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# Optimizing Power Consumption of Freight Railroad Bearings Using Laboratory Experimental Data

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# OPTIMIZING POWER CONSUMPTION OF FREIGHT RAILROAD BEARINGS USING LABORATORY EXPERIMENTAL DATA

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## ABSTRACT

*Based on projected freight truck fuel efficiency, freight railroad and equipment suppliers need to identify, evaluate and implement technologies and/or operating practices to maintain traditional railroad economic competitiveness. The railway industry uses systems that record the total energy efficiency of a train but not energy efficiency or consumption by components. Lowering the energy consumption of certain train components will result in an increase in its overall energy efficiency, which will yield cost benefits for all the stakeholders. One component of interest is the railroad bearing whose power consumption varies depending on several factors that include railcar load, train speed, condition of bearing whether it is healthy or defective, and type of defect. Being able to quantify the bearing power consumption, as a function of the variables mentioned earlier, would make it possible to obtain optimal operating condition ranges that minimize energy consumption and maximize train energy efficiency.*

*Several theoretical studies were performed to estimate the power consumption within railroad bearings, but those studies lacked experimental validation. For almost a decade now, the University Transportation Center for Railway Safety (UTCRS) at the University of Texas Rio Grande Valley (UTRGV) has been collecting power consumption data for railroad bearings under various loads, speeds, ambient temperatures, and bearing condition. The objective of this ongoing study is to use the experimentally acquired power consumption to come up with a correlation that can be used to quantify the bearing power consumption as a function of load, speed, ambient temperature, and bearing condition. Once obtained, the model can then be used to determine optimal operating practices that maximize the railroad bearing energy efficiency. In addition, the developed model will provide insight into possible areas of improvement for the next generation of energy efficient railroad bearings. This paper will discuss ongoing work including experimental setup and findings of energy consumption of bearings as function of railcar load, train speed, condition of bearing whether it is healthy or defective, and type of defect. Findings of energy consumption are converted into approximations of diesel gallons to quantify the effect of nominal energy consumption of the bearings and show economic value and environmental impact.*

## INTRODUCTION

Freight railroad has a diminishing fuel economy competitiveness advantage over freight trucking. Freight trains are known for their locomotive advancements, capable of hauling several tons of goods for several miles with limited fuel consumption. In fact, literature shows that the freight railroad competitiveness advantage is due to, among other factors, the reduction in friction that is created from the wheel assembly contacting the rails, as well as aerodynamics, and engine efficiency advancements [1]. These are the main factors that kept the freight train industry more efficient than trailer trucks. For example, a freight train hauling 3000 tons of material for 500 miles would only consume 3185 gallons of diesel. The 471 ton-miles per gallon performance by freight trains is about 3.5 times more efficient than the performance of trailer trucks that typically run at 134 ton-miles per gallon [2]. However, over the past decade, there have been major efforts to improve the efficiency of trailer trucks. In some studies, researchers have suggested switching to electrical power trailer trucks, as well as to create self-driving trailers that have the ability to synchronize with other trucks allowing them to travel with shorter distances between the trucks to improve the aerodynamics, thus, making them more fuel efficient [3-4].

For freight trains to remain a viable competitor of trailer trucks, constant enhancements and advancements must be made to maintain the competitive edge. A mixture of analytical models coupled with experimental testing can yield favorable results to ensure that trains are performing optimally. Some current analytical modeling of railroad fuel consumption involves a multi-step process. One of the initial steps in the process is being able to estimate the required number of locomotives needed to effectively move the train to its destination. Calculating the fuel consumed during acceleration and determining the resistance forces are other steps in this process. There are several equations that have been developed and are widely used in the field. These equations consider the resistance from drag force which varies with speed, along with wheel rolling resistance, flange resistance, among other factors. Note that these equations assume the tapered roller bearing resistance to be constant and not varying with speed [5], which is not the case. The lack of experimental testing and analysis performed solely on the bearing components is the reason behind the simplified models used. Hence, rigorous experimental testing is essential for quantifying the frictional heating within a bearing as a function of speed in order to optimize the fuel efficiency.

In addition, the condition of the individual freight railcar can also significantly impact the fuel efficiency of the total system. Fright railcar suspension consists of several components such as side frames, springs, dampers, wheels, axles, and tapered roller bearings. Of these components, the bearings are the most susceptible to develop defects at high speeds under heavy cargo loads. The fundamental components of a railroad bearing are the rollers, inner rings (cones), and outer ring (cup). Under optimal conditions, these components produce near-frictionless motion. However, their effectiveness can be compromised under abnormal operating conditions. Deformations in the rollers, cups or cones can result in an increase in frictional heating especially if the bearing develops a defect on any of the raceways [6]. There are two bearings per axle and four axles per wagon in a typical freight railcar. Freight trains can haul up to 59 wagons, which corresponds to a total of 472 bearings. When hauling up to 18,000 tons, even a small change in the condition of the bearings can potentially result in significant differences in the energy efficiency.

To date, very few power consumption studies targeting specific railroad components have been performed. To address this, the University Transportation Center for Railway Safety (UTCRS) research team has been investigating the power consumption of railroad tapered-roller bearings. The ongoing work presented in this paper focuses on finding correlations for the bearing power consumption as a function of load, speed, ambient temperature, and bearing condition.

## EXPERIMENTAL SETUP & PROCEDURES

The UTCRS four-bearing dynamic test rig pictured in Figure 1 was used to perform the experiments for this study. This test rig can accommodate both Class F (6 ½" × 12") and Class K (6 ½" × 9") tapered-roller bearings. The hydraulic cylinder can apply a load of 153 kN (34.4 kip) per bearing, corresponding to a fully loaded railcar, but can go up to 175% of this full load. The data for this study was obtained from testing at full load (100%). Additionally, the bearing tester utilizes a 22 kW (30 hp) variable speed motor that can simulate different train speeds. The speeds used in this study are listed in Table 1. The test bearings were air-cooled utilizing three industrial size fans which simulated a crosswind having an average speed of 6 m/s (13.4 mph).

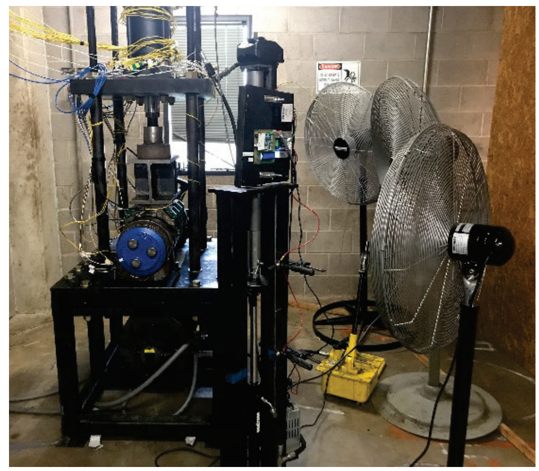

*Figure 1: Four-Bearing Test Rig (4BT)*

*Table 1: Typical speeds used to perform the experiments in this study*

| Axle Speed [rpm] | Track Speed [mph] | Track Speed [km/h] |
|---|---|---|
| 280 | 30 | 48 |
| 420 | 45 | 72 |
| 560 | 60 | 97 |

To simulate field service conditions, only data collected from the middle two bearings was used in this study because these bearings are top loaded (refer to Figure 2) as is the case in field service. Figure 3 shows the locations of the three accelerometers used to acquire the vibration signatures within the bearing. These locations are the Smart Adapter (SA), Mote (M), and Radial (R). The steel adapters for the middle two bearings (B2 and B3) were machined to accommodate two 70g accelerometers (affixed to the SA and M locations), a 500g accelerometer (placed on the R location), and a regular K-type thermocouple aligned with two bayonet thermocouples placed in the middle of the bearing cup width and held in place by a hose clamp. A schematic of the test axle along with sensor locations is provided in Figure 4.

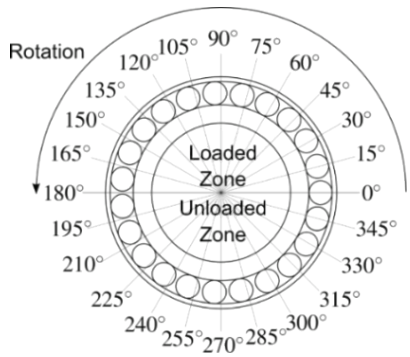


*Figure 2: Schematic diagram showing the loading zones*

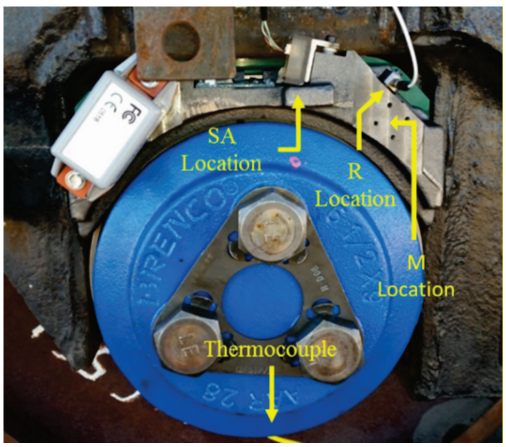


*Figure 3: Modified bearing adapter showing sensor locations*

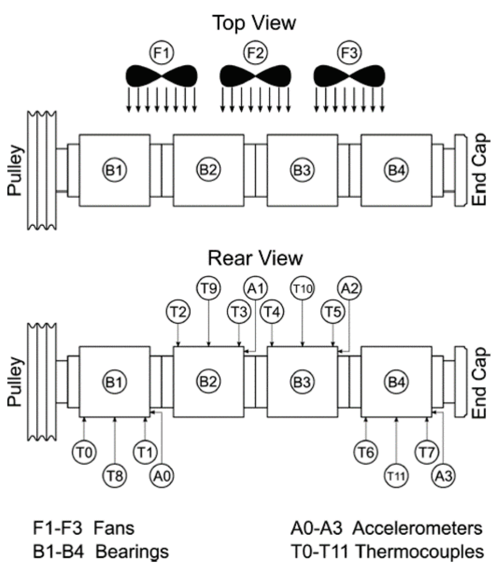


*Figure 4: Top and rear views of 4BT including senor locations*

The National Instruments (NI) PXIe-1062Q data acquisition system (DAQ) programmed using LabVIEW™ was utilized to collect the data for this study. A NI TB-2627 card was used to record the thermocouple temperature readings at a sampling rate of 128 Hz for 0.5 seconds in 20-second intervals.

## ASSUMPTONS

The power consumption calculations in the results presented here are neglecting the power loss in the pully system used in the experimental setup to transfer power from the motor to the test axle with the four bearings. The following process is used to convert the experimental power consumption into gallons of diesel [7]. First, the fuel flow to the engine is calculated using Eq. [1] as follows,

$$\dot{m} = \frac{b_e \cdot P_e}{3600\frac{s}{h} \cdot 1000\frac{g}{kg}} \qquad \text{Eq. (1)}$$

where $\dot{m}$ is fuel flow to the engine in [$kg/s$], $b_e$ is brake specific fuel consumption of the engine in [$g/kWh$] (assumed 224 $g/kWh$), and $P_e$ is the engine power in [$kW$]. Then, Eq. (2) is used to estimate the gallons of diesel as follows,

$$G = \dot{m} \cdot t_e * 0.3105\,\frac{gallon_{diesel}}{kg} \qquad \text{Eq. (2)}$$

where $G$ is the gallons of diesel; and $t_e$ is the total running time of the experiment in seconds.

Now, to calculate the miles per gallon (MPG) and the ton-mile per gallon given in Table 2, the following equations are used,

$$MPG = \frac{Total\ Miles\ Run}{G} \qquad \text{Eq. (3)}$$

$$\frac{ton{\cdot}mile}{gallon} = \frac{MPG \times Load\ in\ lb_f}{2000\ lb_f\ per\ ton} \qquad \text{Eq. (4)}$$

The laboratory results presented in Table 2 through Table 7 do not take into account the resistance caused by drag forces. Table 8 presents simulations of the power consumption and energy efficiency of all the bearings within the train consist based on number of wagons proposed. Each simulated wagon contains four axles for a total of eight bearings per wagon. Therefore, to simulate one wagon, the experimental power consumption obtained from this study is doubled since the test axle used contains only four bearings. Also, when one of the four bearings on the test axle is defective, the simulation considers that 25% of the wagon's bearings are defective.

## RESULTS

Speed Experiments:

Using control bearings (i.e., healthy bearings with no defects), several tests were carried out at 17% load (26 kN or 5.85 kips per bearing) simulating an empty railcar and train speeds of 48, 72, and 97 km/h (30, 45, and 60 mph). The motor power profiles for these tests are plotted in Figure 5. The figure clearly demonstrates that the power consumption increases with speed, which is to be expected. The average motor power values for the profiles displayed in Figure 5 are listed in Table 2 neglecting the initial two hours of run time associated with the tester start-up period.

Examining Figure 5 closely, it can be observed that the motor power, for all three speeds, approaches steady state after the initial two hours of operation. Note that the motor power for the 48 km/h (30 mph) speed exhibits the sharpest decrease in power consumption during the start-up two-hour period, which is due to the fact that this experiment was the first one to be run and the grease in the bearings was still fresh. The initial high-power consumption was needed to overcome the friction caused by the fresh new grease during the initial grease break-in period.

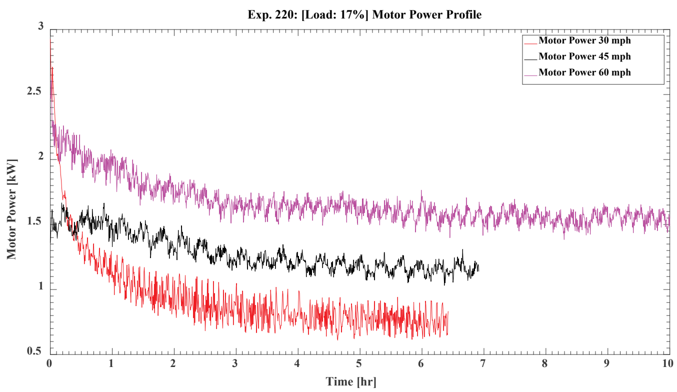


*Figure 5: Motor power profiles at 17% load and speeds of 48, 72, and 97 km/h (30, 45, 60 mph)*

For consistency, the motor power profiles were replotted showing only the operating period used to perform the analyses. Hence, Figure 6 contains the same information as that presented in Figure 5 but for the period from 2 to 6 hours of operation. The average motor power displayed in Table 2 was calculated for this four-hour period of operation.

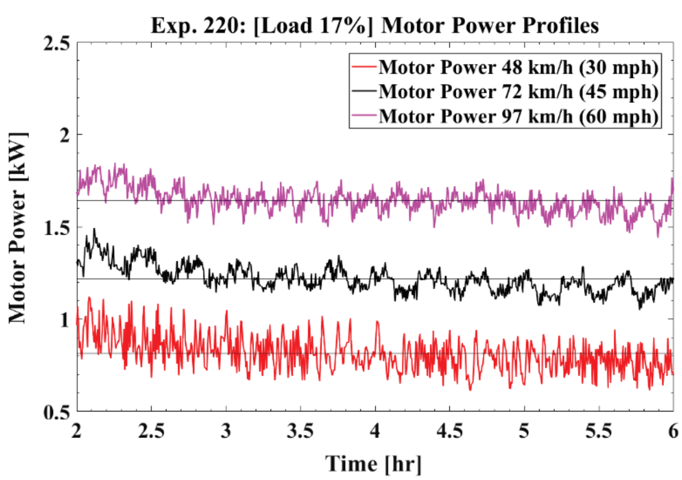


*Figure 6: Motor power profiles at 17% load and speeds of 48, 72, and 97 km/h (30, 45, 60 mph) showing period of interest*

Table 2 provides both the average motor power consumption in [kW] and the ton-mile per gallon which is used as a measure for efficiency of how much load and how many miles traveled theoretically with one gallon of diesel. This information is provided for speeds of 48, 72, and 97 km/h (30, 45, and 60 mph) at a load of 26 kN (5.85 kips). Table 2 shows how the average motor power increases with speed. Using the average motor power, the miles per gallon (MPG) were calculated from Eq. (2) and Eq. (3) for each of the three speeds. The ton-mile per gallon was then calculated from Eq. (4). The miles traveled at each speed over a six-hour period were used for the abovementioned calculations. Examining the MPG and ton-mile per gallon values listed in the table, there does not seem to be a noticeable difference at the three speeds for an empty railcar. One can argue that the 72 km/h (45 mph) speed is slightly more efficient for an empty railcar.

*Table 2: Experiment 220 results at 17% load and speeds of 48, 72, and 97 km/h (30, 45, 60 mph)*

| Exp. No. | Speed [km/h] / [mph] | Load | Average Motor Power [kW] | MPG | $\frac{Ton \cdot mile}{gallon}$ |
|---|---|---|---|---|---|
| 220 | 48 / 30 | 17% | 0.81 | 525 | 1,536 |
| | 72 / 45 | | 1.22 | 530 | 1,551 |
| | 97 / 60 | | 1.64 | 524 | 1,533 |

Table 3 lists the average operating temperatures above ambient for all four bearings on the test axle. The incremental change in the average operating temperatures above ambient between speeds is in the range of 10 to 12°C. As the speed increases, the operating temperatures of the test bearings also increase in response to the increase in motor power needed to rotate the bearings at the higher speeds.

*Table 3: Average operating temperature above ambient results for experiment 220 at 17% load. (Average ambient temperature was 20°C or 68°F)*

| Exp. No. | Load | RPM | Speed [km/h] / [mph] | Average Operating Temperature Above Ambient | | | |
|---|---|---|---|---|---|---|---|
| | | | | ΔT B1 [°C] | ΔT B2 [°C] | ΔT B3 [°C] | ΔT B4 [°C] |
| 220 | 17% | 280 | 48 / 30 | 22.0 | 20.4 | 21.4 | 19.5 |
| | | 420 | 72 / 45 | 31.2 | 30.5 | 32.5 | 28.5 |
| | | 560 | 97 / 60 | 43.4 | 43.0 | 42.7 | 41.4 |

Since the average operating temperatures of all four test bearings are relatively close to each other at all three speeds, one can assume that the average motor power consumption is equally distributed among all four test bearings. With this assumption, the average power consumption per bearing can be obtained by dividing the total power consumption given in Table 2 by four. Hence, the average power consumption per bearing at 17% load (empty railcar) is 0.20 kW, 0.31 kW, and 0.41 kW for speeds of 48, 72, and 97 km/h (30, 45, and 60 mph), respectively.

Load Experiments:

Using the same experimental setup with the four control bearings, the hydraulic cylinder of the test rig was set to apply 100% load (i.e., 153 kN or 34.4 kips per bearing). The motor power profiles for speeds of 48, 72, and 97 km/h (30, 45, and 60 mph) are displayed in Figure 7 with the average motor power consumption given in Table 4. As expected, the average motor power consumption increases with operating speed, and the values for a fully loaded railcar (100% load) are higher than the corresponding values for an empty railcar (17% load).

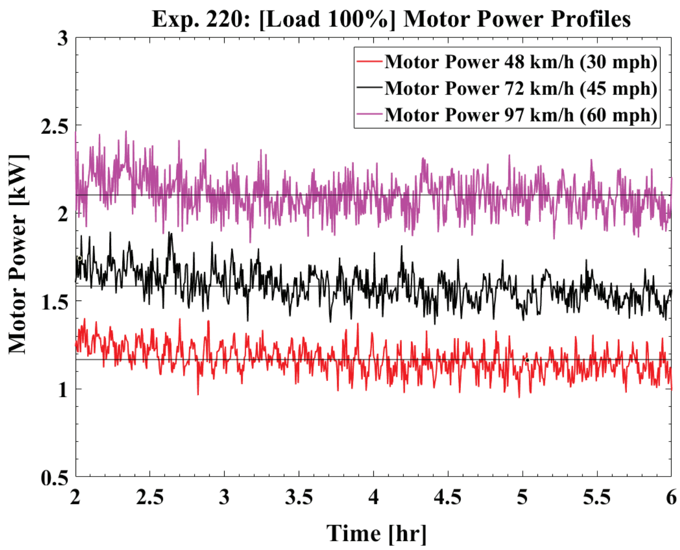


*Figure 7: Motor power profiles at 100% load and speeds of 48, 72, and 97 km/h (30, 45, 60 mph) showing period of interest*

*Table 4: Experiment 220 results at 100% load and speeds of 48, 72, and 97 km/h (30, 45, 60 mph)*

| Exp. No. | Speed [km/h] / [mph] | Load | Average Motor Power [kW] | MPG | $\frac{Ton \cdot mile}{gallon}$ |
|---|---|---|---|---|---|
| 220 | 48 / 30 | 100% | 1.17 | 366 | 6,302 |
| | 72 / 45 | | 1.58 | 407 | 7,003 |
| | 97 / 60 | | 2.10 | 410 | 7,047 |

Examining the results summarized in Table 4, one can immediately notice that the ton-mile per gallon values for a fully loaded railcar are more than four times those for an empty railcar at all three speeds investigated. Moreover, the MPG and ton-mile per gallon values for a fully loaded railcar indicate that there is a significant increase in efficiency going from a speed of 48 km/h (30 mph) to 72 km/h (45 mph), whereas, the difference in efficiency going from 72 km/h (45 mph) to 97 km/h (60 mph) is negligible. Hence, speeds in the range of 72 km/h to 97 km/h are optimal in terms of fuel efficiency for a fully loaded railcar with healthy (defect-free) bearings. In comparing the results of Table 2 to those of Table 4, it becomes apparent that the ton-mile per gallon value provides a better measure for fuel economy and efficiency than the MPG value. Even though the MPG values for a fully loaded railcar (100% load) are lower than the corresponding values for an empty railcar, the ton-mile per gallon values clearly demonstrate that a fully loaded railcar is more than four times as efficient as an empty railcar at all three speeds studied.

Table 5 lists the average operating temperatures above ambient for all four test bearings at a 100% load (full railcar) and speeds of 48, 72, and 97 km/h (30, 45, and 60 mph). As anticipated, the operating temperature increases as the speed increases and all average operating temperatures for a fully loaded railcar are noticeably higher than those for an empty railcar at all three speeds investigated. The increase in operating temperature is a direct result of the increase in the average motor power consumption.

*Table 5: Average operating temperature above ambient results for experiment 220 at 100% load.*
*(Average ambient temperature was 20°C or 68°F)*

| Exp. No. | Load | RPM | Speed [km/h] / [mph] | Average Operating Temperature Above Ambient | | | |
|---|---|---|---|---|---|---|---|
| | | | | ΔT B1 [°C] | ΔT B2 [°C] | ΔT B3 [°C] | ΔT B4 [°C] |
| 220 | 100% | 280 | 48 / 30 | 29.1 | 29.2 | 28.3 | 27.0 |
| | | 420 | 72 / 45 | 35.3 | 36.8 | 37.8 | 35.2 |
| | | 560 | 97 / 60 | 50.4 | 53.9 | 55.3 | 52.9 |

Again, since the average operating temperatures of all four test bearings are relatively close to each other at all three speeds, one can assume that the average motor power consumption is equally distributed among all four test bearings. Hence, the average power consumption per bearing at 100% load (full railcar) is 0.29 kW, 0.40 kW, and 0.53 kW for speeds of 48, 72, and 97 km/h (30, 45, and 60 mph), respectively.

Condition Experiments:

In order to explore the effects of defective bearings on fuel economy and efficiency, the bearing outer ring (cup) of bearing B2 was replaced with a defective cup that has two relatively large spalls pictured in Figure 8.

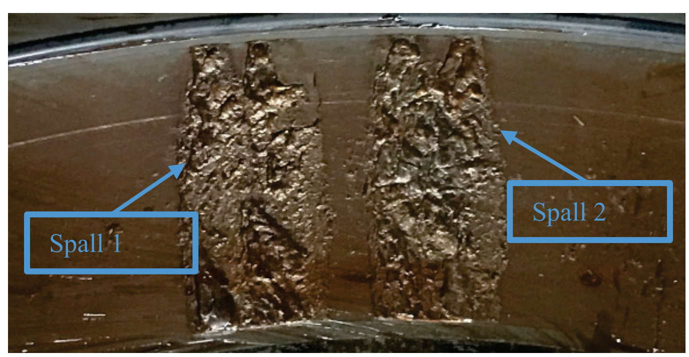


*Figure 8: Bearing 2 (B2) Cup spall*
*Spall 1: 10.2 $cm^2$ (1.575 $in^2$); Spall 2: 9.97 $cm^2$ (1.546 $in^2$)*

Like Experiment 220, Experiment 222 was run at 100% load and speeds of 48, 72, and 97 km/h (30, 45, and 60 mph). The motor power profiles at all three speeds are plotted in Figure 9, and the average motor power consumption, MPG, and ton-mile per gallon values are provided in Table 6.

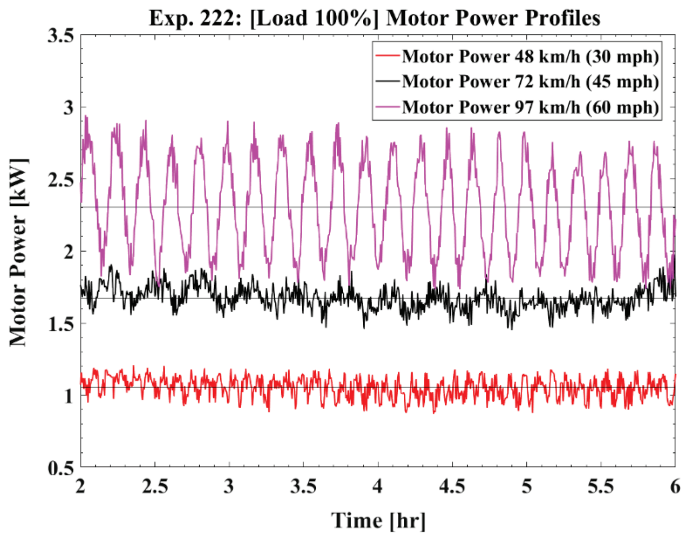


*Figure 9: Motor Power Profile at 100% load versus 30, 45, and 60 mph speeds*

*Table 6: Experiment 222 results at 100% load and speeds of 48, 72, and 97 km/h (30, 45, 60 mph)*

| Exp. No. | Speed [km/h] / [mph] | Load | Average Motor Power [kW] | MPG | $\frac{Ton \cdot mile}{gallon}$ |
|---|---|---|---|---|---|
| 222 | 48 / 30 | 100% | 1.05 | 407 | 6,992 |
| | 72 / 45 | | 1.67 | 386 | 6,631 |
| | 97 / 60 | | 2.30 | 375 | 6,446 |

Examining Figure 9, one can notice the sinusoidal behavior of the motor power at a speed of 97 km/h (60 mph). This behavior is also present at the lower speeds of 72 km/h and 48 km/h but to a much lesser extent. The reason for this noticeable sinusoidal motor power is the defective bearing B2 which contains two spalls. The motor power profile suggests that the two spalls on the bearing cup are causing the tapered rollers to misalign resulting in an abnormal operating condition that generates more friction and, thus, requires a larger motor power consumption to overcome the increased frictional forces. The subsequent decrease in motor power consumption is the result of the rollers re-aligning and returning to normal operating conditions, thus, frictional forces are reduced. The profile clearly demonstrates that the spalled bearing cup results in a noticeable cyclic motor power at full load and a speed of 97 km/h (60 mph) unlike the corresponding motor power profile of a healthy (defect-free) bearing at the same operating conditions. Moreover, results presented in Table 6 also support the abovementioned findings. Examining the average motor power, MPG, and ton-mile per gallon values listed in Table 6 and comparing them to the corresponding values for healthy bearings provided in Table 5, one can notice that the average motor power consumption increases for defective bearings at the higher speeds of 72 km/h and 97 km/h, whereas, the MPG and ton-mile per gallon values which quantify the fuel economy and efficiency decrease for defective bearings, as expected. Note that, at the lower speed of 48 km/h, the defective bearing does not negatively affect the fuel economy and efficiency. That is because the effects of the additional lubrication pockets that form in the spalled regions of the cup overcome the frictional forces at the lower speeds.

Now, comparing the average operating temperatures above ambient for Experiments 220 and 222 listed in Table 5 and Table 7, respectively, one can notice that, at the two higher speeds, the bearing operating temperatures for the setup that contains the defective bearing are slightly higher than those of the corresponding setup for all healthy bearings. This is in agreement with the average motor power consumption values for both setups. Note that, as previously identified, the bearing operating temperatures for the setup containing a defective bearing were not negatively affected at the lower speed of 48 km/h. In fact, the operating temperatures at this speed were slightly lower than those for the setup with all healthy bearings.

*Table 7: Average operating temperature above ambient results for experiment 222 at 100% load.*
*(Average ambient temperature was 20°C or 68°F)*

| Exp. No. | Load | RPM | Speed [km/h] / [mph] | Average Operating Temperature Above Ambient | | | |
|---|---|---|---|---|---|---|---|
| | | | | ΔT B1 [°C] | ΔT B2 [°C] | ΔT B3 [°C] | ΔT B4 [°C] |
| 222 | 100% | 280 | 48 / 30 | 26.7 | 27.5 | 28.8 | 25.5 |
| | | 420 | 72 / 45 | 37.4 | 38.7 | 41.3 | 38.8 |
| | | 560 | 97 / 60 | 52.3 | 53.3 | 58.5 | 53.1 |

Even though the setup for Experiment 222 contained one defective bearing, it seems like the average operating temperatures of all four bearings in the setup are relatively close to one another at all three speeds. Hence, we can assume, within a reasonable approximation, that the motor power consumption is equally divided among all four bearings. Consequently, the average power consumption per bearing at 100% load (full railcar) for a setup containing one defective bearing is 0.26 kW, 0.42 kW, and 0.58 kW for speeds of 48, 72, and 97 km/h (30, 45, and 60 mph), respectively.

Economic and Environmental Impact:

The significance of bearing power consumption may be dismissed when looking at the relatively small experimental results from setups that only contain four bearings. In order to quantify the fuel economy and efficiency resulting from the incremental changes in bearing power consumption, a simulation was proposed for a train consist of 59 wagons hauled by one locomotive. Table 8 summarizes the main results obtained for this simulation. Data from Table 4 and Table 6 were used to acquire the results provided in Table 8. In the table, the average motor power for the simulation was obtained by multiplying the values listed in Table 4 and Table 6 by two to get the total power consumption per wagon and then by 59 wagons to get the total power consumption for the entire train consist. To calculate the total tons hauled by this train consist, the full load of 143 tons (obtained from reference [1]) for one railcar was multiplied by 59 wagons.

*Table 8: Power consumption and energy efficiency of a simulated train consist of 59 wagons hauled by one locomotive.*

| Simulation Results: 59 Wagons Hauled by One Locomotive | | | | | |
|---|---|---|---|---|---|
| Exp. No. | Speed [km/h] / [mph] | Load | Average Motor Power [kW] | MPG | $\frac{Ton * mile}{gallons}$ |
| 220 | 48 / 30 | 100% | 138 | 3.11 | 26,201 |
| | 72 / 45 | | 187 | 3.45 | 29,112 |
| | 97 / 60 | | 248 | 3.48 | 29,344 |
| 222 | 48 / 30 | 100% | 124 | 3.44 | 29,065 |
| | 72 / 45 | | 197 | 3.27 | 27,563 |
| | 97 / 60 | | 272 | 3.18 | 26,794 |

Studying the results of Table 8, one can conclude that the optimal operating conditions for a train consist of 59 wagons and one locomotive with all healthy bearings are full load running at speeds ranging from 72 km/h (45 mph) to 97 km/h (60 mph). In order to maintain a similar fuel economy and efficiency for the abovementioned train consist with 25% of its bearings having cup spalls similar to those pictured in Figure 8, the optimal traveling speed should be decreased to around 48 km/h (30 mph). Traveling at a speed of 97 km/h (60 mph), the train consist with 25% defective bearings will have its fuel economy and efficiency reduced by about 9% as compared to the corresponding train consist with all healthy bearings traveling at the same speed.

## CONCLUSION

There is an urgent need to identify, evaluate, and implement technologies and/or operating practices to maintain railroad competitiveness. This paper focused on the energy consumption of a specific component; i.e., the railroad tapered roller bearing. The bearing power consumption was determined as a function of load, speed, and bearing condition.

The results summarized here demonstrate that the ton-mile per gallon is a better measure for fuel economy and efficiency than the corresponding miles per gallon (MPG). This becomes apparent when comparing these values for an empty railcar (Table 2) versus a fully loaded railcar (Table 4 and Table 6). Interesting to note is that the motor power consumption does not directly correlate to the fuel efficiency of the train. The ton-mile per gallon provides the optimal measure of fuel efficiency because it incorporates both the MPG and the total cargo load being hauled.

The results of the study also conclude that defective bearings significantly affect the fuel economy and efficiency, especially at the higher speeds ($\geq$ 72 km/h or 45 mph). The simulation results, in which a train consist of 59 wagons hauled by one locomotive is analyzed, also support the aforementioned finding. The results listed in Table 8 compare the fuel economy and efficiency of the train consist with all healthy bearings versus the same train consist having 25% of its bearings defective. A direct comparison reveals that the defective bearings were responsible for a 9% reduction in fuel efficiency at a train speed of 97 km/h (60 mph). To quantify this reduction, consider a 10,000-mile trip hauling 59 fully loaded wagons at 97 km/h (60 mph), the train consist with the 25% faulty bearings would require 271 gallons of diesel more than the same train consist with all healthy bearings.

This study summarizes preliminary work conducted to demonstrate how the performance of railroad tapered-roller bearings, which are part of the railcar suspension system, can affect the fuel economy and efficiency of a train under normal and abnormal operating conditions. Note that the effects of drag were not considered in the analyses presented here. Nevertheless, the results provide the reader with a basic understanding of how incremental changes in bearing power consumption affect the overall fuel economy and efficiency.

## FUTURE WORK

Moving forward, new variables will be considered for further analysis such as different bearing conditions, lubricants, and ambient conditions along with the inclusion of drag force at the different velocities.

## ACKNOWLEDGMENTS

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