Energy Harvesting Systems Design for Railroad Safety

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ENERGY HARVESTING SYSTEMS DESIGN FOR RAILROAD SAFETY

by

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A THESIS

Presented to the Faculty of
The Graduate College at the University of Nebraska
In Partial Fulfillment of Requirements
For the Degree of Master of Science

Major: Mechanical Engineering

Under the Supervision of Professor Carl A. Nelson

Lincoln, Nebraska

August, 2011
Railroad grade crossings are locations of significant interest for prevention of collisions, injuries and fatalities. Another area of concern is derailments resulting from mechanically deficient track structures. Many of these incidents occur in remote areas due to the lack of electrical infrastructure to power automated warning systems and/or track health monitoring sensor networks.

Providing electrical infrastructure to railroad crossings in remote areas is often not economical, and other alternative sources of electricity such as solar and wind energy are not reliable or robust. This motivated development of devices for in situ energy harvesting.

Initially, an improved mechanical energy harvesting device capable of harnessing the vertical deflection of the railroad track due to passing railcar traffic was developed. It is mounted to and spans two rail ties and converts and magnifies the track’s entire upward and downward displacement into rotational motion of a PMDC generator. Simulation and testing results of the device show the capability of harvesting power on the order of 40 Watts under loaded train conditions traveling at 60 mph.

If the vertical deflection of the railroad track is small for any reason such as freezing weather conditions and/or unloaded train cars, the mechanical device is constrained to very small amounts of output power levels. This motivated the design of a
device generating electricity by passage of each train wheel. The device consists of two ramp-levers oscillating with cycloidal motion with passage of each train wheel, independent of the direction of the train, and a mechanism to return the ramp-levers to their initial positions.

A hydraulic system was also designed and built, addressing the mentioned shortcomings and limitations of previous approaches. A hydraulic cylinder is mounted under the bottom of the rail such that it directly harnesses the downward deflection of the rail due to railcar traffic. The hydraulic cylinder is compressed and relaxed by passage of each railcar, forcing the hydraulic fluid towards a hydraulic motor and converting the hydraulic pressure and flow into rotational motion and torque. The rotational motion is scaled appropriately to drive a PMDC generator.

This thesis describes the design, development, and testing of these devices. The intent is that this work represents a significant step towards safer and more robust railroad track systems.
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Chapter 1. Introduction

Every year, hundreds of fatalities occur at unprotected rail crossings. The most common types of railroad accidents are collisions with passenger vehicles. On average every 90 minutes a collision occurs between trains and other vehicles at railroad crossings in the United States [1]. Derailments not involving other vehicles also occur due to improperly maintained tracks and mechanical failures, and if a freight car is involved, there is a possibility of environmental contamination if hazardous material is being transported. Many vehicle/train crashes occur at grade crossings in remote areas where electrical infrastructure is not available to power automatic signal equipment and track health monitoring sensor networks. It has been determined that deploying warning light systems at grade crossings and utilizing distributed sensor networks that monitor railroad track health can lead to enhancement of railroad safety in addressing these failure modes. However, due to the lack of reliable electrical power and the high cost of providing electrical infrastructure in remote areas, use of distributed sensor networks and installation of lighted warning systems are impractical for a significant fraction of these remote sites.

Although there are several other possible methods to supply electrical power to warning lamps and other safety implements at railroad grade crossings in remote areas, such as solar energy, wind energy and rechargeable batteries, the lack of reliability and robustness associated with each of them are the major drawbacks of deploying them as a source of energy in remote areas. Accordingly, developing a long-term, low-maintenance, low-cost and efficient electrical power generation plan will facilitate the ability to deliver safety benefits to more remote areas.

With regard to the fact that grade crossings equipped with high-efficiency LED lights satisfying federal requirements require power on the order of 10 Watts per lamp, an energy harvesting device capable of generating 40 Watts or a series of devices each capable of
producing energy on the order of 10 Watts is needed to power 4 lights. Also it is required that the power generation device does not interrupt trains or maintenance operations.

This thesis presents three potential approaches to address the need of providing power to railroad grade crossings in remote areas by designing and developing three different types of power harvesting devices.

In the first approach, improvements to an existing mechanical energy harvesting device [2], capable of harnessing the vertical displacement of the rail and attached ties due to railcar traffic, are carried out to increase the power production capability of the device, and the results of lab and field testing are presented and discussed.

In the second approach, the possibility of generating electrical power with the passage of each railcar wheel is investigated, the mechanical device is designed, the 3D model is developed and the results of computer simulation are presented and discussed.

In the third approach, a hydraulic solution is developed to harvest power from vertical deflection of railroad track from passing railcar traffic. The prototype is built and the results of lab and field testing are presented.

This thesis is organized as follows. Chapter 2 reviews the current power harvesting techniques including piezoelectric and electromagnetic power harvesting methods. Chapter 3 describes improvements to the first prototype of a mechanical energy harvesting device as well as the design and testing results. Chapter 4 investigates the possibility of producing electricity by passage of each train wheel by developing a cam-follower mechanism connected to a generator. Chapter 5 describes a hydraulic system capable of harvesting power from vertical deflection of railroad track; also the testing results are presented and discussed. Chapter 6 concludes the thesis.
Chapter 2. Background

2.1 Existing technologies and techniques

Energy harvesting (EH) is a process that scavenges energy from ambient sources, such as solar power, wind energy, kinetic energy, etc. The harvested energy can be stored usually either with a capacitor or with a rechargeable battery to provide electrical energy for extensive applications including small electronic devices and wireless network sensors [3,4]. In this section, kinetic energy harvesting approaches primarily using piezoelectric and electromagnetic transducers are presented.

2.1.1 Piezoelectric energy harvesting

Piezoelectric materials have the ability to generate electricity as a response to mechanical strain. Several studies have been devoted to the field of piezoelectric power harvesting from human body motion for implanted devices and wearable electronics to regular or random displacements and vibrational energy [3].

In 2001, Shenck and Paradiso [5] generated an average electrical power of 8.4 mW deploying shoe-mounted piezoelectric subjected to mechanical strains by forces exerted on a shoe during walking.

In 2008, Feenstra et al. [6] developed an energy harvesting backpack in which the strap buckle was replaced with a mechanically amplified piezoelectric stack actuator. An average electric power of 0.4 mW was obtained as a response to the tension changes in the strap.
In 2003, Platt [7] employed a piezoelectric device within an artificial knee implant to power a microprocessor. This generated approximately 5 mW, utilizing three 1.2 cm³ piezoelectric actuators subjected to loading which simulated a typical human gait.

In 2003, Ottman et al. [8] developed an approach to harvest vibrational energy utilizing mechanically excited piezoelectric elements with a step-down converter. A maximum power of 31 mW was generated under laboratory conditions.

In 2008, Nelson et al. [9] investigated the possibility of scavenging electrical power from railcar traffic by deploying piezoelectric and inductive voice-coil techniques. Passing railcars cause longitudinal strain in a piezoelectric device mounted to the bottom of a rail. The changing longitudinal strain then is converted to electric current. An alternative technique utilized an inductive voice-coil device attached to the rail such that it was driven by the vertical displacement of the rail. The average power harvested was approximately 1 mW for both techniques.

### 2.1.2 Electromagnetic energy harvesting

Electromagnetic generators operate based on electromagnetic induction, known as Faraday's law, that if an electric conductor is moved relative to a magnetic field, electric current will be induced in the conductor. A number of researchers have investigated energy harvesting techniques leveraging this principle.

In 2005, Rome et al. [10] developed a suspended-load backpack, detaching the load from the body so that the differential movement between them is possible by a set of springs, converting the vertical movement of the backpack’s load during typical walking into rotary motion of the shaft of an electric generator. A maximum electrical power of 7.4 W was generated.
In 2009, Jung et al. [11] developed a new energy harvesting electromagnetic device using wake galloping for low-power applications, such as wireless sensors, whereas ordinary types of wind turbine would be too inefficient and costly. Applying wake galloping (aerodynamic instability) phenomena of structures under wind conditions, a bluff body exposed to a flow field is vibrated by flow-induced forces and a permanent magnet on the end of the vibration-cylinder experiences a reciprocating movement in a solenoid coil. Maximum electrical power of 1.13 W was generated under a wind speed of 5.6 m/s with a natural frequency of 4.8 Hz.

In 2009 Avadhany et al. [12] developed and patented a regenerative shock absorber. A hydraulic motor is driven by a cylinder and piston which act as a hydraulic pump. The piston undergoes a reciprocating motion and works during both compression and extension as a vehicle’s suspension system deflects. The hydraulic motor shaft is directly coupled to a permanent magnet generator. Their primary results show that for a 6-shock heavy truck, each shock absorber is capable of generating up to an average of 1 kW on a standard road [13].

In 2010 Nagode et al. [14] proposed a vibration-based electromechanical system to scavenge mechanical energy wasted in dampers of railcars to provide electrical power for railroad onboard applications such as smart devices that could be added to improve the efficiency of rail operation but which have been held back because of the lack of electrical power. They designed and fabricated two prototypes. The first prototype uses linear motion of several magnets as a rotor with the suspension inside a number of coils. At most 1 W RMS was generated with an input of ±0.75 inch at 1.5 Hz. The second prototype was developed in a manner such that the linear motion is converted into rotary motion, magnified, and rectified in order to turn a generator. Maximum electrical power of 54 W RMS was generated with an input of ± 0.75 inch at 1 Hz due to some modifications to the prototype.
2.2 Power requirements

Highway-railroad grade crossing has always been a major safety concern and a possible danger to motorists. Since oncoming trains cannot stop for vehicles, flashing red lights at grade crossings help the motorists to see far enough down the tracks from the crossing, giving them enough time to either stop or cross safely. Current warning lamps consist of eight 12-inch diameter high-efficiency and low-power light emitting diode (LED) lamps. Satisfying federal regulation, power on the order of 10 watts is required per lamp [15] at the mandated intensity according to the American Railway Engineering and Maintenance-of-Way Association (AREMA) standards [16]. Also at any railroad grade crossing, there should exist four lamps per traffic direction (two sets of two lamps, each pair alternating on and off). Thus, for a two-way traffic crossing, four of the eight lamps are illuminated simultaneously, and approximately 40 W of power is required at a grade crossing warning system.
Chapter 3. Improving an energy harvesting device

3.1 Design

This chapter presents the development of a mechanical energy harvesting device capable of harnessing the vertical displacement of the rail and attached ties due to railcar traffic. The first prototype [2], shown in Figure 3.1, was designed and built to harvest power on the order of 10 W. The device is mounted to and spans two rail ties. As it is shown in Figure 3.2, it consists of: a rack and pinion gear to convert the linear motion to rotary motion, a bearing clutch to rectify the up-and-down motion of the railroad track, a gearbox to amplify the rotary motion, and a permanent magnet DC generator. When the railcar traffic travels over the track, the rail-tie system displaces vertically. The rack gear is fixed in the subgrade, the device travels up and down relative to the rack gear, and the pinion gear moves relative to the rack gear. Thus a small amount of linear motion is converted to rotary motion. Then the bearing clutch mechanically rectifies the up-and-down displacement of the railroad track, the rotational motion is amplified via a gearbox and finally the permanent magnet DC generator produces electricity.

Figure 3.1. The first prototype 3D CAD model shown mounted across two ties of a section of railroad track [2].
While the lab test results of the first prototype indicated power production of 3.9 W for a train speed of 7.4 mph and railroad track vertical displacement of 0.75 inches, an average power of 0.22 W was harvested for a train speed of 11.5 mph and 0.5 inches of track deflection in field testing [17].

Since the power requirement for high-efficiency crossing lights is on the order of 10 W, improvements to the first prototype are of interest to increase the power production capability of the device. In the following sections, the major improvements made to the first prototype are described, and the results of lab and field testing are presented and discussed.
3.2 Physical constraints

Searching for allowable workspace in which the energy harvesting device could be mounted, a 30 inch long space along the track from one tie edge to the next tie edge, a 13-16 inch wide serviceable space from the edge of the tie plate to the end of the rail tie, and a 5-6 inch height, a total space across two railroad ties in between the edge of the ties and the rail, is being considered as the practical space, shown in Figure 3.3.

The device scavenges electrical energy from the vertical displacement of railroad track due to passing railcar traffic. It is mounted to and spans two rail ties such that it directly harnesses the track’s entire upward and downward displacement and converts the linear motion into rotary motion and then magnifies the rotary motion to a permanent magnet DC generator (PMDC) generator.

Figure 3.3. Allowable workspace for energy harvesting device.

3.3 Design and improvements to power harvesting device

In this section, design and improvements to the power harvesting prototype are described. The major improvements to the new prototype include: designing an efficient mechanism to
harvest power from the upward (return) railroad track displacement in addition to the downward
deflection, deploying a higher speed ratio planetary gearhead to increase the angular velocity of
the generator input shaft, utilizing an enhanced generator with greater power production capacity
for the same shaft speed, and improving the way the system is anchored into the subgrade to
maximize captured motion and energy harvested.

3.3.1 Translating linear motion into rotary motion

Converting linear motion into rotary motion was the first problem to be addressed. To
accomplish this, a rack and pinion gear system was used due to its good rigidity, durability and
capability of handling the large loads from railcar traffic, little backlash between components,
variety of gear pitches to choose from, and easy mounting.

A module 2 rack and pinion gear system was selected. A rack gear with a face width of
20 mm and a height of 25 mm for the cross-sectional area [18] and a corresponding spur gear
with 20 teeth [19] were chosen. Although the amount of torque transferred via the rack and gear
system is to be increased compared to first prototype due to improvements carried out to the first
prototype, this size of rack and gear still will handle the loads and resist shearing and bending but
with a smaller safety factor. The size of the rack and gear was not changed since the previously
selected components satisfied available space constraints, had good commercial availability, and
provide sufficient strength for the new load conditions. Since the diameter of shaft through which
the torque is transferred is the same as the first prototype, the pinion gear bore and hub diameters
still satisfy the pinion on shaft mounting requirements. The 12 mm hole was bored to 0.50 inches
and the hub diameter of 30 mm and width of 15 mm allowed for ample space in order to pin the
pinion gear to the shaft to keep the pinion gear from slipping on the shaft during torque
transmission.
3.3.2 Harvesting power from upward displacement in addition to downward

Railroad track travels up and down as railcar traffic deflects it downward and then it rebounds upward. The first prototype presented in [2] was designed such that it harvests power from downward displacement of the track. The bearing clutch, shown in Figure 3.2, freewheels when the track travels upwards; thus the PMDC generator is not driven in the opposite direction, which would otherwise cyclically change the direction of generator shaft rotation and produce reversing current, thus decreasing the power production efficiency.

Therefore one potential way to increase the average speed of the generator shaft for longer durations of time and consequently increasing the amount of generated power is to harvest power from upward displacement of the track in addition to downward displacement.

One possible solution suggested in [20] to harvest power from the upward displacement of the railroad track was to employ a second power harvesting device and simply reversing the direction in which the clutch freewheels. This way the second device will freewheel whenever the track is displaced downwards, and as the track is displaced upwards, the clutch will engage and rotate the generator. Mounting two devices next to each other, the entire upward and downward displacement of the track could be harnessed.

Taking into account the space availability constraints for installing the devices, double cost and time associated with manufacturing two devices as well as difficulties to store output power from them, one efficient mechanism to harvest power from upward displacement in addition to downward in a single device is shown in Figure 3.4 as well as a detailed top view in Figure3.5.
Figure 3.4. 3D CAD model of the device shown mounted in a section of railroad track.

Figure 3.5. Detailed top view of the two-stroke harvesting mechanism incorporated into the device.
The new mechanism works as follows: when the railcar traffic deflects the track downward, the pinion gear rotates counter-clockwise (from the right view) relative to the fixed rack gear; thus the entire right shaft assembly set, including the right shaft coupling and the drive gear, shown in Figure 3.4, rotates counter-clockwise. Hence the driven gear and the left shaft coupling rotate clockwise. Both bearing clutches are identical: they engage when they are driven CCW and freewheel when they are driven CW. Given that, the right clutch engages and the left one freewheels; thus the right gear rotates CCW and acts like a drive gear for the middle driven gear. Then the middle gear rotates CW and finally the generator shaft rotates CW after the gear head. On the other hand, for the upward displacement of the track, the rotation direction of the pinion gear is reversed, and consequently the right clutch freewheels and the left one engages. Thus the left gear rotates CCW and acts as a drive gear driving the middle gear CW. Accordingly this mechanism drives the PMDC generator always CW for both downward and upward displacements. This mechanism also enables the device to harvest power from the entire downward and upward motion of the track.

Bearing type clutches were employed in the second prototype like the first prototype because the bearing clutches are considered as the most suitable mechanism to mechanically rectify the vertical track deflection. Also they are fairly small but robust with virtually zero backlash, and are easily integrated into a variety of mounting configurations.

3.3.3 Changing the gearing

A rack and pinion gear was deployed to convert the linear vertical displacement of the track to rotational motion of the device. Since the magnitude of track deflection is in the range of 0.1 to 1.1 inches, the pinion gear will only rotate less than one full revolution. The generator will also rotate a fraction of a revolution unless the rotational motion is amplified. Accordingly, it was
necessary to magnify the rotational motion in order to rotate the generator with enough speed so that it can start generating useful levels of power. Moreover, taking into account possible rotational motion loss as resulting from backlash present between gears and other components such as flexible shaft couplings, amplification of rotational motion is essential.

Accordingly, a 1:50 planetary gearhead was used for the first prototype as a speed increaser to amplify the rotational motion.

Since the PMDC generator produces electrical power proportional to the speed at which its shaft rotates, the faster and longer the generator rotates, the more electrical power it generates. Therefore, further amplification of the rotational motion of the device would lead to higher power production capability of the device. Accordingly a planetary gearbox with a ratio of 1:100 was used to replace the planetary gearbox having a ratio of 1:50 in the first prototype as a speed increaser [21]. Also the new gearbox is more robust, lifetime lubricated and with low backlash with a slight change in size compared to the 1:50 version.

3.3.4 Utilizing an enhanced generator

A permanent magnet direct current generator was chosen to produce power by integrating it with other components in both prototypes. The PMDC generators are also reliable and durable.

Besides further amplification of the angular speed that the generator experiences, as mentioned in the previous section, an improved generator with larger output levels of power for the same shaft speed was incorporated into the second prototype to increase power output levels (Windstream Power LLC Permanent Magnet DC Generator model 443541 instead of model 443540). The new generator is as efficient as the previous one with the same minimum angular
speed point at which the generator starts producing power, but its performance curve has a steeper slope in terms of power versus shaft speed [22].

### 3.3.5 Improving the anchoring method

To harness the vertical deflection of the track using a rack and gear system, the rack gear needed to be stationary and completely rigid to allow the pinion gear to move relative to it. Thus a good mounting scheme was essential to eliminate the possibility of vertical movement of the rack gear while the track is being displaced downward and/or upward and the device is operating; otherwise the limited vertical deflection of the track is constrained even more by lost motion at the rack/gear input. In other words, since the device was designed based on the fact that it harnesses the vertical displacement of the track, maximum effort should be dedicated to capture as much vertical displacement as possible; otherwise, even if all other parts work properly as they were intended, the device won’t be able to generate power.

Since the vertical deflection of the railroad track is likely to be on the order of 0.5 inches for loaded train cars, any efforts to minimize lost motion or maximize captured motion are worthwhile and result in harvesting higher levels of electrical power. A modified auger bit, shown in Figure 3.6.a, replaced the spike, shown in Figure 3.6.b, which had been used in anchoring the first prototype. The anchoring method to stabilize the rack gear for the second prototype is less likely to move in the subgrade (especially in wet soil). It is also easier to drive the auger in and out compared to the previous design that had to be hammered down, causing damage to the thread and bending/buckling to the spike.
The design of the rack gear mounting scheme, shown in Figure 3.7, consists of a 1.5 inch thick base plate with a slot on the top surface to pin the rack. The bottom surface of the rack gear base has a threaded hole to facilitate connection to the auger bit by a threaded coupling.
About four inches of ballast has to be removed from the area between the two ties and the rail to mount the rack gear. Then the auger bit is driven down into the soil. To do so, a very simple manual driver was designed and fabricated, as shown in Figure 3.8. This makes driving the spike into the ground possible while there is no electrical or generator power available to utilize normal electric or hydraulic auger drivers.

![Auger bit manual driver](image)

**Figure 3.8.** Auger bit manual driver.

### 3.4 Design of other components

In addition to the major improvements mentioned in the preceding sections, there have been other minor improvements made to the second prototype including: making the base plate and slotted guide bracket simpler and lighter, making them easier to be fabricated and installed on the track, making some components more robust such as the L-shaped rack guide mounts, and deploying different kinds of bearings to address the space constraints.

Every component of the power harvesting device, excluding the rack gear mounting setup and fasteners, is shown in an exploded view in Figure 9.
Although some of the major components making up the power harvesting device such as the generator, gearbox, bearing clutches, bearings and gears were purchased from different vendors, there were tens of other parts designed and manufactured at the UNL engineering machine shop, such as shafts, brackets, base plate, bearing housings and rack gear mount. Furthermore, some of the purchased components needed to be modified and machined. Gears were bored to their corresponding shaft diameters, and two holes were drilled out and tapped for each of them to sink the set screws into the shafts.

Based on environmental conditions for outdoor railroad applications, stainless steel was a preferable material chosen for the majority of the components, either purchased ones such as gears and fasteners or manufactured ones such as shafts, base plate and rack gear mount; however, bearing housings and brackets were made out of aluminum.

To compensate for any slight misalignments and possible shocks resulting from intermittent torque that would be transmitted through the pinion gear because of cyclical deflection of ties over which the device is attached, two flexible shaft couplings were deployed.
between the pinion gear at the right shaft set, the driven gear at the left shaft set, and the ball bearing clutches [23].

To support two different shaft sets transferring torque from downward and upward displacements of the railroad track, two different set of bearings were needed: one set with 2 ball bearings, 2780T59 [24], and one with 3 needle roller bearings, 5905K133 [25]. Needle rollers were chosen for the second bearing set due to their small outside diameters allowing support of 3 gear shafts in a small available space.

Also, all bearings were double-sealed and sized for a 0.50 inch shaft diameter. Utilizing some internal and external retaining rings, all of the bearings were fixed on their housings and shafts respectively.

To keep the rack gear engaged with the pinion gear and constrain its motion only to vertical motion while the device is operating, an aluminum bracket with two ball transfers, providing a comparatively low-friction interface, was designed and integrated to the device.

In the whole process of designing, manufacturing and selecting components of the second prototype, the main emphasis was placed on robustness, durability, and reliability, rather than cost since the device was designed to operate in a dirty, harsh environment.

Table A.1 at Appendix A shows cost analysis for the second prototype. Appendix B.1 details all components that were designed, modified, and manufactured as well as purchased components. Appendix C.1 to C.9 detail technical specification of purchased components.
3.5 Second prototype operation

When railcar traffic travels over a section of railroad track where the energy harvesting device is located, the rail and attached ties deflect vertically from 0.1 to 1.0 inch, depending on the railcar weight (whether the train is loaded or unloaded). Regarding the fact that the rack gear is fixed, the device, attached to the ties, has to travel up and down relative to the rack gear. As the device travels up and down, the pinion gear moves relative to the rack gear; thus small amounts of linear motion are converted to rotary motion. For downward displacement of the railroad track, the pinion gear rotates counter-clockwise (from the right view); thus the entire right shaft assembly set, including the right flexible shaft coupling, which reduces any impact loads to protect critical components with minimum energy dissipation, and the drive gear, shown in Figure 5, rotates counter-clockwise. Hence the driven gear and the left shaft coupling rotate clockwise. Both bearing clutches are identical; they engage when they are driven CCW and freewheel when they are driven CW. Given that, the right clutch engages and the left one freewheels; thus the right gear rotates CCW and acts like a drive gear for the middle driven gear. Then the middle gear rotates CW and the CW rotary motion of the middle gear is magnified via a 1:100 gear head. Finally the generator shaft rotates CW and produces positive DC voltage according to the CW direction. For the upward displacement of the track, the rotational direction of the pinion gear is reversed, and consequently the right clutch freewheels and the left one engages. Thus the left gear rotates CCW and acts like a drive gear driving the middle gear CW. Accordingly the PMDC generator rotates CW for both downward and upward displacements of the railroad track. The device harvests power from the entire downward and upward displacements of the track.
The built prototype of the device is shown in Figure 3.10, and the built prototype of the rack gear mounting scheme is shown in Figure 3.11.

Figure 3.10. Second prototype of the power harvesting device.

Figure 3.11. Prototype of the rack gear mounting scheme.
3.6 Comparison of simulation results between the first and the second prototype

Using a mathematical simulation described in detail in [26], a comparison between the first and the second prototype is made. The generator shaft speed and the device output power level for loaded and unloaded railcars are compared. The loaded train consists of 104 loaded railcars and 3 locomotives, and each railcar is estimated to weight approximately 280,000 pounds. The unloaded train consists of 64 unloaded railcars and one locomotive, and each unloaded railcar is estimated to weight approximately 40,000 pounds. Estimated train speeds of 11.5 mph for the loaded train and 13.5 mph for the unloaded train, obtained from the results of first prototype field test [17], were used for simulation, and 60 mph was considered the probable maximum train speed for both loaded and unloaded cases.

The plot of the generator shaft speed from a loaded train traveling at 11.5 mph for the first prototype and the second prototype are shown in Figure 3.12. The average generator shaft speeds for the first and the second prototype are 141 rpm and 566 rpm respectively. The increase is a result of deploying a gearbox with twice the gear ratio in the second prototype compared to the first one as well as harnessing the upward displacement of the track in addition to the downward; therefore, the combined result is that the generator rotates 4 times as fast in the second prototype as it rotates in the first one, on average.
Figure 3.12. Generator shaft speed of (a) the first prototype; (b) the second prototype. Loaded train with speed of 11.5 mph.

The resulting waveforms of output power produced by the generators in the first and the second prototype for the loaded train are shown in Figure 3.13. The average output power of the
first and the second prototype are 0.36 W and 41.82 W respectively. The difference between the 
two prototype output power levels is largely attributable to the difference in generator power 
curves, shown in Figure 3.14.

Figure 3.13. Generator output power of (a) the first prototype; (b) the second prototype. Loaded train with 
speed of 11.5 mph.
Figure 3.14. Power curves [27] for (a) 443540 generator of the first prototype; (b) 443541 generator of the second prototype.

The plot of the generator shaft speed from an unloaded train with a speed of 13.5 mph for the first prototype and the second prototype are shown in Figure 3.15. The average generator shaft speeds for the first and the second prototype are 24 rpm and 95 rpm respectively.
Figure 3.15. Generator shaft speed of (a) the first prototype; (b) the second prototype. Unloaded train with speed of 13.5 mph.

The average output power of both prototypes under unloaded train conditions is 0 W since the output power simulation was made according to the generator curves, as shown in Figure 3.14, producing no power for any shaft speeds less than 360 rpm; however, the field test results for the first prototype indicate an overall average harvested power which, although small, is nonzero under unloaded conditions [17]. The discrepancy between the simulation result and the actual result from the field testing for unloaded conditions might be because the generators produce power even at shaft speeds less than the mentioned threshold of 360 rpm, but the output power is very small and the manufacturer chooses to provide a conservative estimate rather than characterize the nonlinear portion of the curve in detail.

Figures 3.16 and 3.17 show generator shaft speeds and output power levels of the first and the second prototype for a loaded train with the possible maximum speed of 60 mph. The
average generator shaft speeds and the average output power for the first and the second prototype are 738 rpm, 2952 rpm, 10.05 W and 306.28 W respectively.

Figure 3.16. Generator shaft speed of (a) the first prototype; (b) the second prototype. Loaded train with speed of 60 mph.
Figure 3.17. Generator output power of both prototypes for unloaded train. Loaded train with speed of 60 mph.
3.7 Lab testing of second prototype of mechanical energy harvesting device

The second prototype with improvements discussed in section 3.3 was designed and built, as shown in Figure 3.10. Then lab testing was conducted to evaluate the strength, rigidity, and general reliability of various components, and to verify the overall functionality of the prototype. A DC gear-motor with speed control was coupled to the end of the right shaft, shown in Figure 3.17, so that it enabled variation of angular velocity and reversal of rotational direction to validate the functionality of the device in both directions, representing the upward and downward displacement of the track. The first run helped to identify design improvements, such as sinking the set screws into the gear shafts to prevent gears from slipping on their shafts. The device functioned as expected after the mentioned modifications were implemented.

![Lab testing set up.](image)

Figure 3.18. Lab testing set up.
A National Instruments USB 6229 DAQ board and LabVIEW software were deployed to acquire the voltage signal. A hand-held digital tachometer was utilized to measure the angular velocity of the generator shaft.

Two 50-Ω resistors, a 21-Ω and a 10-Ω resistor, all in series (shown in Figure 3.18) such that an overall load of 131 Ω was applied, were connected to the generator to simulate a bank of lamps. The output voltage across the 10-Ω resistor was then acquired to scale the output voltage to within the specified range of the DAQ board (±10 V). The total output voltage and instantaneous power were calculated using Ohm’s law for resistors in series.

![Data acquisition setup](image)

Figure 3.19. Data acquisition setup.

Testing was carried out for two different generator speed, approximately 120 rpm and 560 rpm, representing the speed at which the generator rotates for unloaded and loaded train conditions respectively, according to the mathematical simulation described in detail in [26].
The output voltage and instantaneous power for unloaded and loaded train conditions are shown in Figures 3.20 and 3.21. The output voltages, illustrated as a constant value, were calculated to be 4.3 V and 24.5 for unloaded and loaded conditions respectively. The average power, illustrated as a constant value, was calculated to be 0.19 W and 4.24 W for unloaded and loaded conditions respectively.

Figure 3.20. Output voltage and power for unloaded train conditions.
3.8 Discussion of test results

Comparing the simulation results with lab testing results, the mechanical energy harvesting device generated approximately 10 times less power in lab testing than was expected.
according to mathematical simulation. It was assumed in the mathematical simulation that the output voltage of the generator would be 12 V, while the average voltage was calculated to be about 24.5 V in lab testing. According to the power curve of the generator, shown in Figure 3.22 [27], the output power depends on the output voltage and generator shaft speed. This might explain the difference between simulation output power and lab testing output power. Also, the output voltage of a permanent magnet DC generator is governed by its shaft rpm and the load applied to it. On the other hand, small voltage and high shaft rpm are desirable in order to harvest more power which contrasts with the generator power curve characteristics; solution of this design discrepancy may be addressed as future work.

Figure 3.22. Generator power curve [22].
Chapter 4. Generating electricity by passage of each train wheel

4.1 Introduction

According to the field test results of the mechanical power harvesting devices, the output power is very small and limited under unloaded train conditions with low speeds, about 12 mph. The device also generated a limited amount of power under freezing weather conditions in which the ground was covered with snow; the subgrade was too solid to let the railroad track deflect vertically to the extent that it did in warmer weather conditions. Generally speaking, if the vertical deflection of the railroad track is small for any reason, the second prototype discussed in Chapter 3 and, generally, the mechanical energy harvesting devices are constrained to very small amounts of output power levels. This motivated us to search for alternative approaches capable of generating electricity independent on weather and/or other conditions which affect the deflection of the railroad track. One promising approach can be harnessing electricity by passage of each train wheel.

Accordingly, to design a specific device capable of generating power, it might be desirable to impart an automatic motion to a linkage with the passage of each train wheel. However there exist several electrical solutions, wherever a mechanical solution is required for any reason including the lack of electrical infrastructure in our case, one way can be deploying a cam-follower mechanism.

One example of similar devices deployed in railroad systems is the mechanical wayside lubrication system [28], lubricating the contact surface of the rail and wheel or wheel bearings automatically by passage of each railcar wheel. Figure 4.1 shows the ramp lever and ramp
assembly of the mechanical wayside lubrication system coupled to a piston-type grease pump through a universal joint. Considering impact effects, abrupt changes of velocity and acceleration and excessive jerk at the time of impact, shown in Figure 4.2, and/or loss of contact between the ramp lever and train wheel, the resulting shock and wear on the lubricator device can be a concern. Furthermore, pressure angles less than 60 degrees at most points during contact between the ramp lever and train wheel, especially at the time of impact, shown in Figure 4.3, can lead to considerable load on bearings and other components. The same concerns would be equitable for any similar railroad applications where providing automatic action with the passage of each wheel is desirable. Therefore, developing a well designed cam-follower mechanism is essential to address the mentioned problems.

Figure 4.1. Ramp lever and ramp assembly of mechanical wayside lubrication system.
Figure 4.2. Angular velocity and acceleration of the ramp lever when; a) train wheels contact it from front, b) train wheels contact it from back.
Figure 4.3. Pressure angle of the ramp lever at the time of impact when; a) train wheels contact it from the front, b) train wheels contact it from the back.
4.2 Cam-follower mechanism

Cam-follower mechanisms are utilized in a wide variety of automatic machines and are a convenient and important mechanism to transform either a rotary or translating motion into almost any desired follower motion. According to the shapes and motions, rotary or translating, of the cams, and the shape of the contact surfaces and motions, translation or oscillating, of the followers, cam-follower mechanisms can be classified into several types. However, the most common types are disc cam with translating follower and disc cam with oscillating follower.

In order to design a cam-follower mechanism, the first step is usually to synthesize the motion function of the follower with regard to its prescribed motion constraints. The second step is generating the cam surface profile used to impart the functional motion to the follower.

Whereas most of the published books and papers describing design approaches have been devoted to disc cams (mostly with translating followers) and profile generation and optimization of cams [29,30], design of translating cams with oscillating followers and approaches for follower profile generation in this type of cam-follower system still in some cases remains a challenge.

Due to the definition of a cam as a mechanism component which serves as a driving link [31], the railcar wheel flange is considered the cam, and since the gross motion of the train wheel is translational (ignoring for the moment its local rotation), it is classified as a translating cam. Since the shape of the train wheel is known and in all practicality is not changeable, the problem is defined as determining the surface profile of the oscillating follower driven by prescribed motion of a translating cam.
4.2.1 Cam-follower motion

The first decision to be made is choosing the type of motion for the follower. The selection of follower motion depends on several parameters including [32]: speed of the cam, permissible noise and vibration level, shock effect, life expectancy, geometry considerations, cost, and feasibility of surface profile manufacturing.

With regard to the railroad-specific application, once the train wheel flange, as a translating cam, approaches the device consisting of a cam-follower mechanism, and contacts the follower, the follower and device are vulnerable to significant shock, wear, and substantial lateral components of the impact force on the follower bearings, due to mechanics of the impact. Therefore, the main concern regarding selecting the follower motion is to minimize shock and wear, while considering the geometry constraints as well.

Since acceleration of the follower affects inertia forces influencing shock and wear, and discontinuous or abrupt changes of acceleration at the time of impact and/or loss of contact between the pair of surfaces would cause high levels of jerk resulting in damage as well, the best follower motion is one which avoids discontinuities in acceleration of the components and minimizes jerk. Cycloidal motion satisfies this requirement in terms of the parameters discussed, and can be addressed with geometry and specific application constraints in mind.
4.2.2 Cycloidal motion equation for translating cam with oscillating follower

The equation for cycloidal displacement for a disc cam and translating follower can be written as [32]:

\[
s = h \left( \frac{\theta}{\alpha} \right) - \left( \frac{h}{2\pi} \right) \sin \left( 2\pi \frac{\theta}{\alpha} \right)
\]

(4.1)

where \( h \) is the maximum lift of the follower, \( \alpha \) is the angle of the cam when the follower is at its maximum lift, and \( \theta \) is the cam angle associated with a displacement \( s \).

Figure 4.4. Displacement, velocity, acceleration, and jerk for cycloidal motion.
Displacement, velocity, acceleration and jerk diagrams for cycloidal motion [32] are shown in Figure 4.4 for a dwell-rise-dwell follower motion.

Considering the acceleration diagram, a smooth start from zero, no discontinuities to cause infinite jerk, and a minimum amount of shock, wear and noise, are expected as the advantages of cycloidal motion.

Then, we restate the angular displacement equation of cycloidal motion for a translating cam with oscillating follower in Equation 4.2, taking into account initial conditions of $\beta = \gamma$ at $x = 0$ and $\beta = 0$ at $x = L$.

$$\beta = -\gamma\frac{x}{L} + \left(\frac{\gamma}{2\pi}\right)\sin\left(2\pi\frac{x}{L}\right) + \gamma$$

(4.2)

where $\beta$ is the angular displacement of the follower, $\gamma$ is the maximum angular displacement of the follower, $x$ is the cam center displacement at which the angular displacement $\beta$ occurs, and $L$ is the length through which the cam has passed when the follower achieves its maximum angular displacement. Figure 4.5 illustrates the terms used in Equation 4.2.
4.3 Follower surface profile synthesis

There are several methods and approaches to generate cam profiles. The most common methods are graphical and analytical methods. However, the analytical approach is preferred in cases where highly accurate cams are required. Here let us begin with a graphical approach in order to first investigate the existence of a solution.

Based on availability of space for the follower, $\gamma = 30^\circ$ clockwise and $L = 6$ inches are chosen as constant parameters of Equation 4.2. $L$ is divided into 6 equal intervals, and corresponding angular displacements, $\beta$, are calculated using Equation 4.2. Table 4.1 shows the intervals and corresponding angular displacements. Figure 4.6 shows the graphical approach to generate the follower surface profile. The initial position of the follower is at point 1. While the common type of problem is defined as synthesis of the cam profile, the main difference here is...
synthesis of the follower surface profile. Therefore, unlike the approach for cam profile generation, it is necessary to invert the approach so that the follower is held stationary while the cam first moves linearly for $x_i$ and then rotates about the follower, point O, for $\Delta \beta_i$ in the opposite direction.

Table 4.1. Cam displacement intervals (in inches) and corresponding angular displacements (in degrees).

<table>
<thead>
<tr>
<th>i</th>
<th>$x_i$</th>
<th>$\beta_i$</th>
<th>$\Delta \beta_i$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>30</td>
<td>0</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>29.14</td>
<td>0.86</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>24.14</td>
<td>5.86</td>
</tr>
<tr>
<td>3</td>
<td>3</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>4</td>
<td>4</td>
<td>5.86</td>
<td>24.14</td>
</tr>
<tr>
<td>5</td>
<td>5</td>
<td>.86</td>
<td>29.14</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>0</td>
<td>30</td>
</tr>
</tbody>
</table>

The first cam center is at point $C_0$. As the follower is pivoted at point O, $C_0$ first moves horizontally by $x_1$ to point P, and then an arc of radius OP, with its center at O, is drawn through point P, and point $C_1$ is located. A circle of radius 18 inches, equal to the radius of the train wheel, is drawn. The angle $\Delta \beta_i = 0.86^\circ$ is the angle through
Figure 4.6. Graphical synthesis of follower profile.
which the follower will rotate while the cam moves from point C₀ to point P. Proceeding from point C₀ in the development, the centers of the next positions of the cam are obtained in a similar manner. Figure 4.7 shows the corresponding cam positions. The
next step in the procedure is to draw a smooth curve tangent to these circles, but careful inspection of these circles in Figure 4.7b shows that drawing a curve tangent to these circles from point 1 is impossible. One way to make it possible is by increasing the size of the follower so that it reaches the size of the cam, which is not achievable considering available space constraints.

4.4 Cam-follower solution with curved groove

As mentioned in section 4.3, it is not workable to increase the follower size or decrease the size of the train wheel functioning as a translating cam. An alternative solution can be achieved by utilizing an intermediate component, seen in Figure 4.8, driven by the train wheel but having much smaller size compared to the train wheel. We use a component consisting of three coaxial rollers: one with a diameter of 1.75 inches in the middle driven by the train wheel flange, and two with diameters of 0.375 inches on either side. One of these latter two rollers rides in a curved groove hereafter denoted the stationary groove. The other one functions as a translating cam imparting oscillating motion to the follower as it rides in another groove on the follower, hereafter denoted the follower profile groove. (This configuration is functionally similar to a pin/slot arrangement.) Therefore, the main challenges are to determine the stationary groove curvature and the profile of follower to attain cycloidal motion.

4.4.1 Determining stationary groove curvature

To determine the curvature of the stationary groove, there are some concerns which should be considered. First of all, the roller should not interfere with the train
wheel while moving through the groove. In other words, the roller should lie fully below
the line which is tangent to the bottom of the wheel when it reaches the end of the
groove. This means that the groove can be neither horizontal nor positively sloped
(considering the train wheel moving from left to right). Also, any composite curves such
as splines or multilines are not desirable, since likelihood of sticking or jamming can
increase with changes in curvature. Consequently, the first solution that comes to mind
would be a concave arc with the same constant radius as the train wheel flange radius and
an angle equal to the angle between the impact point and the bottom point of the train
wheel flange. (This latter point is the point where the train wheel flange is assumed to
lose contact with the roller at the end of the displacement.) Figure 4.9 shows the
discussed stationary groove curvature and follower profile groove drawn based on these
considerations.

The stationary groove curvature and consequent follower profile groove, shown
in Figure 4.9, still need to be modified in terms of size and pressure angle. Space
availability constraints in some applications may force us to minimize the size of the
follower. As can be seen in Figure 4.9, the shape and radius of the stationary groove
curvature directly affect the size and shape of the follower profile. The graphical
approach in the next section will also clarify this. Regarding our application-specific
space availability constraints, the radius of stationary groove curvature is chosen to be an
arc with radius equal to half of the train wheel flange radius and an appropriate angle
discussed in the next section. The arc radius is chosen somewhat arbitrarily in order to fit
the overall mechanism in the available space.
Figure 4.8. Cam-follower solution with intermediate component (roller) and curved groove.

Figure 4.9. Stationary groove curvature and consequent follower profile groove.
Pressure angle in our cam-follower, shown in Figure 4.10, is defined as the angle between the instantaneous velocity vector of the follower measured at the roller-follower contact point (along line CN), and the normal to the follower profile at the same point (along line CA). Additionally, the contact force between roller and follower is collinear with the normal to the follower profile, ignoring friction. The pressure angle varies during follower angular displacement. Since a pressure angle greater than 30° is not desirable, especially at the initial impact, because of appreciable lateral force exerted on the follower bearing at the pivot point, pressure angles for the follower profile should be measured and examined accordingly. Graphically measuring pressure angles for the follower profile, drawn based on the concave arc of the stationary groove, shows that angles greater than 30° occur for most points along the follower profile while it is oscillating. As illustrated in the next section, since the follower profile is obtained based on the shape of the stationary groove, changing the stationary groove curvature results in different follower profiles and subsequently different pressure angles. Moreover, it is
observed that when the curvature of the stationary groove is changed from a concave arc to a convex arc with the same radius, and the desired boundary conditions are imposed for the initial point of contact, overall pressure angles improve, particularly at initial impact and early in the follower motion; this is discussed and demonstrated in the next section.

To summarize the parameters discussed in this section, a convex arc with radius of 9 inches is chosen to function as the stationary groove curvature; the length of this arc is discussed in the next section.

4.4.2 Graphical synthesis of the follower profile

Regarding new space constraints caused by the new approach, the follower pivot point location and size and the first position of the roller are determined as shown in Figure 4.11. In Equation 4.2, maximum angular displacement of the follower is treated as a constant parameter having a value of $\gamma = 16^\circ$ clockwise.

First, to find the stationary groove curvature, the angle of arc discussed in the previous section should be determined. We start by drawing a convex arc with radius of 9 inches from the point tangent to the bottom of the roller, point $S_1$, to point $S_2$ at which the roller is fully below the path of the train wheel flange as described at the beginning of section 4.4 and illustrated in Figure 4.11. The length of the arc $S_1S_2$, 3.7 inches, is the constant L in Equation 4.2. In the case of larger train wheels or locomotive wheels, the wheel flange radius can increase by 0.125 inches; thus we extend the arc to point $S_3$ to accommodate the extreme case.
To synthesize the follower profile more accurately, L is divided into 10 equal intervals, and corresponding angular displacements, $\beta$, are calculated using Equation 4.2. Table 4.2 shows the intervals and corresponding angular displacements, where $x_i$ is defined as displacement along the arc form point $S_1$. Since the roller driven by the train wheel flange drives the follower, it is considered as a translating cam. Thus the problem is defined as follower surface profile synthesis with a translating cam riding through a stationary convex-arc groove.

![Diagram of train wheel flange and rail surface](image)

Figure 4.11. Stationary groove curvature synthesis.
Figure 4.12. Follower profile synthesis.

Table 4.2. Cam (roller) circular displacement intervals (in inches) and corresponding angular displacement (in degrees).

<table>
<thead>
<tr>
<th>i</th>
<th>x_i</th>
<th>β_i</th>
<th>Δβ_i</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>16</td>
<td>0</td>
</tr>
<tr>
<td>1</td>
<td>0.37</td>
<td>15.90</td>
<td>0.103</td>
</tr>
<tr>
<td>2</td>
<td>0.74</td>
<td>15.22</td>
<td>0.78</td>
</tr>
<tr>
<td>3</td>
<td>1.11</td>
<td>13.62</td>
<td>2.38</td>
</tr>
<tr>
<td>4</td>
<td>1.48</td>
<td>11.10</td>
<td>4.90</td>
</tr>
<tr>
<td>5</td>
<td>1.85</td>
<td>8.00</td>
<td>8.00</td>
</tr>
<tr>
<td>6</td>
<td>2.22</td>
<td>4.90</td>
<td>11.10</td>
</tr>
<tr>
<td>7</td>
<td>2.59</td>
<td>2.38</td>
<td>13.62</td>
</tr>
<tr>
<td>8</td>
<td>2.96</td>
<td>0.78</td>
<td>15.22</td>
</tr>
<tr>
<td>9</td>
<td>3.33</td>
<td>0.10</td>
<td>15.90</td>
</tr>
<tr>
<td>10</td>
<td>3.70</td>
<td>0</td>
<td>16</td>
</tr>
<tr>
<td>11</td>
<td>3.77</td>
<td>0</td>
<td>16</td>
</tr>
<tr>
<td>12</td>
<td>3.91</td>
<td>0</td>
<td>16</td>
</tr>
</tbody>
</table>
The approach followed in laying out the follower surface profile is similar to that used in section 4.3. However, the only difference is that the roller (functionally a cam), has to travel through a convex arc instead of a horizontal line. As shown in Figure 4.12, the initial position of the roller center is at point $C_0$ and the first position of the follower is at point $1$ coincident to point $S_1$. As it is pivoted about point $O$, $C_0$ moves along the arc by $x_1$ to point $P$, and then an arc of radius $OP$ through an angle of $\Delta \beta_1$, with its center at $O$, is drawn through point $P$, and point $C_1$ is located. A circle of diameter 0.375 inches, equal to the diameter of the concentric circle of the roller, is drawn. Proceeding from point $C_0$ in the development, the centers of the next positions of the roller are obtained in a similar manner. Finally a smooth curve tangent to all of these circles drawn from point $1$ is the follower profile shown in Figure 4.13.
4.5 Simulation and analyses

Using MSC ADAMS software, we conducted dynamic simulations to quantify the follower motion and compare the expected motion parameters to achieved motion parameters obtained for follower motion based on our approach. We set the train speed to 11.5 mph and ignored the rotation of the train wheel since it drives a roller. We also assume that the train wheel flange stays in contact with the roller during the follower motion. Figure 4.14 shows the angular displacement and velocity diagrams of the follower. At first glance, one may notice that the angular displacement and speed diagrams are not symmetrical, in contrast to Figure 4.4. The reason is that the cam velocity for a standard disc cam or translational cam in Equations 4.1 and 4.2 is assumed to be constant, whereas the roller velocity here is not constant and actually decreases from the beginning to the end of the stationary groove, as we can see in Figure 4.15. Because the follower motion starts and ends with zero velocity and zero acceleration, the requirements of minimum wear and shock are satisfied.
Figure 4.14. Angular displacement, velocity and acceleration of follower.

Figure 4.15. Velocity of roller.
4.6 Deploying the cam-follower mechanism in designing an electricity generating device.

The cam-follower mechanism developed in preceding sections, was intended to be employed in an electricity generation plan; however, it is extensible to other vehicular system applications. The oscillating motion of the follower, discussed in the previous section, is a desirable motion in terms of its velocity, acceleration and jerk. Now the challenges are employing this mechanism for an electricity generation plan and modifying it to meet the railroad-specific requirements.

Regarding the designed cam-follower, the follower moves by the passage of each train wheel flange; thus another mechanism is needed to push the follower back to its initial position. Also generating electricity during this return motion is of interest.

On the other hand, since trains are traveling in both directions on railroad track, the electricity generating device has to be bi-directional. In other words, it should be capable of generating electricity for both directions.

Accordingly, a device which includes a mechanism to return the follower back to its initial position, is capable of generating electricity during return motion, and is capable of generating electricity in both directions should be designed.

Since one cycle of the follower oscillation takes place between a passage of two train wheels, the length of the whole mechanism including the follower, the return mechanism and bi-
directional solution, should be somewhat constrained to the length of the center distance between two successive railcar axles to ensure one full oscillation of the follower takes place before the next wheel reaches it, so that the one oscillating motion is carried out without interfering with the next train wheel flange. Additionally, since the follower is driven by the wheel flange, the device has to be installed inside the track between two rails. Thus the height of the device should not exceed the height of the corresponding rail (except for the follower) to leave enough clearance between the top of the device and the bottom of the railcar. Consequently, designing a device which is small enough to not interfere with any other parts of the railcar is presented.

Taking into account all of the requirements discussed above, a device, as shown in Figure 4.16, was designed.
Figure 4.16. a) 3D model of the ramp lever device; b) detailed view of the ramp lever device.

Figure 4.17. 3D model of the ramp lever device.
The device was designed to work as follows: when the railcar traffic travels over the device, assuming that the direction of the train is positive with respect to the coordinate system shown in Figure 4.17, the largest roller (roller number 5) is driven by the wheel flange, making both side rollers (rollers number 1 and 4) ride in the curved grooves. Then the middle roller (roller number 3) drives the first follower (mounted on the short shaft) counter-clockwise. The rotary motion and the torque are transmitted via a pair of spur gears to the middle shaft with clockwise rotational direction due to external gear engagement. Then the rotary motion and the torque are transmitted to the main shaft by means of a pair of chain gears and a chain with no change in the rotational direction, turning the main shaft clockwise. The main shaft can drive a generator to produce electricity. When the train is traveling over the device with negative speed, the largest roller (roller number 5) is driven by the train wheel flange and two side rollers (rollers number 1 and 4) ride in curved grooves. Then roller number 4 drives the second follower (mounted on the main shaft) clockwise, turning the main shaft clockwise. Thus the rotational direction of the main shaft remains clockwise no matter what the direction of the train is.

In addition to the mechanism described above, the mechanism shown in Figure 4.18 was designed to return the followers back to their initial positions as well as to generate electricity in the return motion.
The mechanism functions quite similar to a door closer. It consists of 2 arms, a rack-gear system and a spring. Whenever one of the followers is driven downward, the latch attached to the end of the main shaft pushes the longer arm down and the smaller arm rotates counter-clockwise, seen from the back side. The smaller arm is attached to a rack-gear system inside the box, as shown in Figure 4.18(b). The pinion rotates by the rotation of the small shaft, and then the rotational motion is converted to a linear motion by a rack gear, compressing a spring attached to the end of the rack gear. Consequently the energy stored in the spring makes the main shaft rotate counter-clockwise, returning both followers to their initial positions.

As was mentioned at the beginning of this chapter, the output of the whole mechanism, rotation of the main shaft, can either directly drive a DC generator or be coupled to the second prototype of mechanical energy harvesting device presented in Chapter 3 since the changes in the rotational direction of the output shaft of the ramp lever mechanism resemble the upward and downward deflection of the railroad track. Driving the improved mechanical device by the developed mechanism, no longer is the rack gear deployed in the mechanical device needed to drive the device; instead a proper gearing system is needed to transmit the output motion and
torque of the main shaft of the developed mechanism to the pinion gear in the second mechanical energy harvesting device. Since the pinion shaft axis of the mechanical device is perpendicular to the main shaft axis, a bevel gear set with module of 2 and gear ratio of 2 was deployed [33]. To support the bevel gear on the lever mechanism two roller bearings [34] were used due to support axial loads exerted on the shaft by the bevel gears. They were mounted inside a housing attached to the front plate of the lever mechanism. Also another roller bearing was used to support the small bevel gear. Another spur gear with module of 2 and 50 teeth [35] was used to transfer the rotary motion and the torque to the pinion gear so that the gear ratio between two spur gears is 2.5. Counting the gear ratio of 2 for the bevel gear set, the total gear ratio of the lever mechanism to the device is 5 (2.5×2).

A 3D model of the mechanical energy harvesting device driven by the lever mechanism and mounted in a section of a railroad track is shown in Figure 4.19.
4.7 Speed and output power calculation

The generator shaft speed and output power are calculated in this section to quantify the output power as well as to compare the results with other approaches presented earlier.

Assuming a loaded train is traveling at 11.5 mph over the device, the follower angular displacement would be 16 degrees (0.28 radians). According to Figure 4.14 the follower moves from the initial position to the end position in 0.04 seconds. The angular velocity of the follower and the main shaft is calculated to be 66.7 rpm using Equation 4.3.

\[ \omega = \frac{\Delta \theta}{\Delta t} \]  

(4.3)
where $\omega$ is angular velocity of the main shaft, $\Delta \theta$ is the angular displacement of the main shaft and $\Delta t$ is the period of time in which the follower moves from the initial position to the end position. This is the speed at which the two followers, both spur gears and chain gears would be rotating.

The bevel gear set and spur gears amplify the rotary motion by 2 and 2.5 times respectively. In addition the 1:100 gearhead amplifies the rotary motion of the output shaft to the generator. Thus the generator shaft will be rotating at 33333.3 rpm, by a factor of 500.

It must be noted that the generator shaft will only be rotating at that speed instantaneously. The main shaft speed only depends on the train speed. The main shaft of the lever mechanism will be rotating with this significant speed (33333.3 rpm) only for a period of approximately 0.04 seconds (the span of time from when the large roller contacts the train wheel flange to when it loses contact with the train wheel flange). Assuming the return mechanism is able to force the follower back to its initial position at the same speed, the generator shaft will rotate at the same speed of 33333.3 for another 0.04 second. There are 4 wheels per every side of a railcar, therefore the generator shaft and all other rotating components will be rotating overall for only 0.32 seconds. The maximum generator speed of 33333.333 rpm is equivalent to 555.6 revolutions per second. The main shaft will be rotating only for a period of 0.32 seconds. The product of the main shaft rotation period and generator speed gives an average number of revolutions per train wheel that the generator would experience. For a train traveling at a speed of 11.5 mph, the average number of revolutions per train wheel that the generator will rotate is calculated to be 177.8 revolutions.

The period or frequency of railcars traveling over the device is determined by calculating the time interval between couplers of a railcar. The distance between coupler faces of coal cars is
measured to be 53 feet [20]. For a loaded train traveling at 11.5 mph, the period is calculated to be 3.13 seconds using Equation 4.4.

$$t = \frac{l}{v} \quad (4.4)$$

where $t$ is the period of railcars traveling over the device, $l$ is the distance between coupler faces, and $v$ is the speed at which the train is traveling.

To determine the average speed at which the generator will rotate for a loaded train traveling at 11.5 mph, the average number of generator revolutions per railcar was divided by the period of railcars traveling over the device. Thus the average generator speed is calculated to be approximately 24 revolutions per second, which is equivalent to 1440 rpm.

Regarding the output power curve for the 443541 permanent magnet DC generator [22], for the average generator speed of 1500 rpm, the generator is able to produce 50 W. Also it must be noted that output power was calculated not taking into account frictional losses in the whole mechanism and efficiency of the gear head and generator.

Comparing the calculated results of generator speed and output power between the ramp lever mechanism and rack gear mechanism, driving the improved mechanical device by the passage of each train wheel and harnessing the vertical deflection of railroad track respectively, the ramp lever mechanism approach is capable of generating four times as much electricity as the rack gear system. Since the contact period is relatively short (about 3 seconds), and it may cause fast accelerations in the generator and other mechanical components, applying a clutched torsion spring to store and smoothly transmit the rotational energy into the system could be beneficial.
Chapter 5. Hydraulic energy harvesting system

5.1 Introduction

The major drawbacks of previous approaches discussed in the previous section are: limited power production for unloaded train conditions, and insufficient power to illuminate grade crossing lights under loaded trains traveling at relatively low speeds. The hydraulic system improves on the mentioned shortcomings, intended to harvest power on the order of 10 W and 40 W under unloaded and loaded train conditions, traveling at 11.5 and 13.5 mph respectively (on the order of 20% of maximum speed).

To harvest decent amounts of power from unloaded train conditions, it is essential to amplify the vertical deflection of the railroad track. One way to accomplish this is to deploy a hydraulic cylinder with relatively large bore diameter, mounted under the bottom of the rail such that it is compressed due to vertical deflection of the rail caused by railcar traffic. Therefore, even for small deflection of the railroad track, the volume displacement of the cylinder will be substantial due to its comparatively large bore diameter. Applying a hydraulic motor with smaller displacement than the cylinder, the vertical deflection is amplified depending on the ratio of the cylinder displacement to the motor displacement. With these considerations in mind, a hydraulic energy harvesting system was designed.

5.2 System design

The system, as illustrated in Figure 5.1, consists of a hydraulic cylinder, a hydraulic motor, a check valve and a reservoir tank.
The hydraulic cylinder is compressed by the passage of each railcar axle. To force the piston back to its initial position and keep it in contact with the bottom of the rail, use of a single-acting, spring-loaded cylinder was initially investigated. In most commercial cylinders the spring was located on the rod side of the piston, for retracting rather than extending the piston rod, which was not suitable for our purpose. There existed a few spring-loaded cylinders with the spring located on the blank side, but they were mostly custom made. However, a helical compression spring with a spring rate of 617 lbs/inch was selected and integrated to a 4 inch bore, double-acting cylinder, both purchased off the shelf. The compression spring pushes the piston up, relaxing the pressure in the cylinder.

To convert the linear motion of the cylinder to rotary motion as well as convert the fluid pressure in the cylinder to torque, a compact hydraulic motor with displacement of 0.5 cu in/rev was deployed.
The hydraulic cylinder is compressed and relaxed by the passage of each railcar axle, acting like a pump and forcing the hydraulic fluid towards the hydraulic motor, converting the hydraulic pressure and flow into rotational motion and torque. The unpressurized fluid from the output port of the hydraulic motor is directed towards a 2-gallon reservoir tank. A poppet check valve with low cracking pressure was used to fill up the bottom chamber of the cylinder during extension. The reservoir tank should be located about 11 feet higher than the check valve to provide cracking pressure to open the check valve. An alternative solution could be to use a pressurized reservoir tank to supply the cracking pressure for the check valve. Although the cylinder functions as a single-acting cylinder, to lubricate and enhance the efficiency of the cylinder a line from the reservoir tank to the upper chamber was added. Then the rotary motion of the hydraulic motor is amplified via a planetary gearbox with a ratio of 1:25, turning a permanent magnet DC generator to produce power from downward displacement of the railroad track. Figure 5.2 shows a 3D model of the rotating part of the energy harvesting system.

Figure 5.2. The rotating part of the energy harvesting system.
The hydraulic energy harvesting system was fabricated for testing. Figures 5.3 and 5.4 show the whole hydraulic energy harvesting system and its main components. The cylinder is mounted under the bottom of the rail. The rest of the system is located about 10 feet away from the cylinder, giving a safe distance from the track to observe the rotating part of the system while it is operating. The reservoir tank is placed almost 11 feet higher than the check valve on a stand. Hydraulic hoses with swivel connectors are used to connect the hydraulic components.

Figure 5.3. Hydraulic energy harvesting system.
The hydraulic system is driven directly by the vertical deflection of the rail due to passing railcar traffic. The hydraulic cylinder is mounted under the bottom of the rail such that it is contracted by downward motion of the rail. The rest of hydraulic components are placed 10 feet away from the railroad track. The hydraulic cylinder is connected to a hydraulic system by hydraulic hoses. The vertical motion of the hydraulic cylinder is translated into rotational motion. The rotational motion is amplified to rotate a permanent magnet DC generator. Details of the design are presented in the following sections.

### 5.3 Lab testing

Lab tests were conducted to verify the functionality of the system and quantify the power output. To simulate the railroad track deflection and actuate the cylinder, an MTS machine with a capacity of 90 kN was used, shown in Figure 5.5. A National Instruments USB 6008 DAQ board and LabVIEW software, shown in Figure 5.6, were deployed to acquire the voltage. A hand-held digital tachometer was utilized to measure the angular velocity of the generator shaft.
Initially to test the functionality of the hydraulic part of the system and quantify the angular velocity of the hydraulic motor, the gearbox was decoupled and the testing was performed. The vertical deflection of the railroad track was simulated with sinusoidal cyclic
displacement provided by the MTS machine. Two different vertical displacements (0.25 and 0.50 inches [4]) were used as the magnitude of the sinusoidal displacement to simulate unloaded and loaded train conditions. Two different frequencies (0.319 and 0.375 Hz) at which the cylinder is compressed were used to simulate the frequencies at which the track is deflected for train speeds of 11.5 and 13.5 mph, representing loaded and unloaded train speeds respectively. The testing was performed and the hydraulic part of the energy harvesting system worked as expected. Table 1 shows the testing results.

Table 5.1. Testing results of hydraulics.

<table>
<thead>
<tr>
<th>Track deflection (inch)</th>
<th>Train Speed (mph)</th>
<th>Hydraulic motor average speed (rpm)</th>
<th>Train Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.25</td>
<td>13.5</td>
<td>Unloaded train</td>
</tr>
<tr>
<td>2</td>
<td>0.5</td>
<td>11.5</td>
<td>Loaded train</td>
</tr>
<tr>
<td>3</td>
<td>0.25</td>
<td>60</td>
<td>Unloaded train with possible maximum speed</td>
</tr>
<tr>
<td>4</td>
<td>0.5</td>
<td>60</td>
<td>Loaded train with possible maximum speed</td>
</tr>
</tbody>
</table>

The output power of the generator can be either stored into a battery or directly power the crossing lights and health monitoring sensors. Accordingly two different sets of testing conditions were used, one with battery, voltage regulator and two resistors in series, and one with four resistors in series.

5.3.1 Testing with energy stored into a battery

The harvested electrical energy can be stored in a rechargeable battery, which will supply the required power to illuminate crossing lights. A 12-volt lead-acid battery as well as a charge
controller, preventing overcharging and deep discharging to protect battery life and improve battery performance, were deployed in lab testing. Also a 20-Ω and a 1-Ω resistor in series (overall load of 21 Ω), representing the crossing lights and wire resistance, were connected to the DC generator. The voltage was measured across the 1-Ω resistor to scale the output voltage to within the specified range of the DAQ board (±10 V). The total output voltage and instantaneous power were calculated using Ohm’s law for resistors in series.

The testing was carried out for two different vertical deflections, 2.75 mm (0.108 inch) and 3.75 mm (0.147 inch) with a frequency of 0.375 Hz, representing the unloaded train speed of 13.5 mph.

The minimum deflection at which the generator rotated continuously was 2.75 mm (0.108 inch); however, for deflections smaller than 2.75 mm, the generator started spinning for a short period of time and then stopped. This could be a result of generated heat, hose expansions and internal loses of the hydraulic system. The output voltage and instantaneous power for the deflection of 2.75 mm are shown in Figure 5.7. The average power, illustrated as a constant value, was calculated to be 0.50 W using:

\[
\frac{\sum_{i=1}^{n} P_i}{n}
\]  

(1)

where \(P_i\) is the calculated instantaneous power, and \(n\) is the total number of data samples. The average angular velocity of the generator shaft measured by hand-held digital tachometer was approximately 114 rpm. The maximum force the MTS machine exerted to the hydraulic cylinder was 61 kN.
Figure 5.7. Output voltage and power for a track displacement of 2.75 mm and simulated train speed of 13.5 mph.

The maximum displacement which the MTS machine could provide for these testing conditions was 3.75 mm. The capacity of the MTS machine was 90 kN; for any displacements larger than 3.75 mm, greater force than 90 kN was needed. The output voltage and instantaneous power for the deflection of 3.75 mm are shown in Figure 5.8. The average power, illustrated as a
constant value, was calculated to be 2.73 W. The average angular velocity of the generator shaft was measured to be approximately 256 rpm. The maximum exerted force by the MTS machine was read to be 90 kN.

Figure 5.8. Output voltage and power for a track displacement of 3.75 mm and simulated train speed of 13.5 mph.
5.3.2 Testing with harvested power directly illuminating crossing lights

Powering the warning lamps directly without the need of an electrical storage scheme is an operation mode of interest. Accordingly, two 50-Ω resistors, a 21-Ω and a 10-Ω resistor, all in series such that an overall load of 131 Ω was applied, as shown in Figure 5.5, were connected to the generator to simulate a bank of lamps. The output voltage across the 10-Ω resistor was then acquired. Testing was repeated for the same conditions as described in the previous section.

The output voltage and instantaneous power for the deflection of 2.75 mm are shown in Figure 5.9. The average power was calculated to be 1.90 W. The average angular velocity of the generator shaft was measured to be approximately 165 rpm. The maximum exerted force by the MTS machine was read to be 57 kN.
Figure 5.9. Output voltage and power for a track displacement of 2.75 mm and simulated train speed of 13.5 mph.

The output voltage and instantaneous power for the deflection of 3.75 mm are shown in Figure 14. The average power was calculated to be 11.08 W. The average angular velocity of the generator shaft was measured to be approximately 500 rpm. The maximum exerted force by the MTS machine was read to be 89 kN.
The maximum possible speed which the train can reach is approximately 60 mph; thus the testing was carried out for the deflection of 2.75 mm with a frequency of 1.667 Hz, representing train speed of 60 mph. The output voltage and instantaneous power are shown in Figure 5.11. The average power was calculated to be 2.51 W. The average angular velocity of the generator shaft was measured to be approximately 260 rpm. The maximum exerted force by the MTS machine was read to be 65 kN. The MTS machine was not able to provide enough force for the deflection of 3.75 mm.
Figure 5.11. Output voltage and power for a track displacement of 2.75 mm and simulated train speed of 60 mph.

5.4 Discussion of test results

Comparing the output power of the hydraulic energy harvesting system for the similar deflections but different testing setup (sections 5.3.1 and 5.3.2), the hydraulic system produced almost 4 times more power in the second setup (directly powering the lights) than it generated for
the first setup (charging the battery) while the maximum force provided by the MTS machine remained about the same.

The current (amperage) of a DC generator only depends on the load applied to it, not on its speed [27]. On the other hand, the torque needed to turn a PMDC generator depends on the current it generates. Therefore the more resistive load put on the generator, the lower current it generates and the lower torque has to be supplied to turn it.

The generator is driven by the hydraulic motor coupled to a 1:25 ratio gearbox in our energy harvesting system; thus the generator shaft speed and the torque needed to turn it are determined by the hydraulic motor with scale of 25. The hydraulic motor speed is a function of input flow rate, its displacement and the mechanical load on it or the torque it supplies. Considering the flow rate and the displacement as constant, the higher the load on it or the more torque it supplies, the slower it rotates, mainly due to its internal losses.

Consequently, the more resistive load on the generator, the less current it generates, the less torque needed to turn it, and the faster the hydraulic motor rotates. The output power of the PMDC generator is proportional to its shaft speed [27]; thus the faster it rotates the more power it generates.

Accordingly, since the resistive load on the generator in the second testing setup (131 Ω) was about 5 times larger compared to the first setup (22 Ω), greater power was generated in the second setup compared to the first setup. Also, the output voltage of a permanent magnet DC generator is governed by its shaft rpm and the load applied to it [27]. This may explain why the acquired voltage is higher in the second setup compared to the first setup.
Chapter 6. Conclusion

The hydraulic energy harvesting system was designed and built to address all mentioned shortcomings and failures of previous attempts. The lab testing of the hydraulic system shows promising results. The measured railroad track deflection in previous field testing [20] was reported to be approximately 0.25 and 0.5 inch for unloaded and loaded trains respectively; a loaded and an unloaded railcar weigh about 50,000 and 280,000 lbs respectively, indicating decent displacement and force to actuate the hydraulic cylinder to harvest power on the order of 10-40 W under unloaded and loaded conditions, satisfying existing standards for illumination of warning lights.

Due to the weather conditions (flooding) and difficulties encountered in obtaining measurements of device behavior under real train traffic several times during past year at DeBruce crossing in Nebraska City, the desired field testing was not fulfilled. Therefore, coming to a thorough and conclusive statement about all the systems developed and discussed in this thesis is difficult; however, lab testing was performed for both the mechanical device and the hydraulic system and does provide significant insight.

To verify the functionality of both the mechanical power harvesting device and the hydraulic energy harvesting system, and quantify the output power production level under realistic conditions, further lab and field testing remain as future work.

Also, to optimize the generator performance for the mechanical energy harvesting device future work is needed to come up with an electric solution to lower the output voltage while high generator shaft speed is desirable. Similarly, for the hydraulic energy harvesting device, future work can include developing an electronic solution to transform the high-voltage and low-current
generator output to have a voltage near 12 V and a higher current on the order of 1 Amp for powering warning light systems.
References


Appendix A

Cost analysis
Table A.1. Vendor purchases for the mechanical power harvesting device.

<table>
<thead>
<tr>
<th>Component</th>
<th>Company</th>
<th>Model</th>
<th>Price</th>
<th>Qty</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>3/8&quot; x 3/4&quot; Stainless Steel beam</td>
<td>McMaster-Car</td>
<td>0992K134</td>
<td>62.19</td>
<td>2</td>
<td>124.38</td>
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<tr>
<td>12&quot; x 4&quot; x 1&quot; Aluminum Sheet</td>
<td>McMaster-Car</td>
<td>892150G335</td>
<td>61.01</td>
<td>1</td>
<td>61.01</td>
</tr>
<tr>
<td>Stainless steel pinion gear</td>
<td>Quality Transmission Component</td>
<td>KSUS2-20</td>
<td>50.05</td>
<td>1</td>
<td>50.05</td>
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<tr>
<td>Stainless steel rack gear</td>
<td>Quality Transmission Component</td>
<td>KSUR3-500</td>
<td>67.57</td>
<td>1</td>
<td>67.57</td>
</tr>
<tr>
<td>Stainless Steel Spur Gear</td>
<td>Quality Transmission Component</td>
<td>KSUS2-15</td>
<td>36.08</td>
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<td>108.24</td>
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<td>Stainless Steel Spur Gear</td>
<td>Quality Transmission Component</td>
<td>KSUS2-30</td>
<td>31.37</td>
<td>2</td>
<td>64.74</td>
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<td>Flexible Spider Shaft Coupling Hub</td>
<td>McMaster-Car</td>
<td>9845T402</td>
<td>26.9</td>
<td>4</td>
<td>107.6</td>
</tr>
<tr>
<td>Flexible Spider Shaft Coupling Spider</td>
<td>McMaster-Car</td>
<td>9845T21</td>
<td>9.64</td>
<td>2</td>
<td>19.28</td>
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<td>1/2&quot; x 13.5&quot; Stainless Steel Rod</td>
<td>McMaster-Car</td>
<td>09533X133</td>
<td>4.9</td>
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<td>3&quot; x 12&quot; Stainless Steel Rod</td>
<td>McMaster-Car</td>
<td>09535K751</td>
<td>87.79</td>
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<td>87.79</td>
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<td>25mm x 25mm Stainless Steel Rod</td>
<td>McMaster-Car</td>
<td>1372T31</td>
<td>7.9</td>
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<td>GMN Ball Bearing Chock FKEN 6205 ZZS</td>
<td>GMN Bearing</td>
<td>305006</td>
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<td>Ball Bearing Style Double Sealed</td>
<td>McMaster-Car</td>
<td>2780T59</td>
<td>11.54</td>
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<td>23.08</td>
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<td>Needle-Roller Bearing Style Double Sealed</td>
<td>McMaster-Car</td>
<td>59051K33</td>
<td>9.31</td>
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<td>Internal Retaining Rings</td>
<td>McMaster-Car</td>
<td>91500A171</td>
<td>3.5</td>
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<td>91500A122</td>
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<td>Gearhead 1:100</td>
<td>Ananth Automation</td>
<td>GBPH-0603-NS-100</td>
<td>290.4</td>
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<td>Permanent magnet/DC generator</td>
<td>Windstream, LLC</td>
<td>443541</td>
<td>249</td>
<td>1</td>
<td>249</td>
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<tr>
<td>Shaft coupling</td>
<td>McMaster-Car</td>
<td>61005K331</td>
<td>28.04</td>
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<td>56.08</td>
</tr>
<tr>
<td>Smal-Mount Ball Transfer</td>
<td>McMaster-Car</td>
<td>6460K51</td>
<td>37.55</td>
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<td>75.10</td>
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<tr>
<td>Auger bit</td>
<td>PENG0</td>
<td>95.5</td>
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<tr>
<td>Fasteners</td>
<td>McMaster-Car</td>
<td>59</td>
<td>1</td>
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<td>Other materials</td>
<td></td>
<td></td>
<td>150</td>
<td>1</td>
<td>150</td>
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<td><strong>Total</strong></td>
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<td></td>
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<td>2134.87</td>
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Table A.2. Vendor purchases for the Hydraulic Energy Harvesting System.

<table>
<thead>
<tr>
<th>Component</th>
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<th>Model</th>
<th>Price</th>
<th>Qty</th>
<th>Total</th>
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<tr>
<td>Hydraulic cylinder</td>
<td>Lehigh Fluidpower, Inc</td>
<td>H4050140AB1 700</td>
<td>$722.67</td>
<td>1</td>
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<td>Check valve</td>
<td>swagelok</td>
<td>SS-8C4-1/3</td>
<td>$97.80</td>
<td>1</td>
<td>$97.80</td>
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<td>Hydraulic motor</td>
<td>Grainger</td>
<td>5PZK7</td>
<td>$297.00</td>
<td>1</td>
<td>$297.00</td>
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<tr>
<td>Hydraulic reservoir</td>
<td>Grainger</td>
<td>1DMV5</td>
<td>$200.00</td>
<td>1</td>
<td>$200.00</td>
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<td>Brass ball valve</td>
<td>McMaster-Carr</td>
<td>47865K43</td>
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<td>Brass ball valve,</td>
<td>McMaster-Carr</td>
<td>47865K45</td>
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<td>Hydraulic fluid, ISO 32</td>
<td>Northern Tool</td>
<td>159808</td>
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<td>Compression spring</td>
<td>McMaster-Carr</td>
<td>96483K187</td>
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<td>McMaster-Carr</td>
<td>90475A036</td>
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<td>Washer</td>
<td>McMaster-Carr</td>
<td>91117A240</td>
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Total: $2,403.79
Appendix B

Engineering drawings
B.1 Designed Components for the Mechanical Energy Harvesting Device
ALL DIMENSIONS ARE INCHES

2x 0.4375
2x 0.625
2x 0.625
2.500
1.250
0.250

P.C.D  φ2.7559
1.250

H7
4x φ0.2165

 MATERIAL: ALUMINUM
PART: GEAR HEAD BRACKET
QUANTITY: 1
MATERIAL: STAINLESS STEEL
PART: GEAR HEAD OUTPUT SHAFT
QUANTITY: 1

ALL DIMENSIONS ARE INCHES
ALL DIMENSIONS ARE INCHES

MATERIAL: STAINLESS STEEL
PART: CLUTCH INPUT SHAFT
QUANTITY: 2

AT CENTER POINT, BORE TO A DEPTH OF 0.079 INCHES (2 mm).
B.2 Designed Components for the Hydraulic Energy Harvesting System
Appendix C

Technical details of commercial components
Mechanical energy harvesting device:

C.1 Generator

Windstream Power LLC Permanent Magnet DC Generator (443541)
- Magnets: Two high-energy saturated C3 ceramic magnets.
- Shaft: Steel 12.7mm (1/2") diameter, 40mm length, with 1mm full-length flat.
- Armature: 16-slot armature 52mm diameter wound with AWG25 magnet wire (fusing current: 24 amps).
- Brushes: Extra-long 8x14mm brush assemblies including spring, pigtail, and cap - replacement stock no. 443729
- Bearings: Two double-sealed 32mm OD ball bearings - replacement stock no. 171110
- Rotation: Either direction - The red output wire is positive for clockwise rotation from the shaft end.
- Speed: Zero to 5,000 rpm - generates at all speeds - depends on load.
  Mounting: Four 6mm holes on the front or rear end caps, or by hose clamps on the magnet drum.
- Weight: 4.2Kg (9.2lb) Shipping weight 4.6Kg (10lb), dimensions 150x150x300mm (6x6x12in).
- Resistance: Internal resistance 7.7 ohms. Inductance 16mH.
- Performance: See performance curves below
- Maximum charging current is 10A (see run times below) If you exceed these specifications the generator will overheat.
- Generator is not weather proof and needs to be enclosed if outside.

Generators are non-returnable

10A - 5 minute run time
4A - 30 minute run time
3.3A - 60 minute run time
3A - CONTINUOUS DUTY
C.2 Gearhead

GBPH-060x-CS Series 1:100 Planetary Gearhead (GBPH-0602-NS-100)

The GBPH060-CS Series Planetary Gearbox offers you the precision you need at the prices you want! This Planetary Gearbox is designed for motion control, automation, and robotic applications requiring moderate backlash and stiffness. This Planetary Gearbox will match your servo or stepping motor and is ideal for NEMA 23 motors. Built with the design goal of offering a cost effective product without sacrificing quality, you will find that this Planetary Gearbox offers you an extraordinary value. You will find online prices on this website along with all the information you need to know to select the perfect Planetary Gearbox for your applications.

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<th>Model #</th>
<th>Gear Ratio (21:1)</th>
<th>Nominal Output Torque (oz-in)</th>
<th>Max Output Torque (in-lb)</th>
<th>Maximum Input Speed (RPM)</th>
<th>Stages</th>
<th>Backlash (arc-min)</th>
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<th>L2 Length (mm)</th>
<th>Height (in)</th>
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* xxxx denotes motor pilot, yyy denotes input shaft diameter.
C.3 Bearing Clutch

GMN Ball Bearing Clutch FKNN 6205 2RS (305996)

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C.4 Rack Gear

Quality Transmission Components Stainless Steel Rack Module 2 (KSUR2-500)

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<th>Module</th>
<th>Total length</th>
<th>Face width</th>
<th>Height to pitch line</th>
<th>Effective No. of teeth</th>
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<th>Allowable force (kgf)</th>
<th>Allowable force (N)</th>
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**Specifications**
- Precision grade: KHK R.001 grade 5
- Gear teeth: Standard full depth
- Pressure angle: 20°
- Material: SUS304 Stainless steel
- Heat treatment: Solution heat treatment
- Core hardness: less than HB187
- Surface hardness: -
- Surface treatment: Passivation
- Surface finish: Hobbed
- Datum reference surface for gear cutting: Bottom surface

C.5 Pinion Gear

Quality Transmission Components Stainless Steel Spur Gear Module 2

1- (KSUS2-20)
2- (KSUS2-15)
3- (KSUS2-30)

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<th>Bore</th>
<th>Hub dia.</th>
<th>Pitch dia.</th>
<th>Outside dia.</th>
<th>Face width</th>
<th>Hub width</th>
<th>Total length</th>
<th>Shape</th>
<th>Allowable torque (N.m)</th>
<th>Allowable torque (N.m)</th>
<th>Backlash (mm)</th>
<th>Weight (kgf)</th>
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**Specifications**
- Precision grade: JIS Grade NS38 (JIS B1711:1983)
- Core hardness: less than HB187
- Gear teeth: Standard full depth
- Pressure angle: 20°
- Material: SUS303 Stainless Steel
- Surface treatment: -
- Heat treatment: Solution heat treatment
- Surface finish: Hobbed
- Datum reference surface for gear cutting: Bore

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<th>Hub dia.</th>
<th>Pitch dia.</th>
<th>Outside dia.</th>
<th>Face width</th>
<th>Hub width</th>
<th>Total length</th>
<th>Shape</th>
<th>Allowable torque (N.m)</th>
<th>Allowable torque (N.m)</th>
<th>Backlash (mm)</th>
<th>Weight (kgf)</th>
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<th>Pitch dia.</th>
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C.6 Flexible Shaft Coupling

Zero-Backlash Flexible Spider Shaft Coupling (Coupling Hub 9845T402 / Spider 9845T21)

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Shown Assembled
C.7 Bearings

Ball Bearings (2780T59)

In stock for $13.18

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ABEC Precision Bearing Rating: Not Rated

Dynamic Radial Load Capacity, lbs.: 1,878

Maximum rpm: 5,000

Temperature Range: -20°F to +250°F

Bearing Material: Steel

Seal Material: Synthetic Rubber

Specifications Met: Not Rated

Note: Bearing comes greased.
**Needle Roller Bearings (5905K133)**

In stock for $9.69

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<td>For Shaft Diameter</td>
<td>1/2&quot;</td>
</tr>
<tr>
<td>Outside Diameter</td>
<td>11/16&quot;</td>
</tr>
<tr>
<td>Width</td>
<td>5/8&quot;</td>
</tr>
<tr>
<td>ABEC Precision Bearing Rating</td>
<td>Not Rated</td>
</tr>
<tr>
<td>Dynamic Radial Load Capacity, lbs</td>
<td>1,560</td>
</tr>
<tr>
<td>Maximum rpm</td>
<td>15,000</td>
</tr>
<tr>
<td>Temperature Range</td>
<td>-6° to +212° F</td>
</tr>
<tr>
<td>Bearing Material</td>
<td>Steel</td>
</tr>
<tr>
<td>Seal Material</td>
<td>Rubber</td>
</tr>
<tr>
<td>Specifications Met</td>
<td>Not Rated</td>
</tr>
</tbody>
</table>

**Note:** Has a large grease reservoir. Bearing comes greased.
C.8 Auger Bit

CS Auger

◊ Featuring the NEW 35 Series Fas-N-Lok teeth, eliminates the need for RubR-Lok!

◊ Single flights available in lengths of 36", 32", and 48".


<table>
<thead>
<tr>
<th>Part No.</th>
<th>Description</th>
<th>Diameter</th>
<th>Length</th>
<th>Flight Type</th>
<th>Hub</th>
<th>Weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>500205</td>
<td>2-02-06</td>
<td>2&quot;</td>
<td>30&quot;</td>
<td>Single</td>
<td>7/8&quot; sq. OH</td>
<td>9.0</td>
</tr>
</tbody>
</table>
C.9 DAQ Board

National Instruments USB-6229 M Series Multifunction DAQ (195840A-01)

NI USB-6229

16-Bit, 250 kS/s M Series Multifunction DAQ, External Power

- 32 analog inputs (16-bit, 250 kS/s)
- 4 analog outputs (16-bit, 533 kS/s), 46 digital I/O (32 at up to 1MHz), and 32-bit counters
- NI Signal Streaming for sustained high-speed data streams over USB; OEM version available
- Compatible with LabVIEW, LabWindows/CVI, and Measurement Studio for Visual Studio .NET
- NI-DAQmx driver software and NI LabVIEW SignalExpress LE interactive data-logging software

Overview

The National Instruments USB-6229 is a USB high-performance M Series multifunction data acquisition (DAQ) module optimized for superior accuracy at fast sampling rates.

The USB-6229 is designed specifically for mobile or space-constrained applications. Plug-and-play installation minimizes configuration and setup time, while direct screw-terminal connectivity helps keep costs down and simplifies signal connections.

This USB-6229 also features the new NI Signal Streaming technology, which provides DMA-like bidirectional high-speed streaming of data across the USB bus. For more information about NI signal streaming, check the Resources tab.

NI also offers a USB-6229 OEM version. Check the Resources tab or use the left navigation to get pricing and technical information.

Driver Software

NI-DAQmx driver and measurement services software provides easy-to-use configuration and programming interfaces with features such as DAQ Assistant to help reduce development time. Browse the information in the Resources tab to learn more about driver software or download a driver. M Series devices are not compatible with the traditional NI-DAQ (Legacy) driver.

Application Software

Every M Series data acquisition device includes a copy of NI LabVIEW SignalExpress so you can quickly acquire, analyze, and present data without programming. In addition to LabVIEW SignalExpress, M Series data acquisition devices are compatible with the following versions (or later) of NI application software – LabVIEW 7.1, LabWindows™/CVI 7.x, or Measurement Studio 7.x. M Series data acquisition devices are also compatible with Visual Studio .NET, C/C++, and Visual Basic 6.
## Specifications

### Specifications Documents
- Detailed Specifications
- Data Sheet

### Specifications Summary

<table>
<thead>
<tr>
<th>General</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Form Factor</td>
<td>USB</td>
</tr>
<tr>
<td>Operating System / Target</td>
<td>Windows</td>
</tr>
<tr>
<td>Measurement Type</td>
<td>Quadrature encoder, Voltage</td>
</tr>
<tr>
<td>DAQ Product Family</td>
<td>M Series</td>
</tr>
</tbody>
</table>

### Analog Input

<table>
<thead>
<tr>
<th>Number of Channels</th>
<th>32 SE/16 DI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sample Rate</td>
<td>250 kS/s</td>
</tr>
<tr>
<td>Resolution</td>
<td>16 bits</td>
</tr>
<tr>
<td>Simultaneous Sampling</td>
<td>No</td>
</tr>
<tr>
<td>Maximum Voltage Range</td>
<td>-10..10 V</td>
</tr>
<tr>
<td>Range Accuracy</td>
<td>3100 μV</td>
</tr>
<tr>
<td>Range Sensitivity</td>
<td>97.6 μV</td>
</tr>
<tr>
<td>Minimum Voltage Range</td>
<td>-200..200 mV</td>
</tr>
<tr>
<td>Range Accuracy</td>
<td>112 μV</td>
</tr>
<tr>
<td>Range Sensitivity</td>
<td>5.2 μV</td>
</tr>
<tr>
<td>Number of Ranges</td>
<td>4</td>
</tr>
<tr>
<td>On-Board Memory</td>
<td>4096 samples</td>
</tr>
</tbody>
</table>

### Analog Output

<p>| Number of Channels     | 4             |
| Update Rate            | 833 kS/s      |
| Resolution             | 16 bits       |
| Maximum Voltage Range  | -10..10 V     |
| Range Accuracy         | 3230 μV       |</p>
<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Current Drive (Channel/Total)</td>
<td>5 mA</td>
</tr>
<tr>
<td>Digital I/O</td>
<td></td>
</tr>
<tr>
<td>Number of Channels</td>
<td>48 DIO</td>
</tr>
<tr>
<td>Timing</td>
<td>Hardware, Software</td>
</tr>
<tr>
<td>Maximum Clock Rate</td>
<td>1 MHz</td>
</tr>
<tr>
<td>Logic Levels</td>
<td>TTL</td>
</tr>
<tr>
<td>Maximum Input Range</td>
<td>0.5 V</td>
</tr>
<tr>
<td>Maximum Output Range</td>
<td>0.5 V</td>
</tr>
<tr>
<td>Input Current Flow</td>
<td>Sinking, Sourcing</td>
</tr>
<tr>
<td>Programmable Input Filters</td>
<td>Yes</td>
</tr>
<tr>
<td>Output Current Flow</td>
<td>Sinking, Sourcing</td>
</tr>
<tr>
<td>Current Drive (Channel/Total)</td>
<td>24 mA, 806 mA</td>
</tr>
<tr>
<td>Watchdog Timer</td>
<td>No</td>
</tr>
<tr>
<td>Supports Programmable Power-Up States?</td>
<td>Yes</td>
</tr>
<tr>
<td>Supports Handshaking I/O?</td>
<td>No</td>
</tr>
<tr>
<td>Supports Pattern I/O?</td>
<td>Yes</td>
</tr>
<tr>
<td>Counter/Timers</td>
<td></td>
</tr>
<tr>
<td>Number of Counter/Timers</td>
<td>2</td>
</tr>
<tr>
<td>Resolution</td>
<td>32 bits</td>
</tr>
<tr>
<td>Maximum Source Frequency</td>
<td>80 MHz</td>
</tr>
<tr>
<td>Minimum Input Pulse Width</td>
<td>12.5 ns</td>
</tr>
<tr>
<td>Logic Levels</td>
<td>TTL</td>
</tr>
<tr>
<td>Maximum Range</td>
<td>0.5 V</td>
</tr>
<tr>
<td>Timebase Stability</td>
<td>50 ppm</td>
</tr>
<tr>
<td>GPS Synchronization</td>
<td>No</td>
</tr>
<tr>
<td>Pulse Generation</td>
<td>Yes</td>
</tr>
<tr>
<td>Buffered Operations</td>
<td>Yes</td>
</tr>
<tr>
<td>Debouncing/Glitch Removal</td>
<td>Yes</td>
</tr>
<tr>
<td>Timing/Triggering/Synchronization</td>
<td></td>
</tr>
<tr>
<td>Triggering</td>
<td>Digital</td>
</tr>
</tbody>
</table>
Hydraulic Energy Harvesting System

C.10 Hydraulic Cylinder

JHDH Series Hydraulic cylinder 4” bore (H40S0140AB1700). LEHIGH POWER FLUID, INC
STYLE 17
SQUARE FLANGE ROD END
(NFPA MOUNTING STYLE MF6)

<table>
<thead>
<tr>
<th>BORE</th>
<th>E</th>
<th>EE</th>
<th>E NPTF</th>
<th>SAE</th>
<th>F</th>
<th>FB</th>
<th>G</th>
<th>J</th>
<th>K</th>
<th>R</th>
<th>TF</th>
<th>UF</th>
<th>LB</th>
<th>P</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>4 1/2</td>
<td>1/2</td>
<td>10</td>
<td>5/8</td>
<td>7/16</td>
<td>1 3/4</td>
<td>1 1/4</td>
<td>3/8</td>
<td>3.32</td>
<td>5 7/16</td>
<td>6 1/4</td>
<td>4 7/8</td>
<td>2 5/8</td>
<td></td>
</tr>
</tbody>
</table>
C.11 Check Valve

Female NPT 1/2 in. C series check valve (SS-8C4-1/3)
C.12 Hydraulic Motor

Hydraulic Motor, 0.5 cu in/rev, 5 Bolt (5PZK7)

Hydraulic Motor, Displacement 0.5 cu in/rev, Mount 5 Bolt, 1.25 In Pilot, Keyed Shaft Type, 1932 Max RPM, 9/16-18 UNF Port Thread, Pressure Rating 2400 PSI, Shaft Dia 5/8 In, Shaft Length 1.03 In, Min Oil Viscosity 70 SUS, Max Oil Temp 100 F., Length 4.09 In, Width 2.37 In, Height 2.37 In

<table>
<thead>
<tr>
<th>Item</th>
<th>Hydraulic Motor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement (Cu. In./Rev.)</td>
<td>0.5</td>
</tr>
<tr>
<td>Pressure Rating (PSI)</td>
<td>2400</td>
</tr>
<tr>
<td>Shaft Type</td>
<td>Keyed</td>
</tr>
<tr>
<td>Port Thread</td>
<td>9/16-18 UNF</td>
</tr>
<tr>
<td>Port Type</td>
<td>Side</td>
</tr>
<tr>
<td>Min. Oil Viscosity</td>
<td>70 SUS</td>
</tr>
<tr>
<td>Max. Oil Temp. (F)</td>
<td>180</td>
</tr>
<tr>
<td>Flange Mount</td>
<td>5 Bolt, 1.25&quot; Pilot</td>
</tr>
<tr>
<td>Max. RPM</td>
<td>1932</td>
</tr>
<tr>
<td>Shaft Dia. (In.)</td>
<td>5/8</td>
</tr>
<tr>
<td>Shaft Length (In.)</td>
<td>1.03</td>
</tr>
<tr>
<td>Length (In.)</td>
<td>4.09</td>
</tr>
<tr>
<td>Width (In.)</td>
<td>2.37</td>
</tr>
<tr>
<td>Height (In.)</td>
<td>2.37</td>
</tr>
<tr>
<td>Series</td>
<td>J</td>
</tr>
</tbody>
</table>
C.13 Hydraulic Reservoir

Hydraulic Reservoir, 2 Gal Capacity (1DMV5)

Hydraulic Reservoir, Nominal Capacity 2 Gal, Return Port 1/2 NPT In, Suction Port Size 1 NPT In, Fill/Breather Cap, 11 Gauge Steel Construction, 1/2 NPT Drain Plug, Black Industrial Enamel Paint Finish, Buna-N Cover Gasket Gasket, Vertical Mounting, Removable Cover, Length 10 In., Height 10 In., Width 9 In., Includes Removable Cover, Oil Sight Gauge w/Thermometer, Mounting Feet, Filler Breather, Suction/Return Ports, and Drain

<table>
<thead>
<tr>
<th>Item</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item #</td>
<td>1DMV5</td>
</tr>
<tr>
<td>Price (ea.)</td>
<td>$269.00</td>
</tr>
<tr>
<td>Brand</td>
<td>LUBE DEVICES</td>
</tr>
<tr>
<td>Mfr. Model #</td>
<td>VGA7958</td>
</tr>
<tr>
<td>Ship Qty</td>
<td>1</td>
</tr>
<tr>
<td>Sold Qty. (Will-Call)</td>
<td>1</td>
</tr>
<tr>
<td>Ship Weight (lbs.)</td>
<td>21.02</td>
</tr>
<tr>
<td>Usually Ships</td>
<td>Today</td>
</tr>
<tr>
<td>Catalog Page No</td>
<td>3696</td>
</tr>
<tr>
<td>Country of Origin</td>
<td>USA</td>
</tr>
</tbody>
</table>
### C.14 Hydraulic Reservoir

Compression Spring (96485K187)

<table>
<thead>
<tr>
<th>Type</th>
<th>Compression Springs</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>Steel</td>
</tr>
<tr>
<td>Steel Type</td>
<td>Spring-Tempered Steel</td>
</tr>
<tr>
<td>System of Measurement</td>
<td>Inch</td>
</tr>
<tr>
<td>Outside Diameter</td>
<td>2-7/16'</td>
</tr>
<tr>
<td>Wire Size</td>
<td>3/8''</td>
</tr>
<tr>
<td>Overall Length</td>
<td>4''</td>
</tr>
<tr>
<td>Compressed Length</td>
<td>2.72'</td>
</tr>
<tr>
<td>Ends</td>
<td>Closed and Ground</td>
</tr>
<tr>
<td>Wire Type</td>
<td>Round Wire</td>
</tr>
<tr>
<td>Load</td>
<td>780 lbs.</td>
</tr>
<tr>
<td>Deflection at Load</td>
<td>1.28'</td>
</tr>
<tr>
<td>Rate</td>
<td>617 lbs/inch</td>
</tr>
<tr>
<td>Specifications Met</td>
<td>Not Rated</td>
</tr>
</tbody>
</table>
Wire Diameter: 0.375

Dimensions:
- Length: 4"