston motor.

After flowing through the dump valve, the fluid flows straight through T3. Attached to T3 is the second pressure transducer, P2. P2 relays data on the inlet pressure to the axial piston motor. It gives the user valuable data on pressure losses occurring from the outlet of the Hagglunds pump to the axial piston motor, if any. Like P1, all data is relayed via copper cabling to the data acquisition system. Hose H3 is also a 451TC-20 hose, and connects from T3 to the axial piston motor via a SAE J518 flange.

The axial piston motor takes the high pressure fluid it receives and converts it into mechanical motion. The high pressure fluid pushes down on the pistons inside the pump body, causing the angled swash plate to rotate. The depressurized fluid then exits the outlet port of the pump body to continue through the low pressure circuit.

When the swash plate rotates, the attached driveshaft (on the exterior of the pump body) also spins. This driveshaft is connected to the generator via the second set of LoveJoy couplers (Figure 62). When the generator spins, electrical potential is created. This electrical potential is utilized by the attached load bank (Figure 63). As the load bank's load demand increases, the torque resistance on the output shaft of the generator rises. This increased torque therefore makes it harder for the axial piston motor to spin. This resistance has a net impact of increasing heat gain and pressures in the hydraulic circuit, especially at the motor.

Figure 62: Axial piston motor (far left, gray) is attached to the generator (center, green) via the second set of couplers.

The electricity generated at the load bank is dissipated by heating resistive heater coils. A large fan blows cool ambient air over the hot coils, and the hot air is ducted out of the room. As the load becomes greater, so too does the heat generated at the load bank.

The load bank helps to simulate the effects of electrical demand on a hydraulic system. These effects can later be applied to the full-scale system to establish the net impact, if any, of electrical demand on a full-scale prototype.

Figure 63: Load bank (upper left) is attached to the generator (lower center) via electrical cabling.

3.13.1.1. The DUMP Circuit

The dump circuit has been designed to allow for the bleed-off and elimination of system pressure as needed once the system has been shut down. Additionally, the dump valve allows for complete by-pass of the axial piston motor, necessary during start of the system. After leaving the dump valve body, hose H7 routes to T6, which is connected directly to the inlet of the power pack, the 20 gallon reserve body. When the dump valve is activated, high pressure fluid from the Hagglunds pump surges into the power pack. A breather valve on the power pack inlet helps to bleed out air as needed. The high pressure fluid that enters the power pack is then forced out of the power pack by the natural capacity limitation of the unit. This fluid flows through the outlet and continues into the low pressure circuit.

The axial piston motor is a high inertial load motor. In order to start spinning the driveshaft of the axial piston motor, a minimum pressure of greater than 100 PSI must be met on the inlet of the motor. From start-up, however, the radial piston pump cannot provide these pressures due to the lack of available torque from the electric motor. As a result, the Hagglunds pump fails to rotate and the whole system fails to properly pressurize. The only way to achieve over 100 PSI in the high pressure circuit is to temporarily bypass the axial piston motor, then engage the main circuit once the requisite pressures have been achieved.

3.13.1.2. Safety Circuit # 1: The 55 Bar Check Valve

The 55 bar check valve is designed to bleed off and divert high pressure fluid from the system as needed. It is designed such that when hydraulic fluid at greater than 55 bar should emerge from the radial piston pump *outlet*, it has the ability to circumvent the axial piston motor and move directly back to the radial piston pump. This bypass prevents damage to the axial piston motor and generator, since operation at such high pressures would equate to rotational speeds at the motor which exceed manufacturer guidelines. Should the 55 bar check valve be activated, the system will begin to act erratically, with the axial piston motor operating more slowly than anticipated. The operator should shut down the system immediately to avoid damage to the radial piston pump. If the check valve is continuously used, it becomes hot; this heat is a hallmark symptom of a system operating beyond limits.

Hose H10 emerges from T2 and connects to the 55 bar check valve. From the check valve, the fluid flows through hose H9 into T7. By design, T7 allows fluid flow either backwards towards H16, or forwards through to H6. However, because moderate pressure (100-200 PSI) fluid will be coming from the axial piston motor, it is anticipated that any by-passed fluid will be forced towards H6, as it is the easier route to follow.

3.13.2. The Low Pressure Circuit

The low pressure circuit is designed to return the depressurized hydraulic fluid from the axial piston motor back to the Hagglunds pump to be pressurized again. However, depending on the particular scenario, one of several paths may be employed. They are:

1) The main circuit, which flows from the axial piston motor to the Hagglunds pump and bypasses the power pack, or;
2) The dump circuit, which flows from the high pressure circuit into the power pack and then on to the Hagglunds pump, or;
3) Safety circuit #1, which bypasses the power pack and axial piston motor, and then flows to the Hagglunds pump, or;
4) Safety circuit #2, which flows directly into the power pack from the axial piston motor, and then on to the Hagglunds pump.

The main circuit is the quickest way for the fluid to return to the Hagglunds pump, but is only used when the dump valve is not “on” (scenario 2) the high pressure circuit is less than 55 bar (scenario 3) and the low pressure circuit is less than 10 bar (scenario 4). After depressurization, the fluid flows through a SAE J518 flange into hose H4 (451TC-20) and into T4. At T4, pressure transducer P3 records the discharge pressure of the axial piston motor. From here, fluid flows through H5 into the first of two Hedlands flow meters. The flow rate here is notated via electrical signal, and communicated to the data acquisition system via copper cabling.

From here, T5 allows one of two flow paths to be utilized (Figure 64). If the fluid flows straight from T5, it engages scenario 4 (safety circuit #2) to be described in detail later. Therefore, the main circuit requires the fluid to flow 90 degrees upward to H16 into T7. T7 joins with the first safety circuit, and from here, H6 connects to the outlet of the power pack (T8). The fluid then flows through H11 into the second Hedlands flow meter. The data collected here is transferred to the data acquisition system via copper cabling.

Figure 64: Flow to the left (out of the Hedlands flow meter) through H8 (bottom left) goes to the 10 bar check valve. Flow upwards, through H16, follows the main circuit (center).

After the flow meter, H12 transports the fluid to T9, where pressure is recorded at P4. P4 gives a snap-shot view of the inlet pressure to the Hagglunds pump. A steel 18" pipe, akin to H1, returns fluid into the Hagglunds pump to be pressurized again. Connection to the Hagglunds pump is achieved via a SAE J518 fitting.

3.13.2.1. Safety Circuit #2: The 10 bar Check Valve

The second pressure relief circuit engages at lower pressures than the first safety circuit and is designed to safeguard against excess pressures on the low pressure circuit. This check valve only allows fluid flow if pressures on the outlet port of the axial piston motor exceed 10 bar. When pressures exceed 10 bar, fluid flows through the check valve and into the power pack (Figure 65); this causes a loss of pressure and velocity.

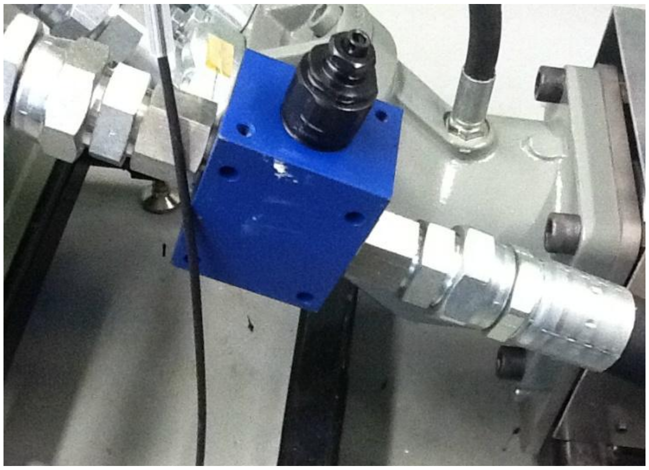


Figure 65: 10 bar check valve (blue) sits near inlet of power pack.

Fluid with greater than 10 bar of pressure flows straight through T5 instead of turning 90 degrees and flowing upwards. The fluid then flows through H8, the 10 bar check valve, H9, T6, and then into the power pack. After having its pressure and velocity reduced, the fluid emerges from the outlet of the power pack and rejoins the main low pressure circuit at T8 (Figure 66). The remainder of the fluid then flows back to the Hagglunds pump, and the circuit is completed. The experimental prototype, as shown in Figure 67, shows the true complexity of the final design.

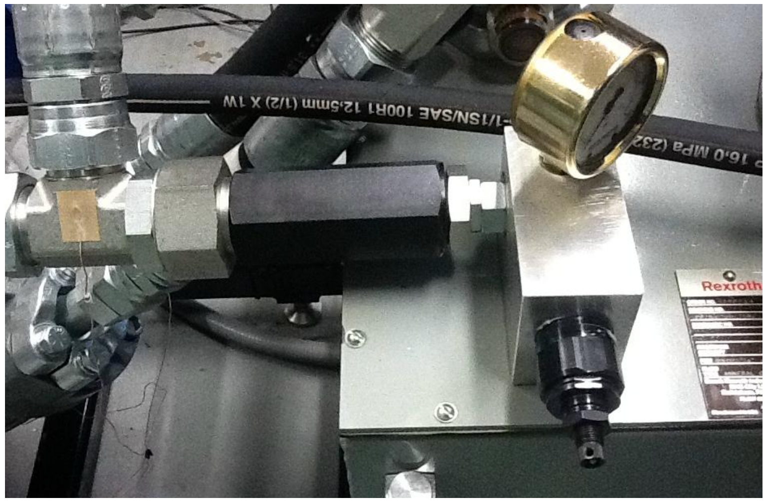


Figure 66: Outlet of the power pack (right, with gauge on top). One way check valve (center, black dowel) prevents fluid from flowing into the power pack at this port.

Figure 67: The experimental prototype.

3.13.3. Crankcase Pressure Relief Circuit

The final circuit in the system uses much smaller hoses, Parker Hydraulics 422/421-8 hoses, to connect to our axial piston motor and radial piston pump. These hoses help relieve pressure on the crankcases and provide additional oil when needed (Figure 68). Emerging from T10, on the top of the power pack, are two -8 hoses; H14 flows into the crankcase port of the axial piston motor, whereas H15 flows into the crankcase port of the Hagglunds pump.

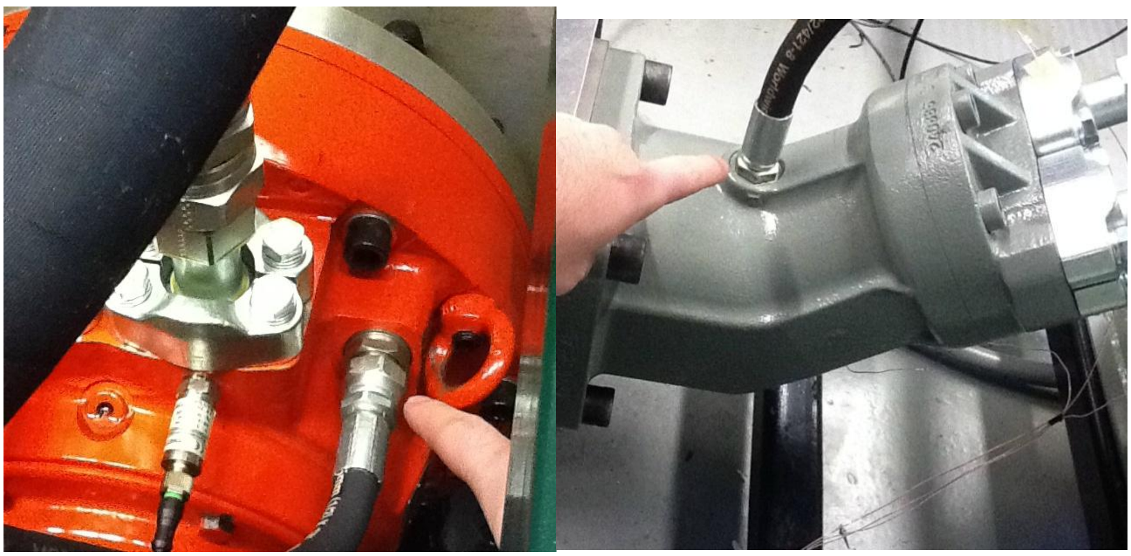


Figure 68: Crankcase pressure relief port openings on the Hagglunds pump (left) and the axial piston motor (right).

3.14. Power Production Summary

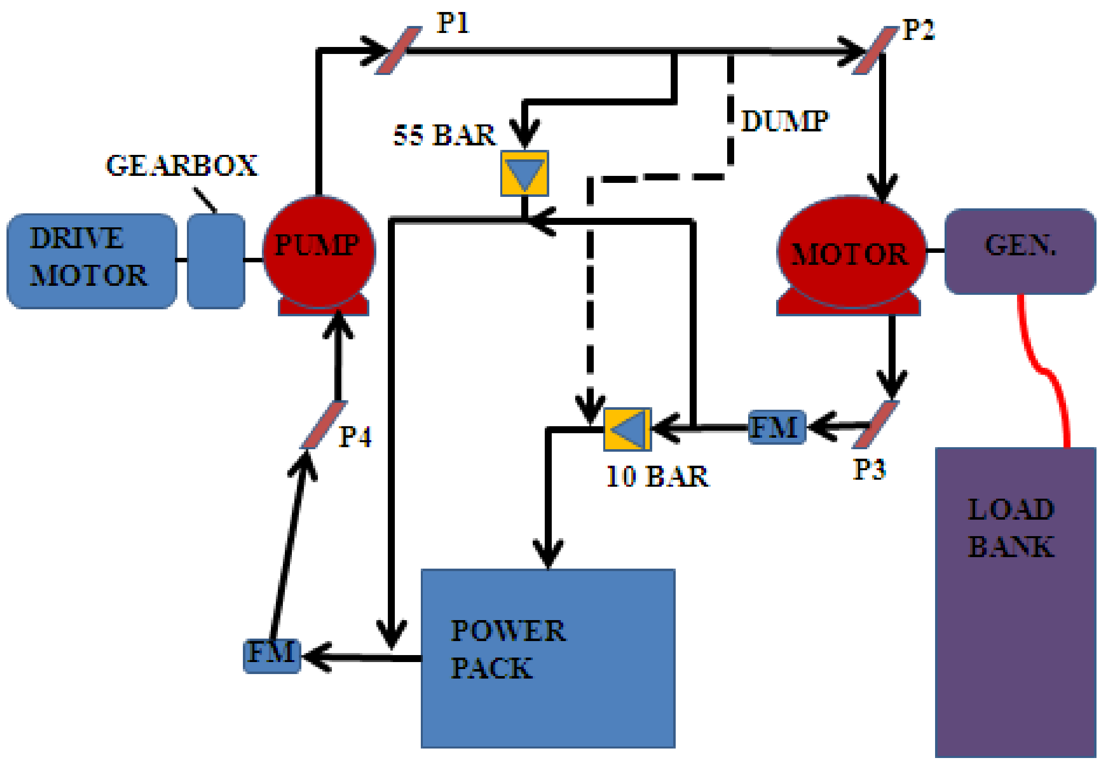


Figure 69: Essentials of the hydraulic circuit.

Figure 69 shows the essential components of the hydraulic circuit employed in power production. Included are the pressure meters (maroon parallelograms labeled P1 through P4) the flow meters (labeled "FM") the two safety circuits with their check valves, and the dump valve circuit, noted with a dashed line.

Power is produced by the prototype through the following sequential, key steps:

1) The electric motor converts electrical energy from the Rutgers University power supply to mechanical motion through use of an armature and standing

magnetic fields. Created torque and rotational motion are transferred via driveshaft to the attached gearbox.

2) The gearbox reduces rotations by a ratio of 14.63:1, and outputs adjusted rotational speeds and torque via driveshaft to the Hagglunds pump.
3) Hagglunds pump converts rotational motion and torque of driveshaft into pressurized fluid through use of pistons. The pistons pressurize the fluid by pressing upon it.
4) Pressurized fluid is discharged from the pump and rushes towards axial piston motor.
5) Pressurized fluid pushes upon axial piston motor’s pistons, converting kinetic energy of fluid into mechanical motion of pistons.
6) Displaced pistons cause rotations upon axial piston motor’s driveshaft via the swash plate, an angular plate which serves as an intermediary between the pistons and driveshaft.
7) The rotational motion and torque generated by the high pressure fluid is then transferred to the Baldor-Reliance generator via the driveshafts of the axial piston motor and generator, joined by LoveJoy couplers.
8) The generator driveshaft is connected to an armature which when passed over the magnetic field in the generator allows electrical current (and therefore load) to be developed and applied, respectively.
9) As the load from the load bank is applied, a resistive torque develops upon the generator’s driveshaft.

10) So long as the axial motor can provide enough torque to overcome this resistance, the load from the load bank is met.
    a. If the load is met, the appropriate amount of power is developed at the load bank as hot air. The hot air is then dissipated via fan and ducted from the room.
    b. If the axial piston motor cannot overcome the resistive torque, the axial motor stalls and stops the entire system.
11) Once the axial piston motor has converted the hydraulic fluid's kinetic energy into rotational motion, the depressurized hydraulic fluid discharges from the motor and is returned to the Hagglunds pump to be pressurized again.
12) Along the return path, the Bosch-Rexroth power pack provides boost pressure to maintain line pressures above the minimum charge pressure for the Hagglunds pump (29 PSI).

Chapter 4.

Experimental Testing, Results and Discussion

This chapter is broken up into five major subsections. The first, procedure, describes the method in which the hydraulic system is started and stopped. In the second, third, and fourth, the phases of construction are outlined, as well as the testing that occurred during each phase. Testing was broken up among each phase to ensure that each series of components worked properly before additional components were installed. In this way, problems with the system could be more accurately isolated and identified, instead of only doing full testing when the system was fully installed.

4.1. Proper Start-Up and Shut-Down Procedure

Operation of a high pressure hydraulics system requires a dedicated and precise start and stop procedure for each component. In the case of this experiment, it is critical that the steps outlined below be followed precisely. It is this order of operations which helps prevent equipment damage and more importantly, personal injury.

4.1.1. Proper Starting Procedure

There are three major sub-systems which must be engaged during start-up: 1) the power pack system; 2) the major hydraulic circuit; 3) the resistive load bank. The correct procedure is outlined as follows:

A) Put on ear protection (ear muffs or in-canal ear plugs) and safety glasses. Ensure all occupants in room are wearing similar protection.

B) Check for leaks: if oil is pooled on ground near fittings, or leak is otherwise suspected, do not continue with start-up. Seek assistance and repair for leak immediately.

C) Throw main circuit breaker box (60 amp service) in rear of room to the ON position. Throw circuit breaker box (10 amp service) for load bank to ON position.

D) Ensure both emergency shut off buttons are fully disengaged (button should be in pulled-out position.) Then, rotate large black handle on power pack/VFD service box to ON position.

E) Start LabView software program entitled JPL_AQUIRE_VI.vi on the data acquisition computer. Begin data recording, as appropriate.

F) Confirm that dump valve is in the OFF position, causing fluid flow only through main circuit.

G) Start load bank by flipping POWER switch to ON position. AIR FAIL light should briefly illuminate, then extinguish. Fan should turn on.

H) Turn on VFD. Set to 20 Hz. Do not engage.

I) Turn power pack switch to ON position. Engagement of power pack will cause Hagglunds pump to rotate 2/3 of a cycle, then stop.

J) Engage VFD by pressing FWD button on screen. Pressure will begin to climb in system, but no motion will be noticed. Distinct hissing noise will develop.

K) Immediately, throw dump valve lever to ON position. Loud clacking and banging should be heard from Hagglunds pump.

L) After 3-5 seconds, or when Hagglunds pump reaches maximum rotational speed, throw dump valve lever to OFF position quickly and firmly.

M) After 3-5 seconds, axial piston pump will begin to rotate. At this point, flip MASTER LOAD switch to ON position on load bank. Flip 2KW switch to ON position.

4.1.2. Proper Shut-Down Procedure

After operating the hydraulic system for a period of time, the shutdown procedure allows for the safest and smoothest depressurization of the system, reducing potential for damage to the machinery. The correct procedure is as follows:

A) Reduce load bank load immediately to 4kW.

B) Reduce the VFD by 10Hz increments, pausing about 30 seconds at each level.

C) When at a minimum of 30Hz, set load bank to 2 kW.

D) Reduce VFD to 20Hz. Allow system to stabilize.

E) Turn MASTER LOAD switch on load bank to OFF.

F) Hit STOP/CLEAR on VFD to completely disengage power to the electric motor.

G) As soon as Hagglunds pump stops rotating, turn power pack switch to OFF.

H) Flip load bank's POWER switch to OFF position.

I) Turn off VFD once VFD cooling fans stop.

J) Rotate large black handle on body of circuit breaker box to OFF position.

K) Depress both emergency shut off buttons.

L) Throw load bank circuit breaker box (10 amp service) lever to OFF.

M) Throw main 60 amp service box lever to OFF.

4.2. Phase One Construction: Electric Motor and Gearbox

Testing of the prototype in this phase only includes operation of the electric motor and gearbox. During this phase, electric service is provided to the motor via the main 60 amp circuit box, the secondary control panel, and the variable frequency drive (VFD). The gearbox and electric motor are directly connected; the Hagglunds pump is not connected, therefore bypassing the hydraulic circuit entirely.

Starting and stopping the motor-gearbox unit is controlled directly via the VFD. The VFD allows control via a "jog dial," a knob which allows adjustment of power output whose frequencies lie anywhere from 0 to 120 Hz. The VFD can also be controlled by amperage, but it has been discovered in operation that the frequency adjustment gives a more stable speed control, the desired parameter to manipulate. Referencing the badge on the electric motor, it is seen that at 60 Hz, 1765 rpms can be achieved on the electric motor output shaft. After a 14.63:1 reduction, the output at the gearbox shaft is 120 rpms. Since the VFD allows for "overdriving" of the electric motor up to 120 Hz, rotations of as high as 240 rpms are possible.

The limiting factor here for operating the motor is the maximum permissible amperage to be pulled from the wall. If the electric motor demands more than 60 amps during operation to maintain a given speed, it will trip the circuit breaker and shut down. However, the amperage pulled has little to do with the actual speed of the electric motor. Rather, the amperage pulled has to do with the torque requirements of the load on the motor. Should the motor be pressed to provide more torque than is readily available, amperage rises to accommodate the demand.

4.3. Phase Two: Integration of Hagglunds Pump, Axial Piston Motor, and Hydraulic Circuit

Phase two of integration was a significantly more complicated affair, involving the installation of the Hagglunds pump, hoses, and filling the system with fluid. Each issue or problem with installation is explored. The solution and future recommendations are provided as well.

4.3.1. LoveJoy Couplers

The LoveJoy couplers are a brand of cast-iron couplers which allow otherwise incompatible shafts to mate. In our experiment, LoveJoy couplers are provided between the axial piston motor output shaft and the generator driveshaft as well as between the Hagglunds pump and the gearbox shaft. Each shaft has three large prongs on it which mate with two components: a brass torque sprocket and the other coupler. The coupler slides onto its respective shaft and is locked into place via a key and slot arrangement, common for this type of coupling. In standard construction, the couplers should fit together (with the intermediate brass sprocket in place) with ease.

However, our couplers did not fit together properly, leaving about 2 inches of engagement completely untouched, as shown in Figure 70. Additionally, only one of the four couplings fit onto its respective drive shafts. The remainder failed to slide on at all, or only did so with great effort. Finally, keys were missing for two of the three key and slot shafts (the fourth shaft is a spline arrangement, and does not need the key and slot lock.)

It was decided to machine the metal couplers, not the brass sprockets, in order to allow for full engagement of the couplers. However, instead of precisely machining the prongs to allow for exact engagement, the prongs were intentionally downsized enough as to provide slack and looseness between the couplings; this allows non-coaxial drive shafts to couple together without issue. In addition, the loose fit allows for shafts which are not precisely concentric; this may occur if mounting holes are too far left or right from center. Additionally, the couplers were bored on their interior channels to allow for looser fitting to the various shafts. The spline faced coupler was not machined.

Alterations were successful and allowed for quick and easy engagement of couplers. In the end, however, it became clear that even with slack allowed in the couplers, incorrect alignment of drive shafts would have significant consequences. In the case of the gearbox and Hagglunds pump, incorrect vertical alignment resulted in a torqueing force upon the gearbox, caused by the Hagglunds shaft axis sitting slightly higher than the gearbox shaft height. This force results in a bobbing effect of the gearbox during slow rotational speeds. Additionally, since the Hagglunds pump axis is canted at a slightly negative angle (about -3 degrees from the horizontal) the couplers slide fore and aft via their prongs, a motion which is not supposed to occur. The couplers come together at the top of the shafts and spreads apart about ½" on the bottom of the shafts. This additional source of friction causes tremendous wear on the couplers, resulting in regular piles of metal shavings on the floor. Since the system is expected to only run for a few weeks, it was determined that this source of friction and wear would be preferable to shimming the Hagglunds motor and/or the gearbox in an attempt to reach better alignment. A long term alignment solution involves raising the gearbox and electric motor units, as well as shimming the Hagglunds motor tank to level it.

Figure 70: (Left) Lovejoy coupling fails to fully engage. (Right) Fully engaged coupler, without brass

fitting.

4.3.2. Failure of the Hagglunds Pump to Rotate

After connecting the Hagglunds pump to the gearbox and establishing the hydraulic circuit, an initial test was conducted. However, an attempt to turn the Hagglunds pump with the electric motor was met with stiff resistance. On every occasion, the Hagglunds shaft would torque, causing the sheet metal tank to which the pump is affixed to rock, but the shaft itself would not turn. It appeared that the Hagglunds motor was seized. Conversations with Bosch hydraulics specialists and Hagglunds service members revealed that several key ports on the Hagglunds pump body may be closed which need to be open, including test ports and crankcase ventilation caps. All were immediately opened, but the pump still failed to turn.

In preparation for shipment to Hagglunds service headquarters for further diagnostic tests, it was discovered that the aluminum mounting plate (which connects the Hagglunds pump body to the sheet metal tank) was binding on the Hagglunds driveshaft's rotating bearing pack. This bearing pack encases the driveshaft port. Removal of the plate confirmed the problem; widening of the plate quickly solved the problem (Figure 71). It was also discovered at this time that the mounting plate in question was missing a key relief feature which if present, would allow the plate to sit flat on the face of the Hagglunds pump. Since the relief was missing, the plate was sitting askew and digging into this feature on the pump face. The cause of both discrepancies was tracked back to a poor drawing schematic from Hagglunds; the blueprints did not provide the proper dimensions of the bearing pack, nor did they provide the dimensions of the feature sitting on the face of the plate.

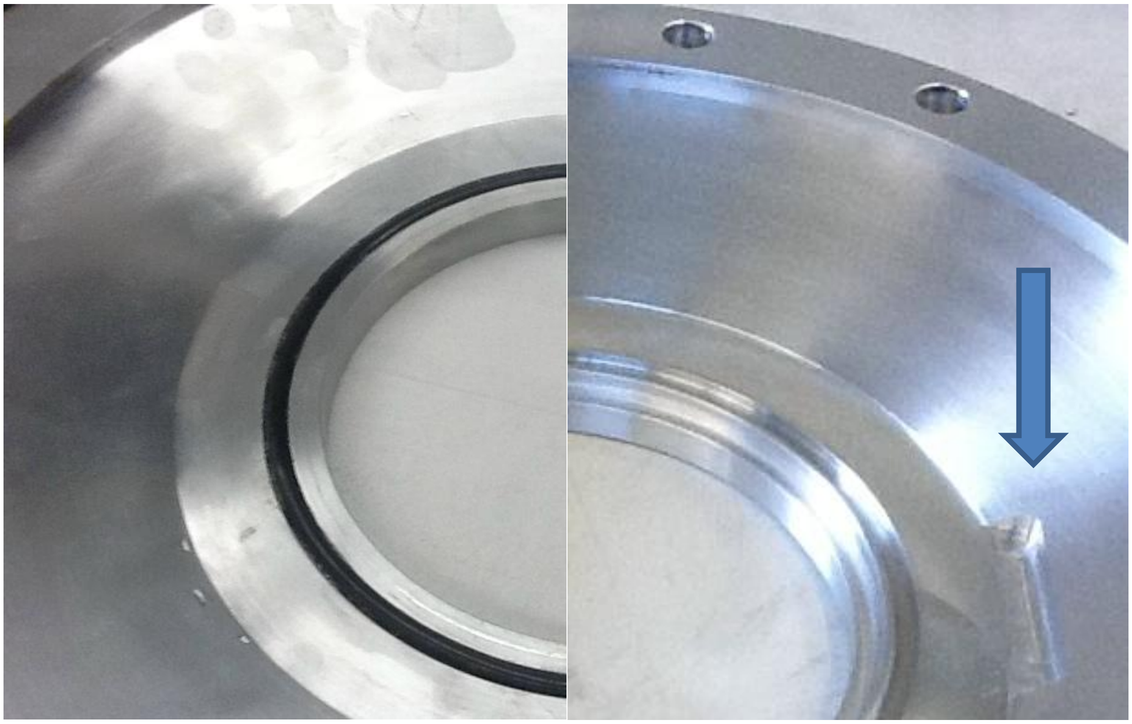


Figure 71: (Left) Original plate, with installed O-ring seal. (Right) Widened, counter bored plate with rectangular relief feature (shown by blue arrow) on right side.

4.3.3. Hagglunds Pump Hesitation and Clatter

Once the Hagglunds pump was re-installed, and the remaining hydraulic hoses reconnected, fluid was added to the system. Initially, Bosch hydraulics recommended that the power pack be started before the pump. After pressurizing the return line, the electric motor was then to be engaged. However, under all variations of VFD frequency, the same result occurred: the Hagglunds pump failed to rotate more than two or three rotations. It appeared that the electric motor did not contain sufficient torque to keep the Hagglunds pump rotating.

The only way to rotate the Hagglunds pump with any measure of success is to bypass the axial piston motor. This is achieved by engaging the dump valve circuit, which according to the hydraulic circuit, causes fluid to flow from the Hagglunds pump to the power pack and back to the Hagglunds pump. In this mode, hereafter called "bypass," the Hagglunds pump will rotate under power of the power pack at a sluggish 5 rpm. If the VFD is engaged at any greater frequency, the Hagglunds pump will quickly accelerate to the appropriate rpm, but shakes violently and clatters loudly. Both operational behaviors, that of needing the by-pass mode to operate and also that of the distinct clatter, worried the hydrokinetics team. Hagglunds reassured that some clatter was normal, but was indicative of very low *charge pressure*. When the return pressure to the Hagglunds pump falls below 29 PSI, clattering develops as the pistons disengage and violently reengage with their respective cam rings. The minimum charge pressure, as outlined in Figure 72, must sit above 29 PSI during most operational periods, but extends as high as 60 PSI during full-speed operation of the hydraulic system.

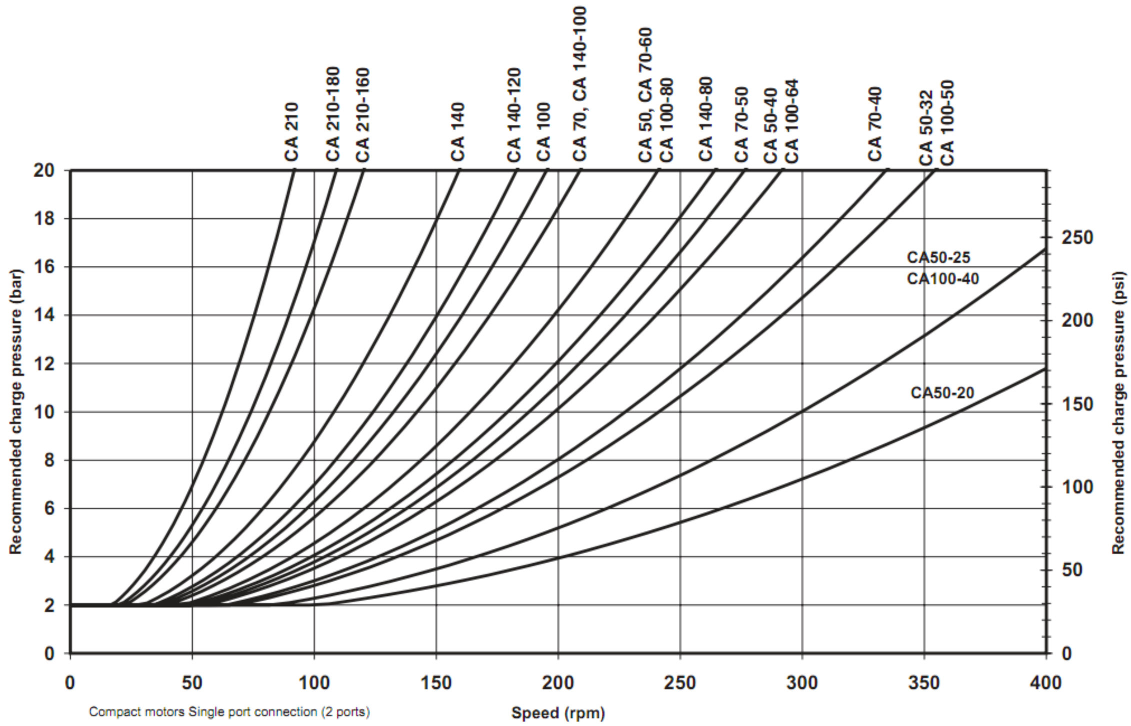


Figure 72: Pressure minimums for the Hagglunds pump. Reproduced without permission [66].

In our scenario, we were "pushing" the pump beyond its available charge or return pressure (provided solely by the power pack). The faster we went, the more of a vacuum developed on the low pressure side. The greater this vacuum, the lower the charge pressure became, and the worse the clatter. Only when the electric motor posed no influence did the system stay clatter free.

During normal operation, the axial piston motor only harnesses a portion of the incoming high pressure fluid's kinetic energy. The remaining kinetic energy in the fluid, upon discharge, contains a certain level of pressure which usually is higher than the charge pressure minimum required by the Hagglunds pump. Adjustment of the power pack can supplement this low return pressure, helping to boost it as needed. However, with no influence provided from the axial piston motor, the power pack was simply overwhelmed and unable to provide the necessary minimum charge pressure as rotational speeds increased.

During this period, we also learned that bypass mode was needed to start the axial piston motor under all circumstances. The inertial load of the axial piston motor is sufficiently great that the Hagglunds motor from rest cannot overcome it. Instead, spinning the Hagglunds motor to a minimum of 40 rpm (20 Hz on the VFD) in bypass mode, and then abruptly engaging the standard hydraulic circuit allows for enough kinetic energy from the pressurized fluid to start the axial piston motor. If a rotational speed of less than 40 rpm is used, the axial piston motor provides enough inertial resistance that the Hagglunds pump will come to a complete stop. Therefore, normal starting procedure involves spinning the Hagglunds pump (in bypass mode) up to 40 rpm, and then quickly throwing the dump valve lever to the OFF position, thereby engaging the axial piston motor after a brief delay. From this point on, the system operates as expected.

The failure of the electric motor to provide sufficient torque can be traced back to the performance curves of the Hagglunds pump verses the performance of the electric motor. The Hagglunds pump, as shown in Figure 73 as “CA 50” demands a maximum torque of approximately 6000 lb-ft from 0 rpm through 200 rpm before beginning to decrease. This torque is used, naturally, to produce high pressure fluid. At maximum torque of 6000 lb-ft, the maximum pressure differential of 5000 psi is generated. For pressure demands below this, less torque is required. At 120 rpm (top output) the electric motor is capable of 875 lb-ft, a value which is far below the 6000 lb-ft rated maximum of the pump. This points to the limited total potential of the hydraulic system, traced back to an underperforming electric motor.

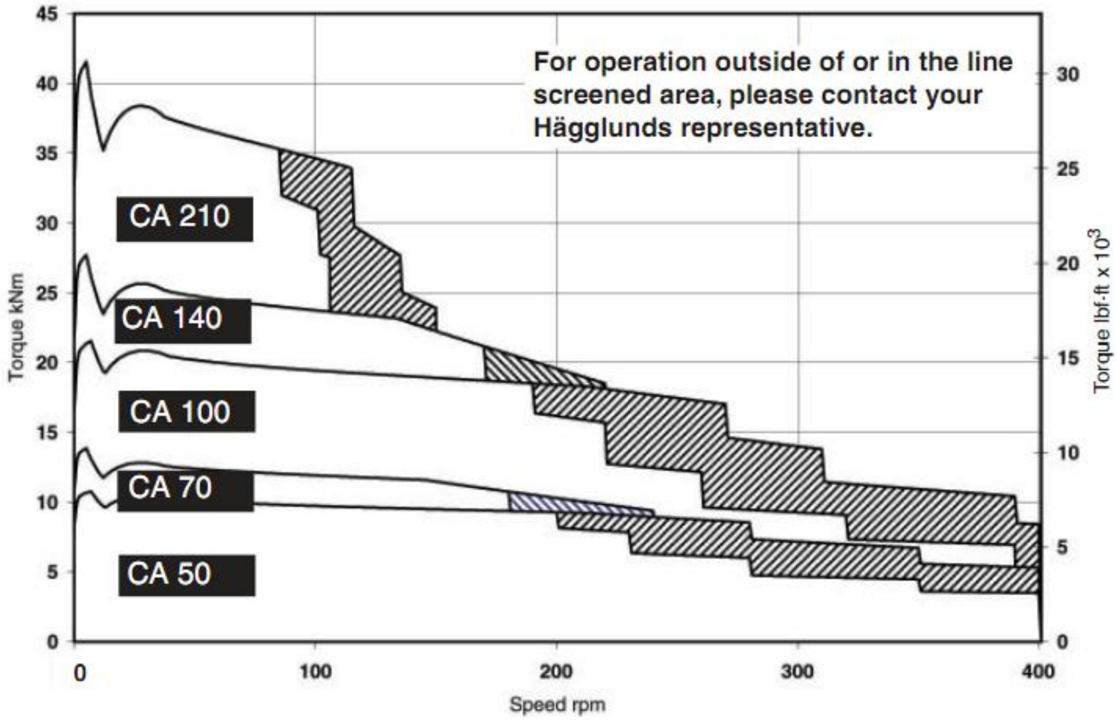


Figure 73: Torque curves for Hagglunds compact motors, CA-series model. Reproduced without permission [66].

The implications of such a weak design are significant for a full-scale system. In such a scenario, the underwater turbine-pump arrangement would not be affected, but the on-land generation station would. In such an arrangement, a precise valve-based flow metering system would bring axial piston motors on-line and off-line as flow rates and pressures change. For example, during the periods when the turbines are spinning slowly and low pressure and flow rates are common, fluid would be funneled into fewer motors instead of the entire bank, keeping a few spinning optimally instead of stalling many. As pressure and flow rates increased, more motors could be brought on-line, provided sufficient line pressure and flow allowed for it. Electricity generation would increase and decrease in a smoother manner because of the valve-based distribution system. Conversely, a system which does not disengage some motors during low flow periods would find the entire system shutting down abruptly when minimum kinetic energy values failed to be met.

4.3.4. Phase Two: Final Test

Once the hydraulic circuit had been fully established, it was important to ensure that the system ran reliably and properly up to 120 rpm, the designated maximum operational condition. During initial testing, however, it became clear that the system had a comfortable maximum of 45 Hz on the VFD. Above 45 Hz, the system exhibited heavy vibrations and some shaking. At 48 Hz, the sound and behavior of the system markedly shifted, becoming louder and harsher, prompting a system shutdown. The test concluded with pressure readings from 20 to 48 Hz, provided via analog pressure gauges installed at the four different pressure transducer locations; the data acquisition system was not yet installed.

The four pressure gauges revealed some interesting system behaviors not expected or anticipated during normal operation. When the system ran from 20 to 48 Hz, the high pressure side exhibited markedly different pressures, with the outlet of the Hagglunds pump always having a higher pressure than the axial piston inlet. This pointed to a line disturbance or loss which affects flow rate. The most probable cause of reduced pressure on the high pressure side is the t-fitting that occurs at the dump valve.

The outlet of the axial piston pump hovered around 90 PSI during all frequencies of operation, but dipped briefly at 40 Hz and then rose at 45 Hz, only to fall again at 48 Hz. This stability is in marked contrast to the rest of the circuit, which tends to increase or decrease in response to system performance. The anomaly at 40 and 45 Hz can be explained by the analog nature of the gauges, which, due to their vague markings, are hard to precisely read.

The final gauge, the Hagglunds inlet gauge, showed declining pressures as the pump worked harder (Figure 74). Such a declining pressure, when the axial piston pump outlet remains steady, points to a starvation condition at the inlet of the pump. As the Hagglunds pump discharges more fluid, it demands an equal amount be returned to its inlet. If the system struggles to meet demand, a vacuum condition begins to occur, causing line pressures to drop. Conversely, if a blockage were to occur, the line pressure would increase dramatically. The readings may be exacerbated by the location of the pressure gauge. Its positioning (at the top of the steel shaft during these readings) is such that fluid is constantly flowing away from it, not near or directly past it. As the fluid velocity increases, the vacuum at the top of the shaft increases.

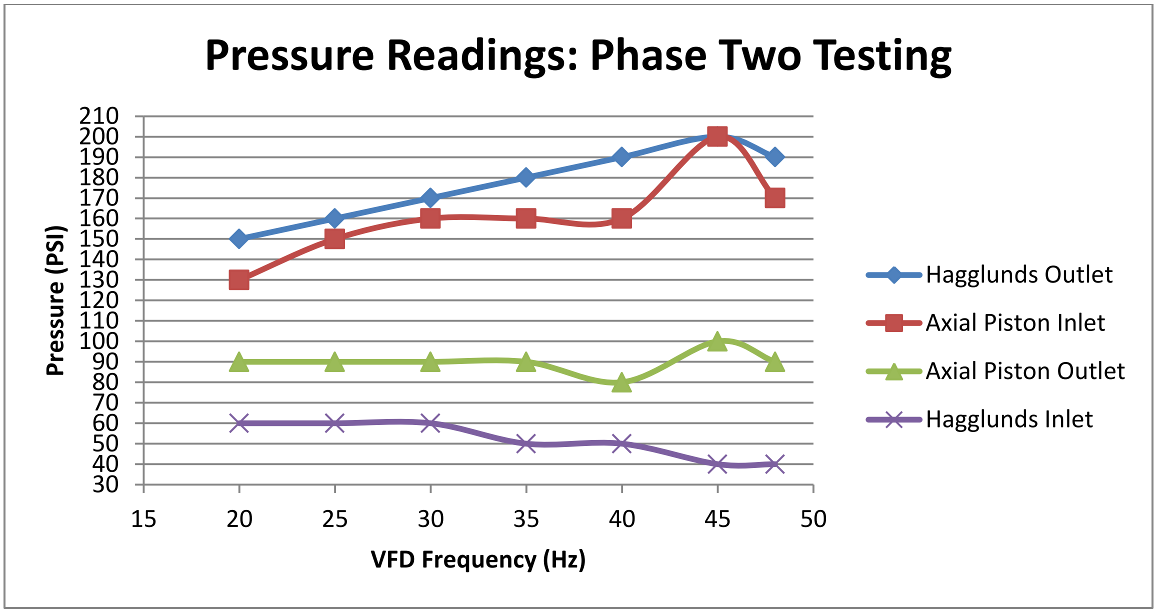


Figure 74: Phase two testing showed low return pressures to the Hagglunds pump, a troubling sign.

Other phenomena observed during operation points to a potential circuit problem, regardless of gauge readings. Namely, at 48 Hz the system began to emit a timbre that was unlike any ever heard before. This change in pitch signaled that the system was not operating normally. Additionally, the Hagglunds pump began to shake lightly, and pulsing felt in the floor became pronounced. Despite the posted high values of charge pressure at the Hagglunds pump, it appears that the Hagglunds pump may have been receiving inadequate return pressure to function normally.

Notes on temperature increase were also taken, an increase which was slight for the hydraulic fluid, hoses, power pack, and Hagglunds pump, but severe for the axial piston motor. The Hagglunds pump exhibited a temperature increase of roughly 37°F, rising from 68 to 105 degrees Fahrenheit over the course of an hour. The hydraulic fluid absorbed enough heat to rise from 68 to 88 degrees Fahrenheit over the same period, but the axial piston pump jumped a remarkable and concerning 72°F over the hour. The motor, pump, and hydraulic fluid all remained within acceptable temperature limits. The axial piston motor had the lowest maximum permissible temperature of the group at 240°F, a value which was never reached or exceeded.

Phase two testing proved that temperature gain and pressure limitations are a serious concern for the future of the project. It became evident that if the system continued to increase in flow rate and pressure, the return pressure to the Hagglunds pump would continue to drop, posing a future dilemma for the project during spirited operation. Additionally, temperature gain without a load applied to axial piston motor showed that the weakest link in longevity would be the axial piston motor. The severe temperature gain showed that only two to three hours of operation would be possible without a cooling circuit in place.

4.4. Phase Three: Integration of Generator, Data Acquisition, and Load Bank

Attachment of the generator to the axial piston motor brings with it connection to the load bank, and the need for detailed data acquisition software to properly operate the hydraulic circuit. The data acquisition system provides real-time feedback from over 10 sensors, including data about flow rates, temperature, pressure, and electrical production and consumption.

4.4.1. Initial Test: Connection and Adjustment of Load Bank

Initial operation of the system with the load bank attached showed that at lower

VFD frequencies, the hydraulic system responded well to increased load from the load bank. The higher the kW switch engaged on the load bank, the heavier the load on the axial piston motor. When too high a load is demanded of the generator, torque levels increase to the point that the axial piston motor comes to a stop, and consequently the hydraulic system stops. Since the electric motor lacks the torque to overcome this resistance by the axial piston motor, the VFD will usually overload, signaling "OL" on the screen and shutting down. The VFD may also flash OL and not shut down immediately, depending on how severe the overload scenario. In these instances, the operator is given a window of time to reduce the load slightly and permit normal operation to be restored before the system stops.

As the system speed increases, the impact of the load bank changes considerably. At low generator speeds, namely in the 500 to 1200 rpm range, the range of permissible load options is wide, with up to 24 kW being permissible from the load bank. However, above 1200 rpm, the range of permissible loads drops to a low of 12 kW at 1900 rpm. Also, the response to load increase is markedly different. At low rpms, an increase in load is met with a reduction in speed, and often, when an overload condition occurs, the VFD will flash OL on the screen and begin to slow down in a manageable, predictable manner. However, at higher rpms, the difference between two steps on the load bank can be the difference between fully functional and immediate system shut down. Therefore, observing system functionality during load increases is critical; at higher rpms, too much load will quickly overwhelm the system.

After considering how the load bank works, and considering how the generator works, the phenomenon is more clearly understood. First is the issue of torque. The generator produces a resistive torque that increases as the load applied upon it grows. This torque must be overcome by the axial piston motor. At lower rpms, the available torque per rotation from the axial piston motor is greater than when it spins faster. This occurs because the available torque remains relatively constant while the revolutions it is spread over increases. For example, if the available torque is 850 Nm, and it is spread over 1000 rpm, the available torque per revolution is .85 Nm. However, if the same torque is spread over 2000 rpm, the available torque per revolution is .425 Nm, about half the original value. Therefore, calls for greater torque are met with a more dramatic drop in performance at higher rpms because the system physically has less torque to provide per revolution.

The second issue is how the generator works. This Baldor-Reliance generator is designed to produce increasing levels of voltage, hitting a maximum of 460 volts in three phase AC from as low as 1800 rpm. Additionally, voltages of greater than 400 are possible above 1200 rpm. Amperage is dictated according to the load, but can hit a maximum of 26.6 at 1800 rpm. Less is available at lower rotational speeds. With this in mind, it becomes clear that throughout most of the operational range of this system, the generator is outputting voltages below the established 460 volts. This has a direct impact on the behavior of the load bank.

When operating at voltages lower than 460, the posted values on the load bank switches are not the actual demand. Those loads posted are only valid for 460 volts. Instead, the kilowatt ratings become much lower. This explains why the load bank can output a posted 24 kW of load on a generator running at 1200 rpm, but only 12 kW of load on a generator spinning at 1800 rpm.

4.4.2. Load Bank Impact on Hydraulic Circuit

During phase three testing, it was learned that operating the hydraulic system with and without a large load has significant consequences on system pressures and flow rates. Two experiments were conducted. In the first, the load bank was set to 2kW at 20 Hz on the VFD; at values above 20 Hz up to 60 Hz, the load bank was held at 4 kW. 4kW was not an acceptable load for the 20 Hz position because of excess loading of the axial piston motor, which would have caused a stall. Data is collected after setting the VFD to a specific Hz reading, increasing in increments of 5 from 20 to 60 Hz. The average values for each frequency level are included in the graphs.

The increase in pressure for the discharge of the Hagglunds pump hits a maximum of nearly 285 PSI at 60 Hz, as shown in Figure 75. The inlet of the Hagglunds pump hits a minimum of 32 PSI, which is dangerously close to the minimum charge pressure on the pump. During testing, it became clear at higher pressures that continuing beyond this point would not allow safe charge pressures for the Hagglunds pump, and testing was therefore aborted at 60 Hz.

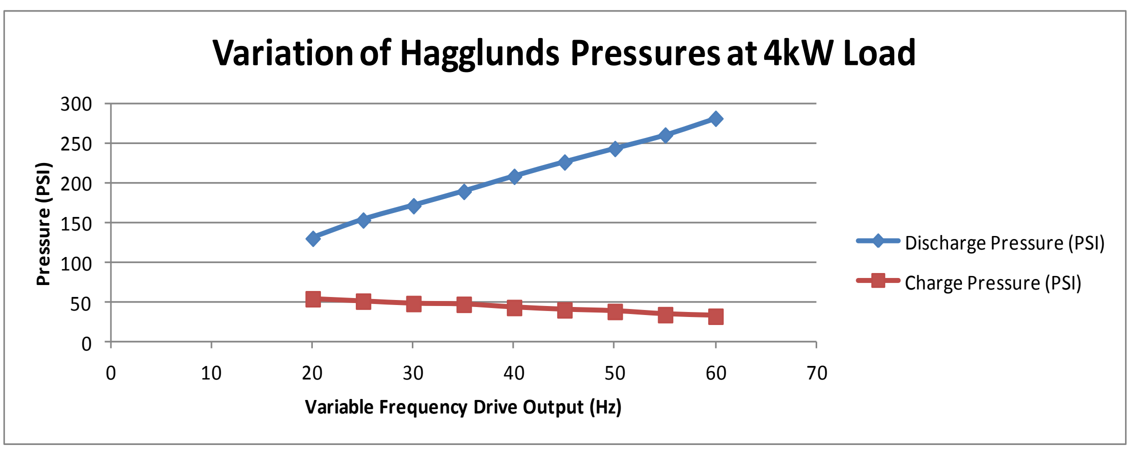


Figure 75: Low loading of axial motor produces less pressure at Hagglunds outlet.

Additionally, testing at a 4kW load provided very poor efficiency readings. During spirited operation, the electric motor pulled at most 11 kW of power from the grid, but the generator only produced a meager 1kW of electricity (Figure 76). The efficiency of operation is a very low 9%. The low efficiency is a product of poor utilization of generator capability since we know that at 60 Hz, the generator is capable of managing at most a 12 kW load.

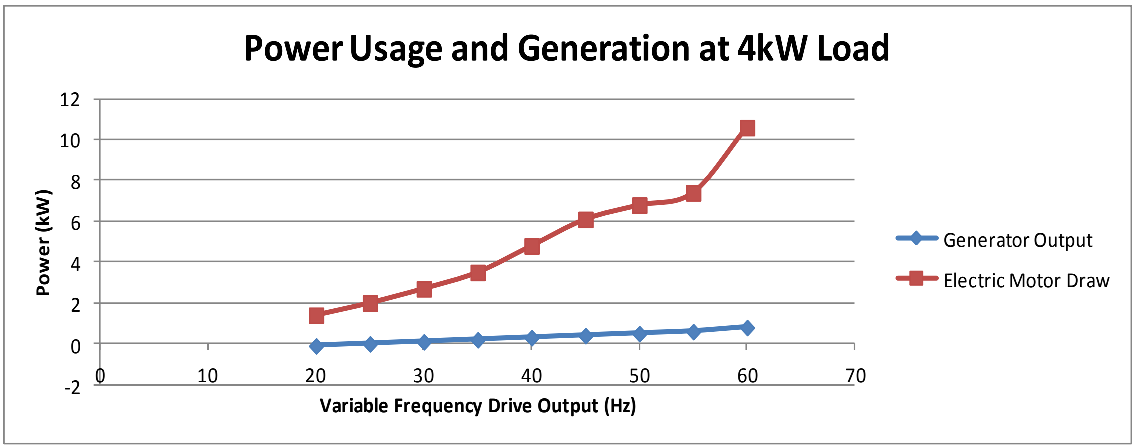


Figure 76: System efficiency drops as speed increases with a 2kW load.

The second test was conducted differently, but gives equally meaningful results. Instead of testing at discrete output values of the variable frequency drive, an effort was made to push the system to 75 Hz as quickly as possible with a 12 kW load attached. The data presented is not discerned by Hz, but by time. Over the course of nine minutes, the hydraulic system was increased to 75 Hz. System pressures varied greatly from those seen in the first test, hitting a maximum of 475 PSI at 75 Hz (Figure 77). At 60 Hz, it was noted that inlet pressure was 425 psi, an increase of almost 150 psi over a barely loaded hydraulic circuit. Return pressure, however, dropped dangerously low to 28 psi, causing significant shaking and vibration from the Hagglunds unit.

It can be concluded from the tests that increased loading causes an increase in pressure on the high pressure side and reduced return pressures. Also, it's clear that return pressures this low are not sustainable, as they fall below the recommended minimum charge pressure of 29 PSI as outlined by Hagglunds.

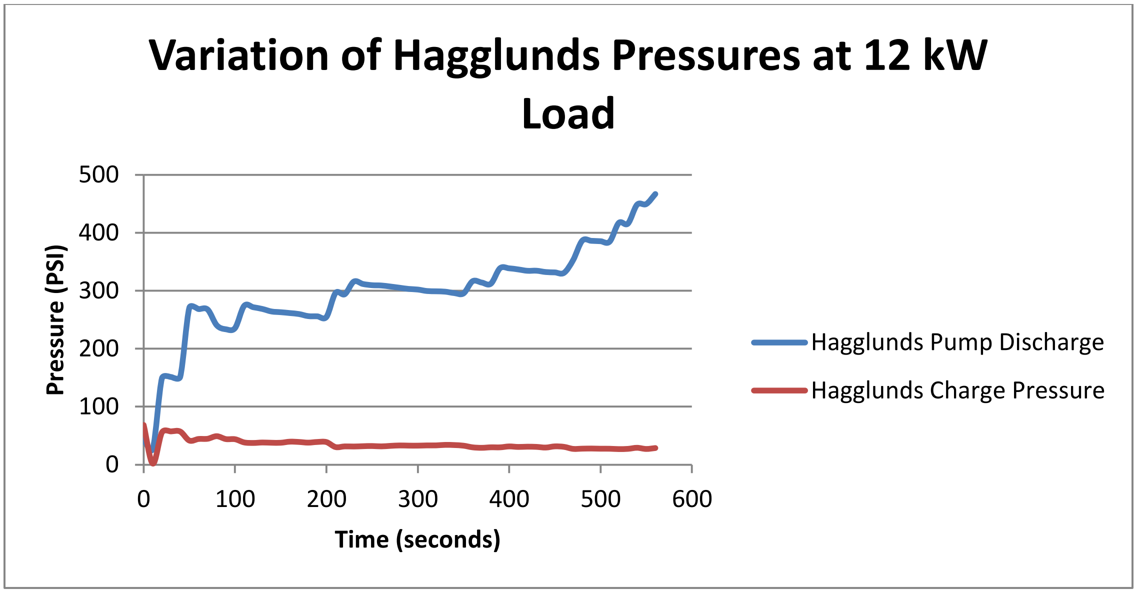


Figure 77: Higher loads on axial motor contribute to drastically increased pressures on Hagglunds output.

4.4.3. Effect of Power Pack on System Pressures

Bosch hydraulics division required installation of the 20 gallon Bosch-Rexroth power pack under the auspices that such a unit provides a needed pressure boost on the low pressure side of the system during start-up. In this scenario, the axial piston motor is not yet discharging high pressure fluid, so the Hagglunds pump is starved for fluid pressurized to a minimum of 29 psi.

Originally, it was thought that the power pack would only be useful during low pressure start-up, and that after the axial piston motor begins to turn, the power pack would become unnecessary. However, a quick test during operation found return pressures to collapse when the power pack was turned off. Therefore, it became clear that the power pack exerts influence on the system at all times.

The power pack employs a 15 bar check valve at its outlet port to help boost pressure inside the unit. Simply put, the power pack retains fluid inside the tank until at most 15 bar of pressure is reached, at which point the fluid discharges. A simple test was conducted to see the effect of adjusting the power pack check valve. The test was started with the check valve fully engaged, at over 15 bar tolerance. The check valve is engaged via an adjustable screw, and was screwed in as far as possible. The notes describe the occurrences of the system as the screw is released:

1) Check Valve fully engaged: Hagglunds pump outlet pressure jumps to 1200 psi immediately. Return to Hagglunds jumps to 900 psi. Hagglunds pump lurches forward then stops abruptly. Power pack motor then seizes and circuit breaker trips.

2) Screw backed off 5 turns: Hagglunds pump lurches forward again, then stops. Power pack motor seizes again, and circuit breaker trips. Outlet to Hagglunds rises to 800 psi, while inlet rises to 650 psi.

3) Screw backed off 7 turns: Discharge of Hagglunds rises to 365 psi, while return is 475. Power pack motor does not seize, but system cannot start normally.

Pressures are too high for Hagglunds pump and electric motor to overcome, even in bypass mode.

4) Screw backed off 8 turns: Discharge and return are lower than at 7 turns, as are system pressures in general. System operates, and pressure data is recorded.

5) Screw backed off 10 turns: Discharge and return pressures drop again, but this time are closer to original, non-adjusted levels.

The differences are stark. When the power pack is left as factory delivered, return pressures are quite low at near 30 psi, as evidenced by Figure 78.

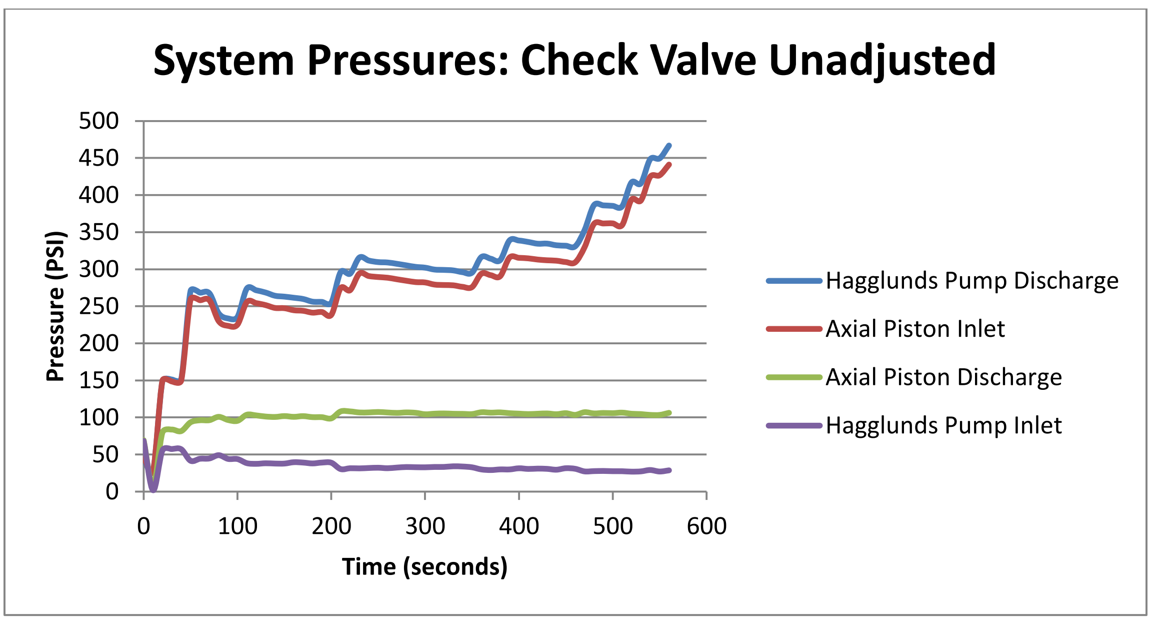


Figure 78: System pressures before check valve adjustment. Hagglunds return pressure becomes critically low.

In both the 8 turns and 10 turns instances, the load bank was set to 12 kW as soon as possible, and pressures are increased greatly. At 8 turns open, the return pressure to the

Hagglunds pump is near 200 psi, as evidenced by Figure 79.

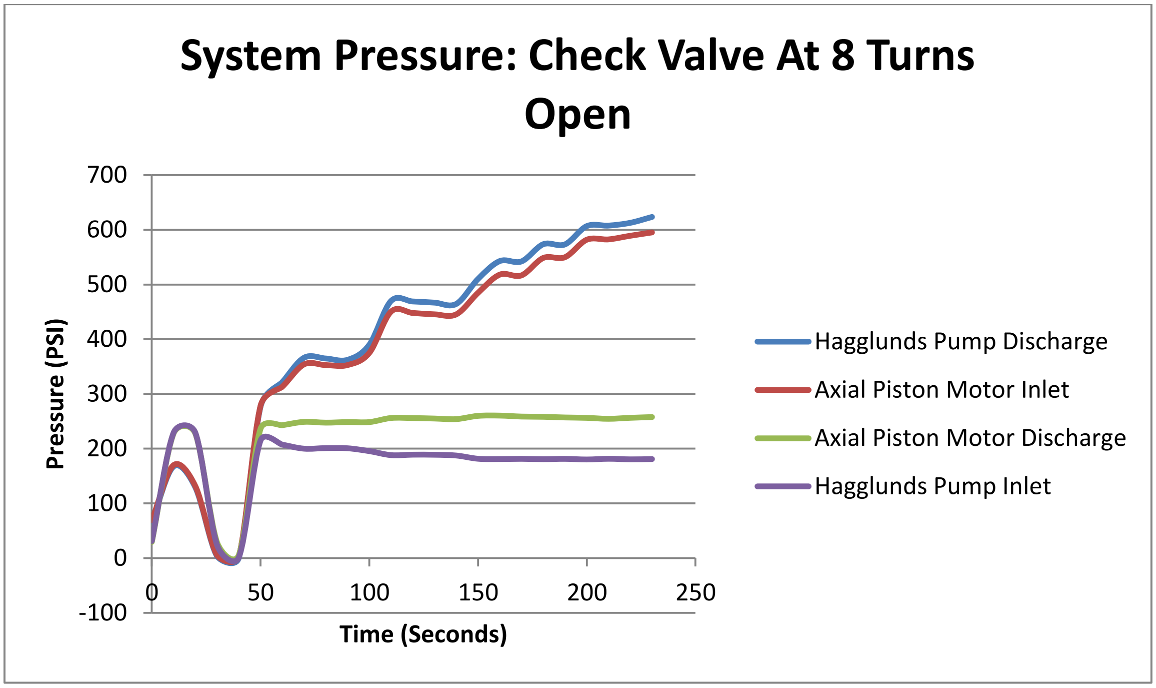


Figure 79: Charge pressures meet minimums handily. Overall system pressures are much higher than before.

Such high return pressures are beyond the intended capability of the power pack, whose aim is to boost pressure only as much as is needed to ensure proper operation. In other words, the power pack was never intended to be a replacement for the Hagglunds pump, for example. Rather, its aim is to boost charge pressure to the minimum needed values. Releasing the check valve an additional two turns provides the following pressure data (Figure 80), which is lower yet, but still much higher than at the original settings.

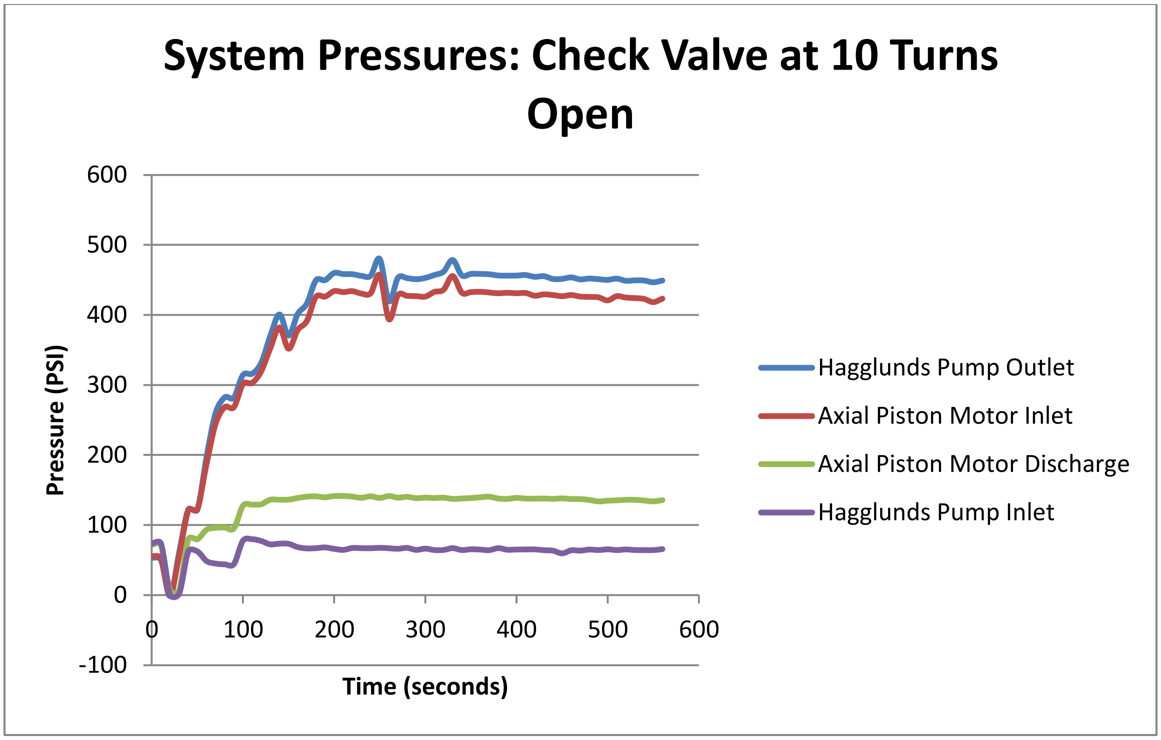


Figure 80: Pressures are still significantly higher than for unadjusted check valve. Minimum charge pressure exceeded by over 40 psi.

4.4.4. Effect of Power Pack on Flow Rates

In short, increased pressure from the power pack has a positive impact on flow rates, although the impact is slight. When the power pack is operating in a low boost pressure mode, as originally set, maximum flow rates occur at about 75 gallons per minute, but trend near 70 gallons per minute on average (Figure 81).

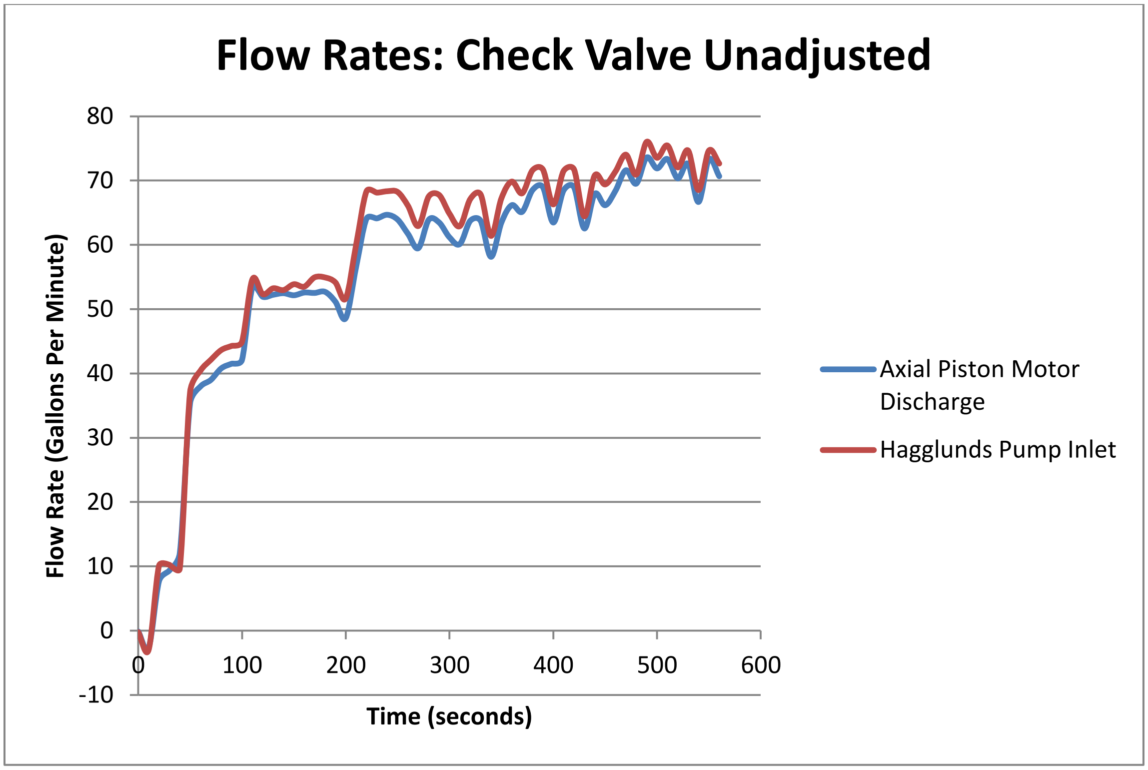


Figure 81: Flow rates peak at near 75 gallons per minute despite low system pressures.

When system pressure is moderately boosted, as seen by 8 turns off the check valve, maximum flow rates rise but only slightly (Figure 82). Flow smoothed out dramatically, but again peaked at 75 gallons per minute. The smoother flow rate, however, helped final high pressure trends to sit well above 70 gallons per minute. When the check valve is opened to 10 turns, there is little overall change compared to 8 turns open, despite a lower system pressure.

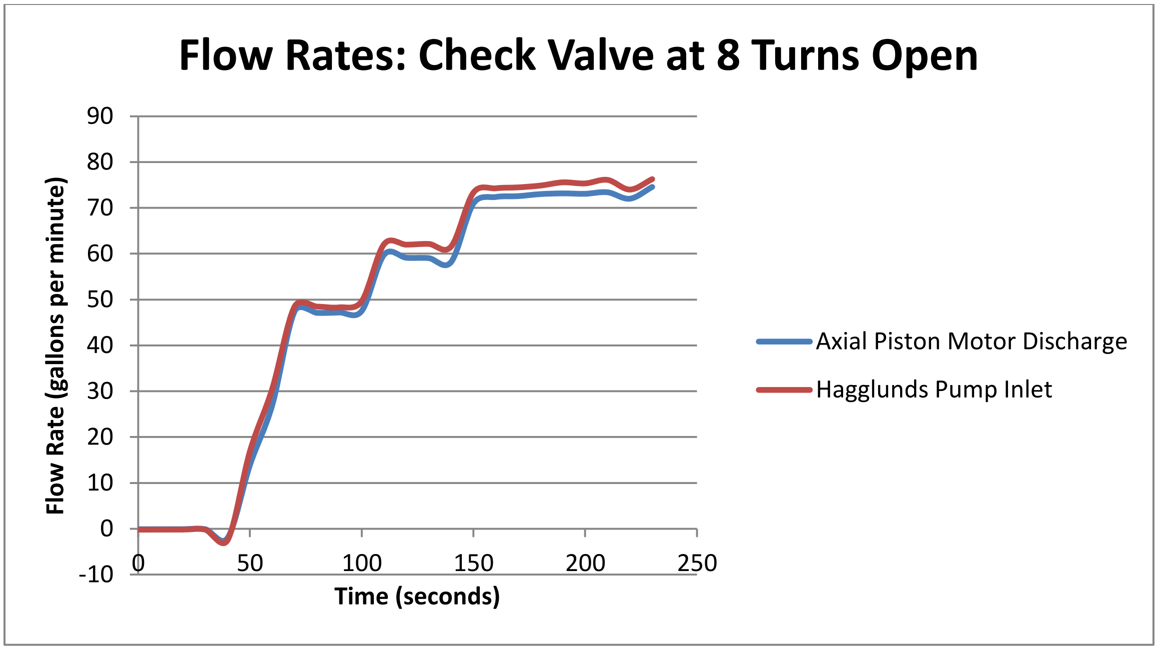


Figure 82: Despite having a minimal impact on flow rate gain, increased pressures did smooth out flow.

4.4.5. Power Pack Effect on Power Production and Consumption

The differences in electrical draw between varying system boost pressures is negligible. The system both generated and drew similar amounts of power for low boost and high boost scenarios, as evidenced by Figure 83 and Figure 84. The electric motor peaked at about 20 kW of power draw, while the generator never topped 5.1 kW of electricity production.

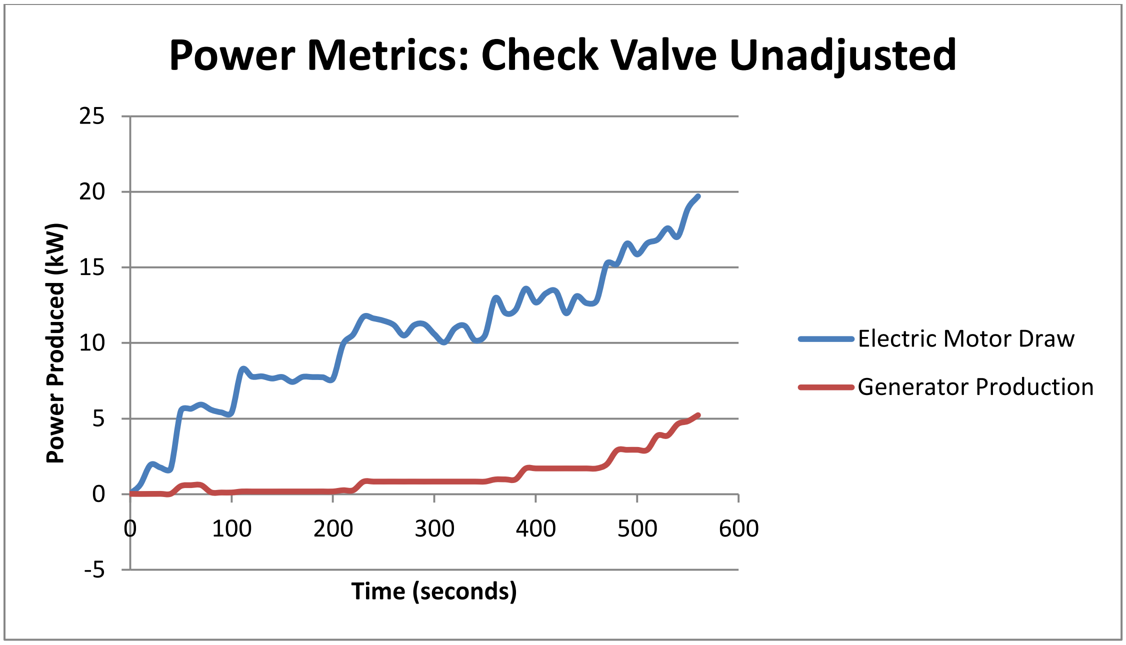


Figure 83: Electricity production and draw for a low boost scenario.

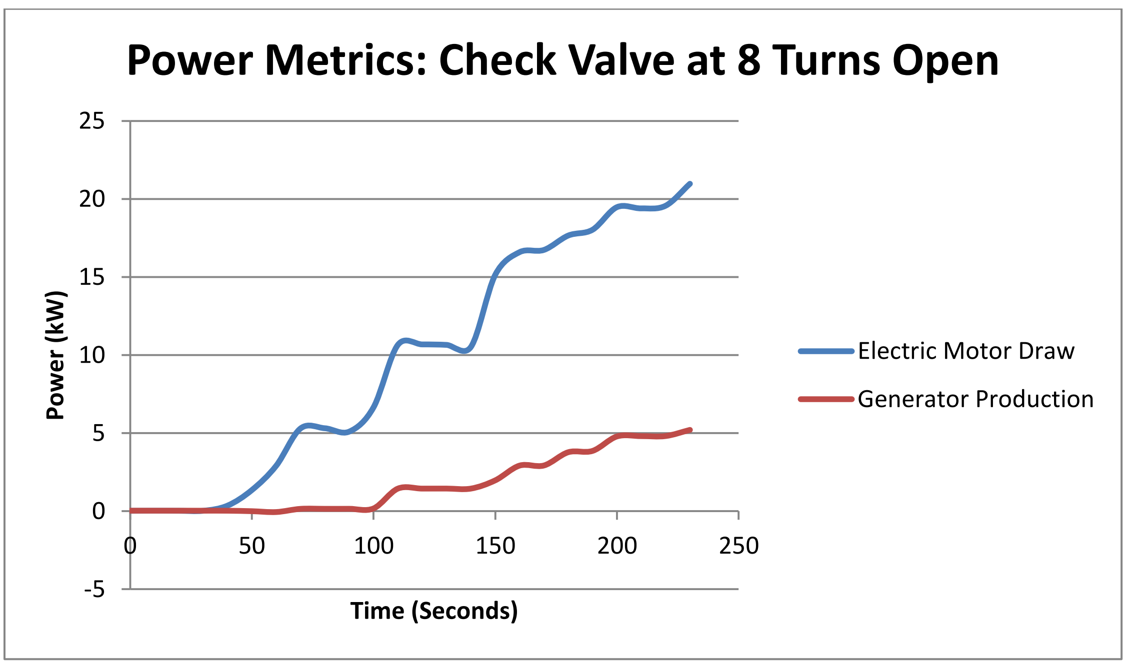


Figure 84: Electricity production and draw for a high boost scenario. Metrics are similar to the low boost scenario.

4.4.6. Effect of Check Valves on System Operation

During phase two operation, a distinct gap in flow rates was noticed between the two flow meters, prompting consideration as to why flow was being lost, as shown in Figure 85. In the graph, the flow returning to the Hagglunds pump was considerably lower, by about 10 gallons per minute, than the flow leaving the axial piston pump. Originally, it was thought that surely the flow meters must be defective; a hallmark behavior of positive displacement pumps which work in tandem is neither a gain nor a loss of fluid over time. In this instance, a steady loss of 10 gallons per minute seemed to theorize that over one minute, over 10 gallons of fluid were being lost from the system. In reality, the meters measure intensity of flow; the kinetic energy in the flow, after leaving the first flow meter, was being lost before meeting the second flow meter.

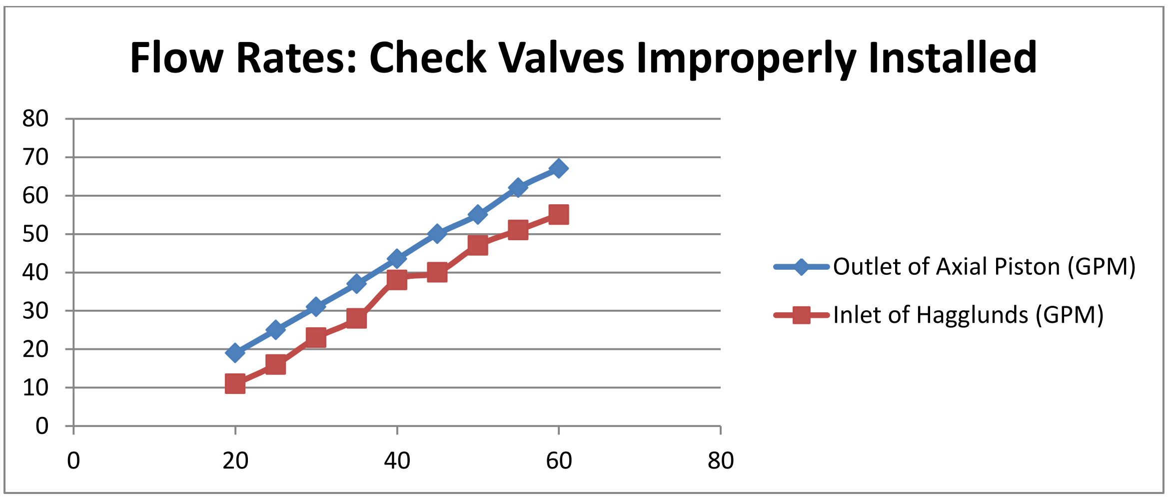


Figure 85: Significant loss of flow is interpreted as kinetic energy loss through power pack.

After reviewing the hydraulic circuit however, it became clear that both check valves were installed improperly and were reversed. A reversed DBDS-series check valve can allow some fluid to pass through by design. When installed properly, however, no fluid can pass through until the minimum pressure is met. In the case of our check valves, a barrier as great as 10 bar or 55 bar, depending on the check valve, must be met.

Reversal of the check valves (to the correct orientation) showed that indeed significant fluid flow was being lost through the 10 bar and 55 bar check valves, as shown in Figure 86. While it was not determined which check valve was the precise culprit of kinetic energy loss, it can be presumed that the 10 bar check valve, by virtue of location and lower pressure threshold, would have contributed more significantly. Its located is such that a strong power pack inlet vacuum can extract fluid through the check valve as needed to replenish the reservoir. This continual process reduces direct flow back to the Hagglunds pump, and instead diverts a portion of flow through the power pack, where a significant amount of kinetic energy is lost.

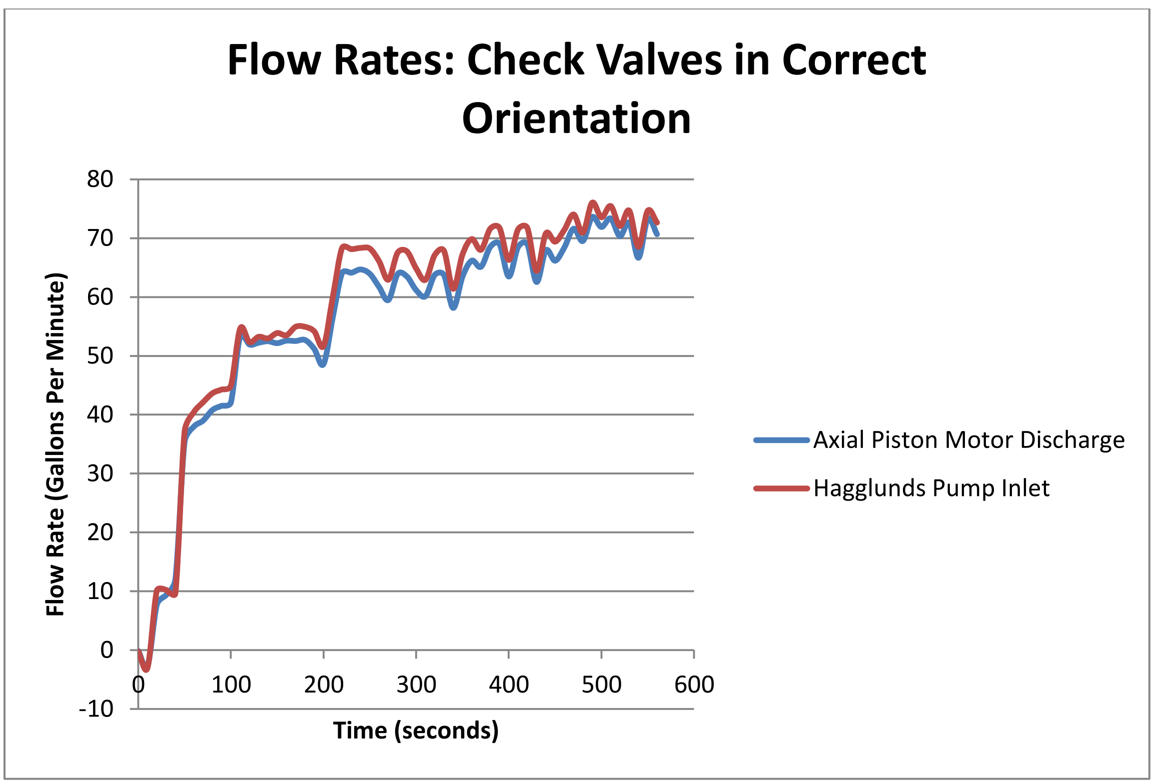


Figure 86: Corrected check valves provide near identical flow rates through both meters, as intended.

4.4.7. Generator Performance

According to Baldor-Reliance, the generator in this experiment is capable of 460 volts in three phase AC at 1800 rpm, but can establish over 400 volts at greater than 1200 rpm [69]. However, during testing, it became clear that the generator was actually producing far lower voltages and amperages than initially expected. A fault in the generator is suspected, and has temporarily halted testing until a resolution can be found. Should the generator be brought up to proper voltage, overall power could greatly increase, as shown in Figure 87.

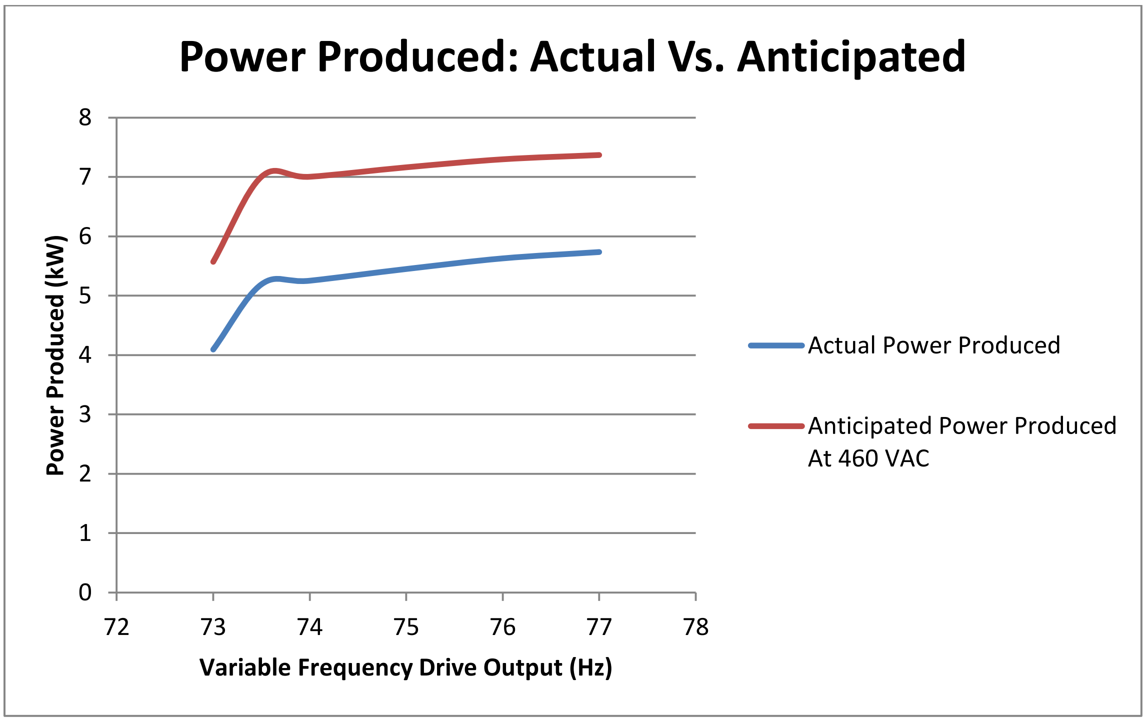


Figure 87: Power produced by the generator could rise significantly if 460 volts is achieved.

Truly, system efficiency is the single most important parameter driving this experiment. Efficiencies which are too low render the experiment and concept irrelevant to the goals of alternative energy. The highest efficiencies of the generator, however, are only achieved at or above 1800 rpm, a revolution speed which is difficult for the hydraulics system to reach. The hydraulics system can only reach 1800 rpm at 70 Hz, at which point the system is clearly operating in overdrive mode.

Overdrive mode has a number of severe and important consequences. Since the circuit breaker box, the VFD, and the electric motor were all only designed to operate at most up to 20 horsepower at 60 Hz, any operation beyond this point becomes excessively stressful on all components. At 70-75 Hz, the electrical conduit, circuit breaker boxes, and VFD begin to warm up.

After about 10 minutes of operation in the 70-75 Hz range, the VFD commonly overheats and shuts down as a thermal protection. The system also frequently trips the main circuit breaker for drawing more than 60 amps for the electric motor and power pack. This excessive electrical draw is indicative of a system which is operating beyond its intended range.

4.5. Discussion of System Operation

Assuming the generator is repaired and brought back on-line, there are still additional fundamental problems with the experimental design which must be rectified before the system can be safely operated for an extended period of time.

Heat generation by the axial piston motor must be addressed, whether through cooling baths, a radiator circuit, or modification of operating style. After operation for a half hour, the axial piston motor can experience temperature gains of over 100°F, and in the 70-75 Hz range, operates in a manner which suggests an unsustainable increase in temperature over time. Temperatures in excess of 240°F would cause damage to the bearings and internal componentry of the motor.

Poor alignment of the Hagglunds motor and gearbox axis is another source of concern. Excessive wear at the couplers will eventually lead to their failure, prompting an expensive repair. Additionally, misalignment at the couplers is indicative of stresses and forces being incurred at the gearbox and Hagglunds pump which may shorten their operational lifespans. In the long term, the health of the gearbox and Hagglunds pump are compromised with misaligned drive shafts.

Generator operation is yet another point of discussion and concern. Fundamentally, the generator is designed to operate optimally at 1800 rpm, a speed which the axial piston motor can only reach while the hydraulic circuit operates in overdrive mode. This is not a safe operating mode, and must be avoided at all costs for the longevity of the various components and safety of operating personnel.

Many options exist in how to remedy this situation. The first and easiest provided the parts exist is to change the generator out with a generator whose top efficiency is reached at a lower revolution range, preferably around 1200-1500 rpm. The generator would operate in a zone which does not overstress or overdrive the hydraulic circuit and electronic components.

The second option is to change out the axial piston motor for the next size down motor. One key issue with the axial piston motor is its large displacement. While ideally the motor would turn at 1800 rpm while the Hagglunds pump is within the 20-60 Hz zone, it hits 1800 rpm at 70 Hz, a value which is simply too high. Even downsizing just one size of the axial piston motor, from the 9.79 in^3/rev displacement model to the 7.63 in^3/rev model would put the 1800 rpm speed at less than 120 rpm for the Hagglunds motor, well within normal operating conditions [113].

The third option is to change the gearbox ratio. Since the hydraulic circuit works reasonably well at 1800 rpm for the axial piston motor and 140-150 rpm for the Hagglunds pump, changing the gearbox ratio to a smaller ratio may help reduce the speed of the electric motor. For example, switching from 14.63:1 to 10:1 would make the top rotational speed of the gearbox and electric motor combination at 176 rpm at 60 Hz. This would keep the electric motor from having to work in overdrive mode to achieve 1800 rpm at a desired 140-150 rpm.

Chapter 5.

Conclusions and Future Work

5.1. Conclusions

The primary aim of this research project has been to assess the feasibility of tidal hydrokinetic systems (with hydraulic energy transfer) as a reasonable alternative to non-renewable energy sources. Work was undertaken in learning about different pumps and their operational performance. It has been learned that the radial piston pump used in this circuit performs as desired under most circumstances, but requires use of a pressure booster on the low pressure circuit to operate safely, and operates poorly if its inlet pressure is below 29 PSI. The axial piston motor has proven to be a motor without such pressure stipulations, but it spins too slowly for our needs and has a great inertial load at start-up which must be overcome before the hydraulic circuit can function as intended. The axial piston motor also becomes quite hot during operation (200°F at 75 Hz) and is a source of concern for long term operation as it may drive hydraulic fluid temperatures too high if not monitored and modulated.

The role of the Baldor-Reliance electrical generator within the hydraulic circuit was explored. When unloaded, the generator has little impact on the hydraulic circuit, and system pressure overall remains low (Hagglunds discharge of 250 psi). However, as load develops on the generator, this torque requirement causes pressures to rise dramatically in the high pressure circuit (over 450 psi) and causes pressures to drop on the low pressure circuit. Loading on the generator is caused by the attached Avtron K595 resistive load bank. The loading behavior of the hydraulic circuit at low speeds is different than at higher speeds. At low speeds (less than 1500 rpm for axial piston motor) a broad flexibility in permissible loading levels exists, whereas higher rotational speeds (1500-1900 rpm) see a narrower range of permissible loads. Additionally, at low system revolutions, loads which are too high (greater than 24 kW) may cause the system to slow and eventually stop, giving as much as 15-30 seconds before stalling to allow for adjustment. At high system revolutions, an overloaded hydraulic circuit may abruptly shut down within one to three seconds.

5.2. Future Work

Operation of the system uncovered critical design issues which need to be rectified before true efficiency calculations can be undertaken. Thus, the future work is comprised of short-term system repairs, which include:

1) Repair of the generator. Currently, even when operating in the 1800-1900 rpm range, as suggested by the manufacturer, the generator fails to produce the rated 460 volts. Without proper voltage, true interaction and behavior of the generator and hydraulic circuit cannot be determined.

2) Adjustment and alignment the drive shafts of the gearbox and radial piston pump. Currently, the misaligned drive shafts are causing excessive wear at the LoveJoy couplings, a condition which will not correct itself. For continued long term operation, alignment will need to be undertaken between the radial

piston pump and the gearbox to ensure that such excessive wear does not continue.

3) Assessment of the root cause and possible solution to the heat gain in the hydraulic circuit. The hydraulic circuit currently operates in an unsustainable fashion, as it tends to heat excessively over the course of loaded operation. It appears that the source of heat gain is the axial piston motor and in remedying the situation, oil coolers or a resized motor may be necessary.

4) Resize or provision of additional cooling to the VFD. During high speed operation, the VFD has shut down on more than one occasion for thermal overload, in situations which were otherwise deemed normal operating conditions. Therefore, a more robust solution needs to be considered.

It has also been determined that modification of several prototype areas would help optimize performance. Long term suggested modifications to the prototype include (but are not limited to):

1) Alternative positive displacement pump selection. Currently, the Hagglunds pump requires a power pack to operate properly. Since a full-scale system will need to operate in the least complex way possible, all effort should be made to choose a pump which does not require pressure boosting of its inlet fluid. A hydraulic circuit should be developed which demonstrates functionality of the system without use of a power pack.

2) Different, lower inertial-load, and cooler-running motor. While the axial piston motor has been a quiet motor, its operation requires use of a by-pass valve and pre-pressurization of the hydraulic circuit before it can be engaged. Efforts should be made to choose a motor which forgoes this operational characteristic. Furthermore, a motor which produces less heat during operation and allows the system to operate in a sustainable manner would be ideal.

3) Reduction of hydraulic system complexity with removal of bypasses and tee-fittings. As a more streamlined hydraulic circuit, lower losses would translate into greater efficiencies and would more closely mirror intended full-scale design.

4) Resized motor and generator unit such that peak efficiency is achieved in a broader system operating range. Currently, generator peak efficiency is only achieved (greater than 1800 rpms) when the system is running beyond intended upper limits, taxing the electric motor and hydraulic circuit unnecessarily.

5) System reversal techniques. In a full-scale system, the hydraulic circuit will need to allow the turbine to rotate in both a clockwise and counterclockwise fashion. Currently, the use of check valves and a complicated flow path do not allow for flow reversal. This could be remedied with the use of a gearbox that

can engage a reverse gear. In this manner, for bidirectional input shafts, the output shaft would always turn in one direction, mitigating the issue. Studies into durability and efficiency losses would then need to be conducted.

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