Agricultural tire dynamometer test stand

Colton Finley
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Agricultural tire dynamometer test stand

by

Colton D. Finley

A thesis submitted to the graduate faculty
in partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE

Major: Agricultural and Biosystems Engineering

Program of Study Committee:
Stuart Birrell, Major Professor
Amy Kaleita
Brian Steward

The student author, whose presentation of the scholarship herein was approved by the program of study committee, is solely responsible for the content of this thesis. The Graduate College will ensure this thesis is globally accessible and will not permit alterations after a degree is conferred.

Iowa State University

Ames, Iowa

2018

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ABSTRACT

The agricultural world is developing tires with new capabilities that improve heavy machinery’s fuel efficiency, tractive force, and ride comfort. To minimize cost, a manufacturer could use the Brixius or Wismer-Luth soil - traction dynamic model to predict how a vehicle’s tire would perform in the field. However, these models have two major drawbacks, inconsistent soil cone index readings and inaccurate tire dimension representations. Since empirical equations are used to generate a tire’s tractive capability characteristics, only the tire evaluated and the field conditions present during testing are represented well by the model. This increases error when comparing different tires in different conditions.

To provide a model that provides a reliable evaluation of tractive performance, an agricultural tire dynamometer was proposed. The proposed dynamometer eliminated the cone index altogether so that engineers could see what relationship a wheel’s dimensions have on its tractive performance independent of cone index readings. To eliminate the cone index, the test stand will utilize a set of belts to study the interaction between the tire and a well-defined surface. The test stand is able to control the torque, down pressure, contact area, and velocity of the agricultural tire in question, while providing accurate and reliable data to analyze.

This thesis contains a literature review of the soil traction dynamic models. Then, a detailed description of how the parts were sized and selected for the dynamometer. Finally, the characteristics and outputs of the test stand were modeled. The system was simulated to investigate its performance.

The dynamometers capabilities were discussed along with future work that could provide improvements. Future work could include the system control of the contact area between the tire and belts. Also, a method of quickly changing the belts if needed.
CHAPTER 1. LITERATURE REVIEW: SOIL - TRACTION DYNAMIC MODELS

An accurate understanding of how force is transmitted into the ground and the pull generated from that force is especially valuable information in the agricultural machinery world. With the technology and information systems implemented in modern heavy machinery, accurate and reliable models are critical to understanding soil - traction dynamics. The Brixius and Wismer-Luth soil - traction dynamic models have been used for decades. They are used to predict the tractive efficiency that a wheel can provide. Both of these soil - traction dynamic models can be unreliable when conditions of a soil are significantly different from the conditions present when field-testing was completed for the development of the models. Since there are so many in-field factors involved in a soil’s condition, because it changes so frequently, accurate trials are hard to repeat.

The Wismer-Luth and Brixius models are both based on the soil cone index. The cone index (CI) is a commonly used soil mechanical parameter used to assess soil strength (ASAE, 2013). A major problem with cone index readings is that they will change due to the soil’s moisture content and other conditions in a matter of hours. Some factors that influence a cone index reading could be the time of day, weather conditions, location, and traffic frequency. Since the cone index frequently changes, collecting a sufficient amount of data with a constant soil condition is difficult. When manufacturers want to compare tire sizes or brands to choose a tire that best fits their machine the error created from these inconsistencies pose a problem. Thus, there is always a possibility that the reason one tire outperformed the other is because of the soil condition variability and not because of the tire’s superior capabilities.
The Wismer-Luth and Brixius models are used to predict the tractive capabilities of a wheel using the cone index, dimensions of the tire, and wheel load. Both of these models were developed empirically. This empirical method involved taking field research data and forming a “best fit” equation to predict the output. This process led to these equations predicting tire capabilities reliably only if conditions present are similar to the conditions present during the development of the models themselves.

When comparing tires for consideration a reliable and repeatable model is needed so that the tires are judged solely on their performance and not by the soil conditions present at the time of testing. It would be valuable if manufacturers were able to use a traction model that was not dependent on the field conditions present during the development of the model. This could be achieved by eliminating the greatest source of error, the soil itself. Development of a model like this could lead to a better understanding of the traction dynamics of large agricultural tires. It could also lead to more accurate and reliable soil-traction dynamic models in the future.

Although considerable progress has been made in the past few decades, our understanding of the soil-traction device interaction and our capability to quantify these interactions are far from satisfactory. Numerous attempts have been made to quantify soil-traction device interaction (Upadhyaya, 2009).

Research in this area has been going on since the end of WWII when the U.S. Army Corps of Engineers Waterways Experiment Station (WES) wanted to find some way for a soldier to assess the soil and know if military equipment could traverse it successfully. They developed the vehicle cone index (VCI_N) as a way to determine if travel was possible (Upadhyaya, 2009).
The VCI\textsubscript{N} is the minimal value of the rating cone index (RCI) at which a vehicle could successfully complete N passes in the same rut (Lutz, 2009). This VCI\textsubscript{N} is found experimentally by measuring the RCI for each of the N passes tested, the RCI value after all passes equals the VCI\textsubscript{N}. Taking the cone index (CI) and multiplying it by the remold index (RI) determines the RCI (Equation 1, Stevens et al., 2013). The cone and remold indexes are measured with a variety of soil equipment, including a cone penetrometer, Hvorslev sampler, drop hammer, and a cylindrical tube mounted to a base plate (Figure 1). The remold index (RI) is a measure of the sensitivity of soil to strength losses under vehicular traffic (Stevens et al., 2013). A table was created relating the determined VCI\textsubscript{N}’s of different military vehicle classes to predicted traffic frequency (Table 1).

When heavy traffic is present soil compacts and deforms, which leads to vehicles experiencing different soil conditions over time. In a go-no-go situation, the VCI performed well but it was not without its shortcomings. For cohesive soils, for example clay, the index related well to the soil shear strength. When the index predicted the shear strength for frictional soils, for example sand, it did not match up well with the experimental results (Reece & Peca, 1981). A need for a model that could predict consistently for a broad array of soil types was apparent. This vehicle cone index was just the start of over 50 years of research still going on today.
The agricultural world adopted the cone index (CI) as the basis for the research and development of more efficient off-road locomotion. Wismer and Luth (1974) developed one of the most widely used traction models for soils that are not highly compactable. The Wismer-Luth model consists of a series of equations.

\[
RCI = CI \times RI
\]

where

\[
RCI = Rating\ Cone\ Index\ (lb/\ in^2)
\]

\[
CI = Cone\ Index\ (lb/\ in^2)
\]

\[
RI = Remold\ Index\ (unitless)
\]
Figure 2. Pull-torque-slip relation, wheels on soil (Wismer & Luth, 1974)

Figure 3. Tire Cross-section Dimensions (Herigemblong, 2013)
The VCI\textsubscript{N} modeled a vehicle’s interaction with soil. Whereas, in the agricultural world the interaction between a vehicle’s wheel and soil is of more interest. To understand the soil - traction interaction of a wheel better, the Wismer-Luth wheel numeric consisting of the vertical load present on the wheel, the cone index, and critical dimensions of the tire was created (Equation 2). Figure 3 shows a cross sectional view of a tire with dimensions labeled.

Figure 2 shows the free body diagram of a towed wheel. A towed wheel is present, when motion resistance is assumed equal to towed force. The towed force (TF) can be thought of as the force needed to pull a non-powered wheel over the ground, axle torque is zero neglecting bearing friction (Wismer & Luth, 1974). Equation 3 shows the empirically developed Wismer-Luth motion resistance ratio.

\[
C_n = \frac{Clbd}{W} \tag{2}
\]

where

- \(C_n\) = Wheel Numeric (unitless)
- \(b\) = Tire Section Width (in)
- \(d\) = Overall Tire Diameter (in)
- \(W\) = Wheel Load (lbf)
- \(CI\) = stated in previous figures

\[
\rho = \frac{TF}{W} = \frac{1.2}{C_n} + 0.04 \tag{3}
\]

where

- \(\rho\) = Motion Resistance Ratio (unitless)
- \(TF\) = Towed Force (lbf)
- \(W, C_n\) = stated in previous figures
Empirical analysis was conducted on a driving wheel as shown in Figure 2. A net tractive force or wheel pull (P), as used in Equation 4, was equated to gross tractive force (F) provided by the vehicle minus the tire’s towed force (TF). The Wism-Luth net tractive force coefficient (Equation 4) found by experimentation along with the motion resistance ratio led to the finding of the gross tractive force coefficient (Equation 5).

\[
\mu = \frac{P}{W} = \frac{F - TF}{W} = 0.75(1 - e^{-0.3C_nS}) - \left(\frac{1.2}{C_n} + 0.04\right)
\]

where

- \(\mu\) = Net Tractive Force Coefficient (unitless)
- \(P\) = Net Tractive Force (lb)
- \(F\) = Gross Tractive Force (lb)
- \(S\) = Wheel Slip Percentage (in decimal format)
- \(TF, W, C_n\) = stated in previous figures

However, during the development of the model, several assumptions were made. For example, the coefficient 1.2 was used to assume that the height of an agricultural tire at normal air pressure deflects by 20% compared to a tire with no load (Equation 3). In reality, different tires deflect differently under load. The coefficient 0.3 assumed that the tire has a section width to diameter ratio (b/d) of 0.3 (Equations 4 & 5). The Wism-Luth model was specific to certain tire dimensions because of these assumptions (Upadhyaya, 2009). Any large deviation away from

\[
\mu_g = \frac{F}{W} = 0.75(1 - e^{-0.3C_nS})
\]

where

- \(\mu_g\) = Gross Tractive Force Coefficient (unitless)
- \(F, W, C_n, S\) = stated in previous figures
these assumed parameters affect the prediction accuracy. Modern agricultural tires come in many different sizes making the model inaccurate when comparing tires that do not satisfy the assumed parameters.

The net and gross tractive force coefficient equations contain wheel slip (S) also known as travel reduction, which is a very simple calculation. It is the relation of the actual velocity over the theoretical velocity (Equation 6).

It could be assumed that, for a manufacturer’s purpose, the smaller the towed force or motion resistance ratio and greater the gross tractive force coefficient the better. Meaning more engine power would be transmitted into the ground for the pulling of implements and not the moving of the wheel itself. Essentially, this means that the greater the gross tractive force coefficient is compared the motion resistance ratio, the more efficient the wheel in question is at transmitting an off-road vehicle’s power to the ground.

The Wismer-Luth model has used throughout the large off-road vehicle industry to predict soil traction dynamics on a vast array of soil types. However, the model is really only valid when the conditions present are similar to the conditions present when the empirical Wismer-Luth model’s experimental data was being recorded. It is especially inaccurate in highly compactible soil, like a freshly tilled plot (Upadhyaya, 2009). A major contributor to the model’s error was due to the cone

\[
S = 1 - \frac{V_a}{V_t}
\]  

(6)

where

- \( S \) = Wheel Slip Percentage (in decimal format)
- \( V_a \) = Actual Travel Velocity (mph)
- \( V_t \) = Theoretical Travel Velocity (mph)
index readings variability. Moisture present, traffic frequency, and time of day are some of the many factors that affect the wheel numeric via the cone index reading.

An accurate and repeatable traction dynamic model is critical for manufacturers wanting to compare tires to see what best meets their needs. Along with the use of an inconsistent cone index and being restricted to specific tire dimensions because of assumptions previously described, using the Wismer-Luth model to compare tractive performance capabilities of different varieties of tires may not be suitable.

In a clay soil, the Wismer-Luth model tends to under predict the draft and torque obtained with the more common radial-ply tires. It should be noted that the Wismer-Luth equations were developed based on bias-ply tires (Upadhyaya, 2009). Since radial-ply tires deform differently under a load compared to bias-ply tires it should be expected that the Wismer-Luth model would not be able to predict the motion resistance ratio accurately for radial-ply tires. Thus, a way to quantify the deformations of these tires was needed.

Zoz and Brixius (1979) developed a new empirical model that added a deformation input as an attempt to account for the shortcomings of the Wismer-Luth model at representing different tire dimensions (Equations 7 & 8, Upadhyaya, 2009). Deere and Co. developed this model in collaboration with the Nebraska Tractor Test Laboratory. When using the Zoz and Brixius model, the coefficient k was originally found to be constant. Later on, it was found that its value was actually dependent on the tire’s contact surface (Equation 7) (Leviticus and Reyes 1983, 1985). Later, tests were conducted to validate the model but performed poorly for U.K., U.S., German, and French tire tests on concrete (Dwyer, 1987). Thus, these two studies suggest that this model’s results were hard to validate.
A more generalized model for the soil–traction dynamic of radial-ply pneumatic tires was later developed to expand the capabilities of the Zoz and Brixius model (Brixius, 1987). The Brixius model was based on a modified mobility number referred to as $B_n$, as explained in (Upadhyaya, 2009). This $B_n$ combined the wheel numeric, $C_n$, with the deflection ratio, and width-to-diameter ratio of the tire being considered (Equation 9, Elwaleed et al., 2004). Using the empirical method, the data recorded was used to equate an updated motion resistance ratio (Equation 10) and net tractive force coefficient (Equation 11) resulting in a gross tractive force coefficient (Equation 12). The empirical data was collected in a similar fashion as the Wismer-Luth model (Equations 2-5).

The Wismer-Luth and Brixius soil traction dynamic models both focused on finding the best way to represent the motion resistance ratio, where correct tire dimension representation and accurate soil condition readings were critical to minimizing error in tractive performance capability predictions. To eliminate the size of a vehicle form causing more variance in data

\[
\mu = \frac{P}{W} = \frac{F - TF}{W} = 1.02\left[1 - e^{-k(b/d)S}\right] \tag{7}
\]

where

- $\mu$ = Net Tractive Force Coefficient (unitless)
- $k$ = Coefficient (unitless)
- $P, W, F, TF, b, d, S$ = stated in previous figures

\[
\mu_g = \frac{F}{W} = \frac{P}{W} + 0.02 \tag{8}
\]

where

- $\mu_g$ = Gross Tractive Force Coefficient (unitless)
- $F, W, P$ = stated in previous figures
collected, the towed force (TF) and gross force (F) were normalized. To normalize the towed force (TF) and gross force (F) over a range of vehicles they are each divided by the axle weight experienced by the wheel. Normalizing these forces over a range of axle weights makes the evaluations of tire performance independent of vehicle size and weight. With the net tractive and

\[ B_n = \left( \frac{CI \times b \times d}{W} \right) \left( \frac{1 + 5 \frac{\delta}{h}}{1 + 3 \frac{b}{d}} \right) \]  

(9)

where

- \( B_n \) = Brixius Mobility Number (unitless)
- \( \delta \) = Tire Deflection (in)
- \( h \) = Tire Section Height (in)
- \( CI, b, d, W \) = stated in previous figures

\[ \rho = \frac{TF}{W} = \frac{1.0}{B_n} + 0.04 + \frac{0.5 \times S}{\sqrt{B_n}} \]  

(10)

where

- \( \rho \) = Motion Resistance Ratio (unitless)
- \( TF, W, B_n, S \) = stated in previous figures

\[ \mu = \frac{P}{W} = \frac{F - TF}{W} = [0.88(1 - e^{-0.1B_n})(1 - e^{-7.5 \times S}) + 0.04] - \left[ \frac{1.0}{B_n} + 0.04 + \frac{0.5 \times S}{\sqrt{B_n}} \right] \]  

(11)

where

- \( \mu \) = Net Tractive Force Coefficient (unitless)
- \( P, W, F, TF, B_n, S \) = stated in previous figures
gross tractive coefficients normalized, quantifying and comparing tire tractive performances independent of vehicle weight was possible. Equation 13 shows the tractive efficiency equation. This equation takes the net tractive force times the actual linear velocity of the vehicle, divided by the gross tractive force time the rolling radius and angular velocity of the axle. Essentially, the drawbar power over the axle power. In other words, a numerical representation of how well a tire converts the vehicle’s tractive force into a pulling force through drawbar.

The methods of finding the net tractive force coefficient and motion resistance ratio are straightforward and have not changed for decades. Measuring the amount of force needed to pull a towed wheel resulted in the towed force and thus a motion resistance ratio. Whereas, measuring the amount of force produced from a driving wheel resulted in the net tractive force coefficient. Then, by combining the motion resistance ratio equation with the net tractive force coefficient equation, a gross tractive force coefficient equation is determined. How the dimensions of a tire are represented in the motion resistance ratio critical in the development of a reliable net tractive force coefficient and thus a gross tractive force coefficient. To be reliable the two tractive force coefficients must accurately encompass all tires and not just specific tires with specific conditions present during development of the model in question.

An accurate motion resistance ratio is crucial to understanding the tractive performance capabilities of large agricultural tires. Tests were conducted at the University Putra Malaysia to test the validity of the Wismer-Luth and Brixius models. This study calculated the actual motion efficiency equation:

\[
\mu_g = \frac{F}{W} = 0.88(1 - e^{-0.1B_n})(1 - e^{-7.5*S}) + 0.04
\]

where

\[
\mu_g = \text{Gross Tractive Force Coefficient (unitless)}
\]

\[
F, W, B_n, S = \text{stated in previous figures}
\]
The tests were conducted with a custom test stand that consisted of a moving trolley over a 6.4 m (21 ft) long soil bin. The trolley consisted of a mounted wheel that was pulled by an electric motor with a drive chain. The wheel was non-powered and in towed mode (Figure 4). The study evaluated the effect of tire’s air pressure on the motion resistance ratio, and reported that varying air pressures changed the dimensions of the tire, specifically the width to diameter ratio and deflection.

The soil bin was compacted and prepared before each run to cover the wheel numeric \( C_n \) range of 5-70. Thirty runs were prepared corresponding to standards defined by the ASAE Standard D497.4 (Table 2, ASAE, 2000). These runs were conducted for three levels of tire inflation (166, 193, and 221 kPa or 24.1, 28.0, and 32.1 psi respectively). These tire pressures

\[
TE = \frac{P_{db}}{P_a} = \left( \frac{\mu_g - \rho}{\mu_g} \right) * (1 - S) = \frac{F_{db} * V_a}{T_a * \omega_a} = \frac{P * V_a}{F * r * \omega_a}
\]

where

- \( TE \) = Tractive Efficiency (unitless)
- \( \mu_g \) = Gross Tractive Force Coefficient (unitless)
- \( \rho \) = Motion Resistance Ratio (unitless)
- \( S \) = Wheel Slip Percentage (in decimal format)
- \( F_{db} \) = Drawbar Force (lb)
- \( T_a \) = Axle Torque (ft-lb)
- \( \omega_a \) = Axle Angular Velocity (rad/s)
- \( r \) = Rolling Radius (ft)
- \( V_a, P, F \) = stated in previous figures

resistance ratio of agricultural tires to compare with these model’s predictions (Elwaleed et al., 2004).
defined specific tire dimensions for a Bridgestone 5-12, 4-ply, lug-M garden tractor tire described in (Table 3). Then the actual motion resistance ratio was calculated for each trial run.

An analysis of covariance (ANCOVA) was carried out to determine the level of significance that the effect of tire pressure and Wismer-Luth wheel numeric \( C_n \) have on the motion resistance ratio (Table 4). Both the tire pressure and wheel numeric were found to have a highly significant effect (alpha = 0.01), on the mean of the motion resistance ratio’s trial runs (Elwaleed et al., 2004); strongly suggesting that tire pressure and wheel numeric affect the motion resistance ratio. A 0.01 significance level means that there is a 1.0% probability that the ANCOVA test results were incorrect. The mean of the motion resistance ratio’s trials (dependent variable) was tested across the range of wheel numerics (covariate) while including the variation of the wheel numeric into the variation of the tire pressure (independent variable) to minimalize the error variance. When the

**Table 2. Classification of Typical Agricultural Soils**
(Elwaleed et al., 2004)

<table>
<thead>
<tr>
<th>Soil Classification</th>
<th>( C_n )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hard soil</td>
<td>50</td>
</tr>
<tr>
<td>Firm soil</td>
<td>30</td>
</tr>
<tr>
<td>Tilled soil</td>
<td>20</td>
</tr>
<tr>
<td>Soft soil or sandy</td>
<td>15</td>
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</tbody>
</table>
variance is partitioned like this, an adjusted mean is created. The adjusted means of the ANCOVA test are found in Table 5. These adjusted means represent a “best fit regression” of the data with respect to the variance of the wheel numeric and tire pressure.

From the adjusted means in Table 5, when the tire pressure was reduced from 221 kPa (32.1 psi) to 193 kPa (28.0 psi) there was a 5.01% mean reduction in motion resistance ratio (Figure 5). Conversely, when inflation pressure was reduced from 221 KPa (32.1 psi) to 166 kPa (24.1 psi) the mean motion resistance ratio increased by 9.96% (Elwaleed et al., 2004). Since decreasing tire pressure does not necessarily lead to a smaller motion resistance ratio, there is an optimum tire pressure and tire dimensions that correspond to the minimum motion resistance ratio.

ANCOVA was also performed using the Brixius model’s mobility number ($B_n$) (Table 6). Both the tire pressure and mobility number ($B_n$) were found to have a highly significant effect (alpha = 0.01), on the mean of the motion resistance ratio’s trials (Elwaleed et al, 2004). Thus, there was evidence that the tire pressure and mobility number affect the motion resistance ratio.

Table 3. Bridgestone Tractor Tire Dimensional Characteristics (Elwaleed et al., 2004)

<table>
<thead>
<tr>
<th>Tire Pressure kPa (psi)</th>
<th>Overall Diameter $d$ cm (in)</th>
<th>Rolling Radius $r$ cm (in)</th>
<th>Section Width $b$ cm (in)</th>
<th>Section Height $h$ cm (in)</th>
<th>Loaded Deflection $\delta$ cm (in)</th>
<th>b/d ratio</th>
<th>$\delta/h$ ratio</th>
<th>r/d ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>166 (24.1)</td>
<td>54.96 (2.164)</td>
<td>26.74 (10.53)</td>
<td>12.40 (4.88)</td>
<td>12.24 (4.82)</td>
<td>1.21 (0.48)</td>
<td>0.226</td>
<td>0.099</td>
<td>0.487</td>
</tr>
<tr>
<td>193 (28.0)</td>
<td>55.01 (2.166)</td>
<td>26.81 (10.56)</td>
<td>12.40 (4.88)</td>
<td>12.27 (4.83)</td>
<td>1.14 (0.45)</td>
<td>0.225</td>
<td>0.093</td>
<td>0.487</td>
</tr>
<tr>
<td>221 (32.1)</td>
<td>55.06 (2.168)</td>
<td>26.87 (10.58)</td>
<td>12.40 (4.88)</td>
<td>12.29 (4.84)</td>
<td>1.08 (0.43)</td>
<td>0.225</td>
<td>0.088</td>
<td>0.488</td>
</tr>
</tbody>
</table>

Table 4. ANCOVA for Tire Motion Resistance Ratio in terms of the Wismer-Luth Wheel Numeric (Elwaleed et al., 2004)

<table>
<thead>
<tr>
<th>Source</th>
<th>DoF</th>
<th>SS</th>
<th>MS</th>
<th>F ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tire Pressure</td>
<td>2</td>
<td>0.0101</td>
<td>0.005</td>
<td>9.23**</td>
</tr>
<tr>
<td>$C_n$</td>
<td>1</td>
<td>0.2113</td>
<td>0.2113</td>
<td>387.06**</td>
</tr>
<tr>
<td>Error</td>
<td>86</td>
<td>0.0469</td>
<td>0.0005</td>
<td></td>
</tr>
<tr>
<td>Total</td>
<td>89</td>
<td>0.2683</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

** Significant at 0.01 probability level
From the adjusted means in Table 7, when the tire pressure was reduced from 221 kPa (32.1 psi) to 193 kPa (28.0 psi), there was a 4.05% mean reduction in motion resistance ratio. Conversely, when inflation pressure was reduced from 221 KPa (32.1 psi) to 166 kPa (24.1 psi) the mean motion resistance ratio increased by 12.09% (Elwaleed et al., 2004). This reiterates that there is an optimum tire pressure that leads to a minimum motion resistance ratio.

In this study, regression analysis was performed on the individual tire pressures to find an equation that best fit the recorded data (Figure 5). The motion resistance ratio increased exponentially as the wheel numeric decreased or as the soil strength decreased. There was a notable advantage to the motion resistance ratio when decreasing tire pressure from 221 kPa (32.1 psi) to 193 kPa (28.0 psi). The motion resistance ratio deteriorated as the tire pressure continued to decrease. It can then be assumed that an optimum tire pressure exists for a particular tire on a particular surface.

**Table 5. Mean and Adjusted Mean of Motion Resistance Ratio in terms of the Wismer-Luth Wheel Numeric (Elwaleed et al., 2004)**

<table>
<thead>
<tr>
<th>Tire Pressure, kPa (psi)</th>
<th>Motion resistance ratio, mean</th>
<th>Motion resistance ratio, adjusted mean</th>
</tr>
</thead>
<tbody>
<tr>
<td>166 (24.1)</td>
<td>0.1664</td>
<td>0.1645</td>
</tr>
<tr>
<td>193 (28.0)</td>
<td>0.1411</td>
<td>0.1421</td>
</tr>
<tr>
<td>221 (32.1)</td>
<td>0.1486</td>
<td>0.1496</td>
</tr>
</tbody>
</table>

From the adjusted means in Table 7, when the tire pressure was reduced from 221 kPa (32.1 psi) to 193 kPa (28.0 psi), there was a 4.05% mean reduction in motion resistance ratio. Conversely, when inflation pressure was reduced from 221 KPa (32.1 psi) to 166 kPa (24.1 psi) the mean motion resistance ratio increased by 12.09% (Elwaleed et al., 2004). This reiterates that there is an optimum tire pressure that leads to a minimum motion resistance ratio.

In this study, regression analysis was performed on the individual tire pressures to find an equation that best fit the recorded data (Figure 5). The motion resistance ratio increased exponentially as the wheel numeric decreased or as the soil strength decreased. There was a notable advantage to the motion resistance ratio when decreasing tire pressure from 221 kPa (32.1 psi) to 193 kPa (28.0 psi). The motion resistance ratio deteriorated as the tire pressure continued to decrease. It can then be assumed that an optimum tire pressure exists for a particular tire on a particular surface.

**Table 6. ANCOVA for Tire Motion Resistance Ratio in terms of the Brixius Mobility Number (Elwaleed et al., 2004)**

<table>
<thead>
<tr>
<th>Source</th>
<th>DoF</th>
<th>SS</th>
<th>MS</th>
<th>F ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tire Pressure</td>
<td>2</td>
<td>0.0101</td>
<td>0.005</td>
<td>9.28**</td>
</tr>
<tr>
<td>Bn</td>
<td>1</td>
<td>0.2115</td>
<td>0.2115</td>
<td>389.31**</td>
</tr>
<tr>
<td>Error</td>
<td>86</td>
<td>0.0467</td>
<td>0.0005</td>
<td></td>
</tr>
<tr>
<td>Total</td>
<td>89</td>
<td>0.2683</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

** Significant at 0.01 probability level
A logarithmic model was found to have the highest $R^2$ value or best fit of the actual data recorded during this study. This model was used as a benchmark to compare with the Wismer-Luth and Brixius models (Figure 6). The Wismer-Luth and Brixius models under predicted the motion resistance ratio when the wheel numeric was greater than 5 (Elwaleed et al., 2004).

There are significant differences between the predicted motion resistance ratio of the Wismer-Luth and Brixius models compared to the logarithmic model (Table 8). With a tire pressure of 193

<table>
<thead>
<tr>
<th>Tire Pressure, kPa (psi)</th>
<th>Motion resistance ratio, mean</th>
<th>Motion Resistance ratio, adjusted mean</th>
</tr>
</thead>
<tbody>
<tr>
<td>166 (24.1)</td>
<td>0.1664</td>
<td>0.1645</td>
</tr>
<tr>
<td>193 (28.0)</td>
<td>0.1411</td>
<td>0.1421</td>
</tr>
<tr>
<td>221 (32.1)</td>
<td>0.1486</td>
<td>0.1496</td>
</tr>
</tbody>
</table>

**Table 7. Mean and Adjusted Mean of Motion Resistance Ratio in terms of the Brixius Mobility Number (Elwaleed et al., 2004)**

![Figure 5. Motion Resistance Ratio at Different Tire Pressure Levels (Elwaleed et al., 2004)
kPa (28.0 psi), the predicted tire motion resistance ratios were 36.8%, 41.2%, 42.9%, and 45.5% higher than the predicted values made by the Wismer-Luth model and were 26.3%, 29.4%, 35.7%, and 36.4% higher than the predicted values made by the Brixius model for soft, tilled, firm, and hard soil, respectively (Elwaleed et al., 2004).

From the University Putra Malaysia’s study, it can be concluded that the Wismer-Luth and Brixius soil–traction dynamic models do not predict the motion resistance ratio of all agricultural tires accurately. This leads to more variance when trying to compare tractive performance capabilities.

The greatest problem faced with the development of a soil-traction dynamic model that encompasses a wide array of soil surfaces and tire types is finding an accurate and reliable way to

![Figure 6. Regression Lines of Motion Resistance Ratios for Logarithmic, Wismer-Luth, and Brixius Models (Elwaleed et al., 2004)]
predict the motion resistance ratio. Soil shear strength and tire dimensions are major contributors in finding a tire’s motion resistance ratio and tractive force coefficients. The more recent Brixius model was an attempt to overcome the Wismer-Luth model’s tire dimension representation shortcomings. However, according to the University Putra Malaysia’s study, both soil-tractraction dynamic models still do not meet an acceptable standard when manufacturers are trying to compare tires. It should be noted that the University Putra Malaysia’s study utilized a Bridgestone 5-12 agricultural tire. The size of tire that would be common on a small garden tractor. Whereas, the Wismer-Luth and Brixius equations were developed with the idea of predicting tractive performance for significantly larger agricultural tires.

A soil’s cone index is a large source of error in both the Wismer-Luth and Brixius equations. Since soil is not a homogenous material and its condition changes so frequently, field test results can vary wildly. Both the Wismer-Luth and Brixius models struggle with relating tire dimensions to tractive performance. A need for a new traction model that accurately and reliably represents tire’s dimensions and deflection is needed. The development of such a model would lead to a better understanding of the soil – traction dynamic. To create such a model, a way to measure the capabilities of different tires in a controlled environment is vital. An agricultural tire dynamometer was built in an attempt to show the relationship between the dimensions of a tire and the capabilities of said tire in different scenarios.

### Table 8. Comparison of Motion Resistance Ratio of the New Logarithmic, Wismer-Luth, and Brixius models (Elwaleed et al., 2004)

<table>
<thead>
<tr>
<th>Soil classification</th>
<th>Wheel numeric, $C_s$</th>
<th>Mobility number, $B_s$</th>
<th>Predicted motion resistance</th>
<th>Percentage difference, %</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>New equation</td>
<td>Wismer - Luth</td>
</tr>
<tr>
<td>Soft</td>
<td>15</td>
<td>13.13</td>
<td>0.17</td>
<td>0.12</td>
</tr>
<tr>
<td>Tilled</td>
<td>20</td>
<td>17.50</td>
<td>0.16</td>
<td>0.10</td>
</tr>
<tr>
<td>Firm</td>
<td>30</td>
<td>26.25</td>
<td>0.13</td>
<td>0.08</td>
</tr>
<tr>
<td>Hard</td>
<td>50</td>
<td>43.75</td>
<td>0.10</td>
<td>0.06</td>
</tr>
</tbody>
</table>
The agricultural tire dynamometer test stand’s goal was to study how the dimensions of a tire influence tractive capabilities independent of soil conditions. To do this, the test stand used a series of belts to run the tire on eliminating the soil variance. Along with the belt, an airbag assembly controlled how much contact area present between the tire and belt. Wheel load was controlled using a hydraulic cylinder. Hydraulic pumps were used to change the torque and speed settings of the tire depending on the experiment being ran. These components were essential in changing the tire’s environment reliably and making accurate measurements possible, which could lead to the development of a better representation of the way a tire’s dimensions impact tractive performance.
Objectives

- Design and develop a test stand to model the forces inflicted on a large agricultural tire in a controlled and repeatable environment. The test stand should include the following three modes.
  - Slip Control Operating Mode
    - User defined speed and slip values, along with which drive to target as the leading drive (belt or tire).
  - Speed Control Operating Mode (No Slip)
    - User defined speed value, zero slip, so belt and tire drives operate at same speed.
  - Tractor Simulation Control Operating Mode
    - User defined draft and speed values, simulate tractor engine torque curve setting the torque seen by the tire at selected speed and gear.

- To model real life situations efficiently, the test stand must meet the following.
  - Able to perform at the power level of the majority of tractors available on the market.
    - Sizing based on a large four-wheel drive tractor, as it delivers the greatest pulling force and provides the most wheel load.
  - Must be able to mount any large agricultural tire on the market.
  - Minimize the supply pump of the hydraulic circuit.
    - Simulink model of regenerative hydraulic drive circuit.
Design Overview

The idea of the agricultural tire dynamometer test stand was to study how the dimensions of a tire influence the motion resistance ratio and tractive force coefficients independent of soil conditions. One attribute desired was to maintain and change the amount of contact area between the belts and tire (Figure 7). The tire in question was run on a series of belts to allow the deformation of those belts to form to the shape of the tire and maintain the desired contact area.

Figure 7. Agricultural Tire Dynamometer Test Stand Model
To change the contact area between the tire and belts the test stand has a bank of truck airbags that allow the belt to deform accordingly (Figure 8).

The test stand’s controller was able to provide the user with the torque and/or angular velocity of both the belt and tire drives. This data could then be equated into the tractive force of the tire and resultant slip.

Since the belts of the test stand experience different velocities and torques, an auto-tensioning system was created to maintain optimum belt tension, increasing the belt life. Linear bearings operate in conjunction with two hydraulic cylinders to tension the belts (Figure 9). The belts adjust to a minimum tension as to not allow slippage between the belts and sheaves.

Figure 8. Idler Airbag Assembly
Belt Sizing

The Gates program, Design Flex® Pro, was used to size the belts being selected for the test stand, based on belt power, and force requirements. The program inputs included center distance between sheaves, diameter of sheaves, power transmitted through belts, and maximum angular velocity of the belts.

The Challenger MT975B is one of the largest, most powerful tractors on the market. The capabilities of this tractor were used to size the belts of the test stand. The maximum power the MT975B is capable of is 518.2 hp (Appendix A1, Hoy et al., 2008). Assuming that the power was equally split between eight wheels and neglecting power losses, each wheel would experience 64.8 Hp. The test stand design power used was 100 Hp, providing a factor of safety of 1.54, for the power requirements. The maximum drawbar power for most tractors occurs at the 5.0 to 6.0 mph
range. With 20-inch drive sheaves, the belts were determined to travel at a tangential speed of 472 fpm.

A Gates Predator® PowerBand® 10/CP225 was selected from the list of recommended belts. The program results showed that a single 10/CP225 belt to be capable of all of the torque and forces required, although three 10/CP225 belt belts were used in the test stand. The three belts each have 10 ribs/strands, a C-belt cross section, and are 225 inches in length (Figure 10). Three

![Industrial Belt Design - Drive Detail Report](image)

**Figure 10. Gates Flex® Pro Belt Drive Details Report**
belts, 30-inches in width total, encompass the entire cross-sectional width of a large agricultural tire. This provided the ability to fully test narrow (single belt) and wide tires (3 belts) at the design power.

Shaft Design

According to the report provided by the Gates Flex® Pro software, the belts used on the test stand require a 14,200 lb\(f\) tension force to achieve the 100 hp at 472 fpm (Figure 10). The 14,200 lb\(f\) tension force was used as the transverse loads acting on the main drive sheaves, each sheave would then see 4730 lb\(f\).

The weight of the drive and idler sheaves are 169.7 lb\(m\) and 74.1 lb\(m\) respectively (QD Sheaves). These weights were added to the transverse loads stated previously. Although the weight of the sheaves and tension force are acting perpendicular to each other, they were assumed to act in the same direction to make the equation simple, along with providing an even worse case scenario for the sizing. There are three drive sheaves on a drive shaft and three idler sheaves on an idler shaft. Any axial loads present on both the idler and drive shafts were considered negligible if non-existent.

A modified equation derived from the Von Mises theoretical failure envelope determined the appropriate diameter of the drive and idler shafts (Equation 14). The unitless fatigue stress concentration factors of the Von Mises theoretical failure envelope are used whenever there is a change in diameter of a shaft. These changes in diameter could be a fillet, shoulder, or taper. Since a uniform solid shaft was chosen and there are no diameter changes, both the torsion and bending fatigue concentration factors are equal to 1.

The shafts were made from 1045 carbon steel. This steel has a yield and ultimate tensile strength of 70,000 psi and 80,000 psi respectively. Equation 14 uses this information to determine
a minimal shaft diameter while maintaining bending and torsional stresses within the failure
envelope of the decided factor of safety.

A free body diagram is used to show the location of the transverse forces and torsion
inflicted on the drive shafts (Figure 11). These forces relate to a shear and moment diagram
(Appendix A2). From these diagrams a maximum bending moment was found, which was used in
the modified Von Mises equation. Since the torque was consistent throughout the entire length of
the shaft, a maximum torque was found using the assumed power, tangential speed, and the
diameter of the large drive sheaves (Appendix A2).

Using the modified Von Mises equation (Equation 14, Budynas et al., 2015), it was found
that the required diameter of shaft needed to meet the maximum bending moment and torque
required was 3.63 inches, assuming a factor of safety of 1.20. A maximum bending moment and
torque of 139,000 in-lbf and 74,100 in-lbf, respectively, were found to size the shaft. A 4-inch shaft

\[
d = \left\{ \frac{16 \times n}{\pi} \left( \frac{2K_f M_a}{S_e} + \frac{\sqrt{3}K_{fs} T_a}{S_{ut}} \right) \right\}^{1/3}
\]

(14)

where

- \(d\) = Recommended Diameter of Shaft (in)
- \(n\) = Factor of Safety (unitless)
- \(K_f\) = Bending Fatigue Stress Concentration Factor (unitless)
- \(M_a\) = Alternation Bending Moment (in-lbf)
- \(S_e\) = Endurance Strength (psi)
- \(K_{fs}\) = Torsion Fatigue Stress Concentration Factor (unitless)
- \(T_a\) = Alternating Torque (in-lbf)
- \(S_{ut}\) = Ultimate Tensile Strength (psi)
was chosen to power the drive sheave assemblies, a common size for the QD bushings used with the sheaves. Since a 4-inch shaft was chosen, the factor of safety was increased to 1.61. This makes it an appropriate choice to hold up to the amount of stress that the test stand could inflict onto the shafts.

The Challenger MT975B has a gross weight of 50,175 lb with duals, equating to 6,339 lb per tire (Hoy et al., 2008). A wheel load of 7000 lb was assumed as the downforce present on two idler shafts. Two sheaves were assumed to experience this load per shaft. This was the transverse load present on the idler shafts used to size them.

The idler sheave assembly shafts were sized in the same manner as the drive shafts. However, since they are idler shafts, no torque was transmitted through them. A similar free body diagram is used to show the spacing of the forces present on the idler shafts (Figure 12). Again, a shear force and bending moment diagram was developed (Appendix A2).

With no torque present, the maximum bending moment of 29,200 in-lb was used to size the idler shafts. With a factor of safety of 1.20, the modified Von Mises equation suggested a 2-inch diameter shaft. A 2.5-in shaft was used as the idler shaft of the test stand to coincide with

*Figure 11. Main Drive Sheave Assembly Free Body Diagram*
stock QD bushings. With the use of a 2.5-inch shaft, the factor of safety was increased to 2.35 which was adequate for our application.

Keyways were cut into the shafts at the locations of the bushings. A 5/8-in x 5/8-in key was used for the idler shafts and a 1-in x 5/8-in key was used for the drive shafts.

**Hydraulic Machine Sizing**

The test stand will need hydraulic pumps to operate as a pump or motor depending on the experiment being ran. To avoid confusion, the dynamometer’s hydraulic pumps will be referred to as hydraulic machines. The hydraulic machines were sized based off of the MT975B test results.

The large 4-wheel drive Challenger was shown to produce a maximum drawbar pull of 32,100 lb at 2101 rpm in 8th gear traveling at 5.45 mph (Appendix A1, Hoy et al., 2008). Dividing this maximum drawbar pull between the 8 tires used by the MT975B, each wheel provided 4,010 lb of net tractive force (P) to the ground. The test stand’s tire has a diameter that is 3.1 times greater than the diameter of the drive sheaves, meaning that the tire drive will require the most torque to achieve this 4,010 lb (Appendix A3). With a 62 inch diameter tire, a torque of 124,000 in-lb was needed.

*Figure 12. Idler Sheave Assembly Free Body Diagram*
The Oerlikon Fairfield torque hub® has a continuous torque rating of 48,700 in-lbf (4,060 ft-lbf) (Torque Hub Wheel Drive Products, 2003). This rated torque of the torque hub would achieve 39.3% of the torque needed to match the torque generated by the MT975B that the test stand must be able to generate. However, the rated torque is dictated by the durability and lifespan of the bearings within the torque hub. Normally, heavy machinery would use these torque hubs to power their wheel drive systems, where they would experience significant transverse and axial forces. Whereas, on the dynomometer itself, both shafts of the tire and belt drives are fully supported by bearings. The torque hubs then only experience a transverse force from their own weight. The hubs will also experience no axial force. Torque will be the sole significant force that the hubs experience. Additionally, the test stand will not achieve the amount of hours of wear that these torque hubs would see in more common applications. Thus for these reasons the hubs were chosen to transfer the torque of the hydraulic machines into the shafts of the drives.

The belt drive has a mechanical advantage on the tire drive because of the smaller diameter sprocket. To stop the belt from being able to overpower the tire, the two torque hubs must have a similar ratio as the tire and belt diameters, 3.1:1. To generate the maximum speed possible, the smallest ratio available of 14.88:1 was chosen for the belt drive. The closest available reduction ratio to achieve the ideal 3.1:1 ratio was 47.6:1. This ratio reduced the maximum torque of the tire drive from 124,000 in-lbf to 2,610 in-lbf at the hydraulic machine (Appendix A3).

Danfoss 53 cm³ displacement H1 hydraulic pumps were sized to meet these requirements (Appendix A3). Again, these pumps perform as hydraulic machines being able to run as a pump or a motor depending on the swash plate angle. However, they are not able to rotate in both directions. These conditions were based off of the MT975B and H1 pump’s rated parameters (Table 9).
The belt drive’s H1 pump limits the speed of the test stand because of the smaller diameter sheaves. The belt drive’s torque hub has a reduction ratio of 14.88:1. This reduction ratio was used to find the maximum speed that the test stand can operate at. The torque hub reduces the maximum rated output speed of the hydraulic machine from 3,400 rpm (Table 9) to 228 rpm. Referencing to the circumference of the belt sheave, the maximum angular velocity of the test stand was found to be 13.5 mph.

**Primary Mover Sizing**

The primary mover used to power the test stand was an electric motor. This electric motor was connected to a hydraulic pump stack containing a main supply pump, charge pump, and actuator circuit supply pump. The power necessary to rotate this stack of hydraulic pumps was determined by multiplying the pressure and flowrate required by the test stand.

The charge pump sized for the test stand operated at 290 psi with a flowrate of 1.2 gpm. This equated to 0.2 hp. Likewise, the actuator circuit pump required a maximum of 2 gpm at 500 psi. This equated to 0.6 hp. Thus, the auxiliary pumps required 0.8 hp to operate.

A MATLAB® Simulink model was created to simulate and prove the concept for the hydraulic drive system of the test stand. Chapter 3 presents this in detail. Hydraulic flowrates were recorded and used for the sizing of the electric motor from the model’s simulation.

<table>
<thead>
<tr>
<th>Hydraulic Machine Sizing Conditions</th>
<th>218 ft-lbr</th>
<th>2610 in-lbr</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Torque</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Maximum Rated Output Speed</td>
<td>3400 rpm</td>
<td></td>
</tr>
<tr>
<td>Working Pressure</td>
<td>5510 psi</td>
<td></td>
</tr>
</tbody>
</table>

**Table 9. Hydraulic Machine Rated Parameters**

(H1 Axial Piston Single Pumps Size 045/053, 2015)
The test stand required the most power while in the slip control operating mode, where the drive system required 16.1 gpm at 5590 psi. Thus, the main supply pump required 52.6 hp. However, although the slip control operating mode is attainable by the test stand, it was not used in the sizing of the primary mover. The slip control operating mode was predicted to cause a large amount of wear and strain on the belts of the test stand. Since the belts are some of the most expensive and critical components of the test stand, the electric motor was undersized to discourage future operators from performing such an experiment.

The test stand’s primary use will be to simulate the forces present on a tractor’s tire, thus the tractor simulation operating mode will be used to size the motor. The tractor simulation operating mode required 5.37 gpm at 5590 psi, resulting in a 17.5 hp requirement.

With the summation of the auxiliary pumps and main supply pump power requirements the test stand required 18.3 hp to operate. The primary mover of the hydraulic circuit of the test stand selected was a 3-phase, 80 amp, 30 hp electric motor. This provided a surplus of 11.7 hp to deal with any system losses or inaccuracies of the simulation.

Hydraulic Reservoir Sizing

The main supply pump has a 53 cm$^3$ displacement. The charge pump and actuator circuit pump have a displacement of 44 cm$^3$ and 4 cm$^3$, respectively. Thus, the max flowrate possible of the main supply pump and auxiliary pumps combined is 92 gpm. To sufficiently cool the hydraulic system, a 30 sec dwell time was recommended. Therefore, a 46 gallon or larger reservoir was necessary for the test stand. A 50 gallon reservoir was chosen.

Hydraulic Circuit Design

The regenerative circuit design for the test stand consisted of a drive circuit and actuator circuit fed by a stack of three pumps attached to the primary mover (Figure 13). The three
stacked pumps consisted of the charge pump, actuator pump, and main supply pump. For reference, a non-regenerative hydraulic circuit diagram is shown in Appendix A5.

The drive circuit is the regenerative circuit as described more thoroughly in Chapter 3. It is a closed loop circuit supplied by the main supply pump where the stroke of the pumps were controlled by a Plus +1 controller depending on the experiment being ran. The charge pump and case drains replenished cool oil into the drive circuit. The drain line was fed into a heat exchanger controlled by a thermostat.

The actuator circuit has two sections plumbed in parallel. The first part is the downforce cylinder where the Plus +1 controller controlled the amount of wheel load that the wheel of the

![Figure 13. Complete Regenerative Hydraulic Circuit Diagram](image-url)
dynamometer saw. Then, the second section contains two other hydraulic cylinders plumbed in parallel that controlled the tension of the belt. By measuring the velocity of the belt and sheave assemblies, a slip was calculated between the belts and sheaves. This slip was used to maintain the tension of the belts at the minimal force to prevent any slippage. In this way belt life was kept to a maximum. The actuator circuit was able to control force exhibited by the hydraulic cylinders of the two sections of the circuit independently.
CHAPTER 3. REGENERATIVE HYDRAULIC CIRCUIT MODEL

Overview

MATLAB® Simulink was used to create a model of the proposed regenerative hydraulic circuit. The goal of the regenerative circuit was to minimize the size of the hydraulic power pack needed to power the test stand. To minimize the power pack, the individual belt and tire drives were hydraulically linked (Figure 14). The separate hydraulic machines, could perform as a pump or motor. Therefore, when the hydraulic machine of the tire drive was in motor mode, the tire was being driven. Since the tire was rotating on the belt, the belt was also being driven by the tire’s hydraulic machine. The belt’s hydraulic machine, in pump mode, then supplied supplementary flow back into the tire’s hydraulic machine. With a change in the swash plate angle, the tire and belt hydraulic machine drives could help power one another, if needed. Thus, the hydraulic power pack or supply pump only needed to make up the energy lost between the difference in drive diameters, speeds, or torques of the drives depending on the specific experiment being ran. When using the Simulink model, the H1 pump rated parameters were the limits as to which the model was compared to (Table 9).

The Simulink model contained the drive hydraulic machines and corresponding main supply pump to represent the regenerative circuit. Along with the fluid power components, a mechanical model and control circuit were also included to represent the test stands operating conditions (Figure 15). The hydraulically linked regenerative circuit was compared to a simpler, non-regenerative, circuit containing two individual hydraulic motors (Figure 16). These two motors had two individual supply pumps providing them with flow. This non-regenerative circuit was used to show the advantages of using a regenerative hydraulic circuit.
**Figure 14.** Simplified Regenerative Hydraulic Circuit Diagram

**Figure 15.** Regenerative Hydraulic Drive Simulink Model
Mechanical Model

A mechanical model was developed to simulate the tire and belt interaction (Figure 17). This model utilized a translational friction block to simulate the interaction of the contact area of the tire and belt drives. The speed of the belt and tire was measured in this translational circuit in miles per hour. Then, a wheel and axle block turned this translational movement into rotational movement, where the torque was read for the tire and belt drives.

The test stand’s tire and belt drive inertias were represented in the model. Inertia is the amount that an object resists rotating. It is analogous to mass except that inertia deals with the rotation of objects, whereas mass deals with the translational movement of objects. For example, a translational movement can be thought of sliding a box along the ground in a linear movement, where mass is the culprit for the resistive force, also known as friction force, felt when sliding the box.

The belt drive has an inertia of 5,360 lbm-ft² (Appendix B2). This inertia was determined by adding the inertia of both main sheave assemblies. The inertia of the tire drive is the summation

**Figure 16. Non-Regenerative Hydraulic Drive Simulink Model**
of the tire and shaft’s inertias used to drive the tire, which was determined to be 2,170 lb\textsubscript{m}-ft\textsuperscript{2} (Appendix A4).

A Coulomb friction force was needed by the translational friction block to know at which point the tire would lose traction with the belt. A dynamic rubber on rubber friction coefficient could not be found. However, for rubber on concrete the dynamic friction coefficient is approximately 0.80 (Coefficient of Friction, 2004). This value was used as the assumed dynamic friction coefficient. By taking the wheel load of 7,000 lb\textsubscript{f} and multiplying by the assumed dynamic friction coefficient, a 5,600 lb\textsubscript{f} Coulomb friction force was calculated and implemented into the model.

\textbf{Figure 17. Mechanical Simulation of Simulink Model}

Control Circuit

The controls of the model utilized two PID controllers to individually control the tire and belt drive hydraulic machines. This system of control was also implemented onto the test stand (Figure 18).

The main control circuit consisted of three modules; the control parameters, tire PID control, and belt PID control. The PID control modules both utilize a proportional-integral-derivative (PID) controller. This controller uses the difference (error signal) between a desired value (set point) and a reading of the current value (feedback signal) as its input. Then the controller outputs a control signal to the actuator, in this case, the displacement of the hydraulic
machines. The controller uses a proportional gain to reach the constant set point as quickly as possible, simultaneously, the derivative gain stunts this proportional gain if it is increasing too rapidly and decreases the overshoot when the feedback signal overtakes the set point (Figure 19). As the feedback signal starts a sinusoidal curve around the set point, the integral gain minimizes the amount of area that the feedback signal has in comparison with the set point. In this way, the PID controller attempts to minimize the error signal as quickly as possible without significant overshoot.

The tire and belt drive’s hydraulic machines were controlled by the two individual PID control subsystems. Depending on the control mode desired by the user, the set point and feedback signal were either the torque or angular velocity of the corresponding drive. Irrespective of the feedback signal (torque or angular velocity), the PID control output a

**Figure 18. Main Control Circuit of Simulink Model**

**Figure 19. Unit Step Response of a PID Controller**
command to change the hydraulic machine stroke, effectively changing the swash plate angle of the corresponding hydraulic machine. Essentially, the controllers changed the displacement of the hydraulic machines to control either the angular velocity or torque of the individual drives.

The control parameters module contained the subsystems containing the tractor torque simulation, speed control, and slip control operating modes. An input of 1, 3, or 4 changed the control operating mode to tractor torque simulation, belt leading speed and slip control, or tire leading speed and slip control respectively.

The operating modes, speed control and slip control were both possible with the belt or tire leading. Both belt and tire leading speed and slip control operating modes required a set speed and slip input. In fact, the only difference between the two operating modes was the choice of slip input. If the slip input equaled zero the simulation would be operating in speed control operating mode, where the belt and tire drives would be rotating at the same velocity. Whereas, if slip were greater than zero then the simulation would operate in slip control operating mode. When ran in slip control operating mode, with the tire leading, the tire drive would reach the desired speed, with the belt drive achieving the reduced speed due to the inputted slip or travel reduction. Conversely, if the belt were the leading drive, the tire drive would achieve that reduced speed. Although, there is the capability of using the belt as the leading drive, it was assumed that the slip control operating mode would have the tire leading in most cases.

**Slip control operating mode.**

One of the objectives of the test stand was to control the individual velocities of the tire and belt drives to reach desired slip percentages between them. By comparing the torque outputs of the model to the hydraulic machine rated conditions (Table 9), hydraulic machine damage was avoided. This determined whether or not the test stand was capable of a suggested testing situation.
In slip control operating mode, the model showed that the tire and belt drives experienced 10,700 ft-lbf and 3,470 ft-lbf, respectively. When reduced by the torque hub, the hydraulic machines of the tire and belt drives saw 226 ft-lbf and 233 ft-lbf, respectively. This was true for all velocity and slip combinations as it was the torque required to break the grip that the tire had on the belt. The tire drive hydraulic machine saw 226 ft-lbf, which was 3.7% more load than what the hydraulic machine was rated for (Table 9). With such a small percentage, this overloading of the H1 pump was not of concern.

A wheel load of 7,000 lb resulted in too much friction force for the hydraulic machines to overcome and allow the tire to slip on the belt. As stated before the friction coefficient used was not a direct rubber on rubber dynamic friction coefficient. With this reiterated, it was assumed that the model may be inaccurate when finding the amount of downforce that the test stand is able to overcome to induce slip. Investigative testing is recommended to find the maximum downforce that slip can be studied. Regardless, the model performed and showed a correct slip between the tire and belt drives could be achieved with a downforce of 5,200 lb and lower. Meaning that the model’s hydraulic machines were able to overcome a friction force of 4,160 lb.

Assuming that whatever tractor was being studied had eight wheels and the weight of the machine was evenly distributed across all wheels, the model stated that the maximum weight of a simulated tractor possible in slip control operating mode was 41,600 lb. Meaning that all mechanical front wheel drive (MFWD) tractors could be tested in slip control operating mode along with some smaller four-wheel drive articulating tractors underneath the weight limit of 41,600 lb.

A simulation was conducted with the model at the belt drive’s maximum rated speed. Appendix B1 shows the outputs of the regenerative circuit model in slip mode with a set speed and
slip of 13.5 mph and 0.08, respectively. Appendix B2 contains the same slip mode simulation, but instead using a more traditional non-regenerative circuit. Refer to Table 10 for steady state values of the regenerative and non-regenerative circuit models tested in slip control operating mode.

The individual drives in the non-regenerative circuit were connected to separate hydraulic pumps to provide flow to the individual motors. In this configuration, the belt and tire drive supply pumps supplied a combined 68.4 gpm to the system. The regenerative circuit only required 16.1 gpm to achieve the same velocity and slip. Thus, the regenerative circuit required 76.4% less flow than the non-regenerative circuit. The tire and belt drive hydraulic machines required 52.5 gpm and 15.9 gpm respectively.

To evaluate the effect that the slip input had, a comparison of the supply flow of the regenerative and non-regenerative circuit was performed over a range of slip percentages (Figure 19a). This comparison was completed at 7 mph with a range of 0.01 – 0.25 slip.

<table>
<thead>
<tr>
<th>Slip Mode Parameters at Steady State</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Regenerative Circuit</strong></td>
</tr>
<tr>
<td>Hydraulic Machine Torque</td>
</tr>
<tr>
<td>Belt</td>
</tr>
<tr>
<td>Tire</td>
</tr>
<tr>
<td>Individual Drive Torque</td>
</tr>
<tr>
<td>Belt</td>
</tr>
<tr>
<td>Tire</td>
</tr>
<tr>
<td>Hydraulic Machine Speed</td>
</tr>
<tr>
<td>Belt</td>
</tr>
<tr>
<td>Tire</td>
</tr>
<tr>
<td>Individual Drive Travel Speed</td>
</tr>
<tr>
<td>Belt</td>
</tr>
<tr>
<td>Tire</td>
</tr>
<tr>
<td>Supply Flow</td>
</tr>
<tr>
<td>16.1 gpm</td>
</tr>
<tr>
<td>System Pressure</td>
</tr>
<tr>
<td>5590 psi</td>
</tr>
</tbody>
</table>

*Table 10. Predicted Steady State Torques, Speeds, Flows, and Pressures of Regenerative and Non-Regenerative Hydraulic Drive Circuits in Slip Mode Performed at 13.5 mph and 8% slip.*
At 0.01 slip, the regenerative circuit required 81.5% less flow than the non-regenerative circuit; whereas, at 0.25 slip the regenerative circuit required 67.7% less flow than the non-regenerative circuit. In average, the regenerative circuit (9.95 gpm) required 74.7% less flow than the non-regenerative circuit (39.3 gpm) over the specified slip range.

Figure 20. Supply Flow Difference between Regenerative and Non-regenerative Circuits in Slip Operating Mode. Slip Range (a) (7mph, 1-25% Slip) and Speed Range (b) (1-12.5mph, 5% Slip)
Another comparison test was created to show the difference in supply flow when the velocity of the test stand was changed from 1 mph to 12.5 mph while in slip mode (Figure 19b). This was done at 5% slip with the belt as the leading drive component. There was a large change in the supply flow for the non-regenerative circuit over this speed range. There was also a change in the supply flow of the regenerative circuit, but not as substantial as the non-regenerative circuit.

At 1 mph the regenerative circuit (1.54 gpm) required 77.3% less flow than the non-regenerative circuit (6.78 gpm). Conversely, at 12.5 mph the regenerative circuit (13.9 gpm) required 78.4% less flow than the non-regenerative circuit (64.3 gpm). Regarding the average supply flow of the two circuits over their velocity range, the regenerative circuit (7.62 gpm) required 79.0% less flow than the non-regenerative circuit (36.2 gpm).

The non-regenerative circuit required 89.5% more flow as the linear speed of the model increased from 1 mph to 12.5 mph. Whereas, the regenerative circuit required 88.9% more flow, which was close to the non-regenersives circuit’s 89.5% increase in flow. With similar percentage increases in supply flows, the non-regenerative and regenerative circuits had similar sensitivies to the speed input.

The difference in flow between the non-regenerative and regenerative circuits justified the decision to use the regenerative circuit over the non-regenerative circuit. With such a drastic flow difference, the regenerative circuit was able to use a much smaller supply flow that was able to perform all the duties of a two motor, two supply pump, non-regenerative circuit. With a smaller and more efficient singular supply pump, the primary mover could then be sized smaller. This made the test stand able to perform at the levels needed using a much smaller electric motor; saving money, energy, and space. The power requirements for the regenerative circuit in slip control
operating mode was 52.5 hp. Whereas, the power requirements for the non-regenerative circuit in slip control operating mode was 223 hp.

**Speed control operating mode.**

When the commanded slip was equal to zero, the simulation was considered to be in speed control operating mode, where the speed was critical and the torque was not. A simulation was run at the top speed, where the test stand is limited to 13.5 mph due to the rated speed of the belt drive hydraulic machine (Table 9). Either the tire or belt drive could have been targeted to achieve whatever set speed was desired, as both would achieve the same speed. In the regenerative circuit, having the tire or belt drive targeted only changed which drive’s hydraulic machine would be acting as a motor or pump.

The non-regenerative and regenerative circuits converged to zero torque when the target speed was achieved (Appendices B3 & B4). Both hydraulic circuits achieved the same rotational speed, drive travel speed, and system pressure when compared to one another (Table 11). In the speed control mode, the regenerative

<table>
<thead>
<tr>
<th>Table 11. Predicted Steady State Torques, Speeds, Flows, and Pressures of Regenerative and Non-Regenerative Hydraulic Drive Circuits in Speed Mode Performed at 13.5 mph and 0% slip.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Speed Mode Parameters at Steady State</strong></td>
</tr>
<tr>
<td>Regenerative Circuit</td>
</tr>
<tr>
<td>Hyrdraulic Machine Torque</td>
</tr>
<tr>
<td>Belt</td>
</tr>
<tr>
<td>Tire</td>
</tr>
<tr>
<td>Individual Drive Torque</td>
</tr>
<tr>
<td>Belt</td>
</tr>
<tr>
<td>Tire</td>
</tr>
<tr>
<td>Hydraulic Machine Speed</td>
</tr>
<tr>
<td>Belt</td>
</tr>
<tr>
<td>Tire</td>
</tr>
<tr>
<td>Individual Drive Travel Speed</td>
</tr>
<tr>
<td>Belt</td>
</tr>
<tr>
<td>Tire</td>
</tr>
<tr>
<td>Supply Flow</td>
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<tr>
<td>0.795 gpm</td>
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<tr>
<td>System Pressure</td>
</tr>
<tr>
<td>5590 psi</td>
</tr>
</tbody>
</table>
circuit (0.795 gpm) required 83.3% less flow then the non-regenerative circuit (4.75 gpm). The tire and belt drive hydraulic machines required 2.76 gpm and 1.96 gpm respectively.

The power requirements for the regenerative circuit in speed control operating mode was 2.59 hp. Whereas, the power requirements for the non-regenerative circuit in speed mode was 15.5 hp.

**Tractor simulation operating mode.**

To simulate the torque curve and draft that a tractor’s wheel experiences, the velocity and torque must be controlled individually. To control the torque, the PID controller of the tire drive used the torque of the wheel before being reduced by the torque hub as its feedback signal. The controller, then, managed the torque that the wheel experienced simulating the torque curve of a tractor’s engine. The PID controller of the belt drive, however, achieved a set speed rather then a set torque. Thus, the test stand remained in a suitable velocity range for the hydraulic machines and also simulated the torque that a tractor’s wheel would experience under load.

Due to the limited availability of a modern tractor engine torque curve, an assumed engine torque curve was created to showcase the tractor simulation operating mode. The maximum drawbar produced by a 4-wheel drive Challenger MT975B tractor is 32,089 lb ft in 8th gear at 5.45 mph (Appendix A1). This drawbar pulling force was divided by the 8 tires of the tractor and then converted into a torque using the radius of the wheel. A maximum torque of 10,000 ft-lb was found for each wheel of the tractor. Then, after the determined torque was reduced by the test stand’s torque hub, the tire’s hydraulic machine would experience 218 ft-lb. Since the actual torque curve is unknown it was assumed that this torque occurred at 1600 rpm (Figure 21). From Appendix A1 the engine speed and travel speed were known in 8th gear, leading to the calculation
of an overall gear ratio. At an engine speed of 2000 rpm and a travel speed of 5.45 mph the overall gear ratio was equal to 385.5.

This overall gear ratio was used in the tractor engine subsystem of the Simulink® model to convert the wheel velocity into an engine speed, this was then inputted into the 1-D lookup table (Figure 22). This engine speed was then turned into the engine’s torque with the 1-D lookup table containing the same torque curve as in Figure 21. Then, after being reduced by the over gear ratio and split between 8 wheels, the corresponding torque of the tire would be sent to the PID controller operating the tire drive. The wheel would then experience the torque of the engine over the simulation’s run time.

To control the belt drive, a relationship had to be formed between the engine’s torque and tire velocity. Thus, a tractive force was calculated as an input of the draft subsystem (Figure 22). This tractive force was derived by dividing the wheel’s corresponding torque from the tractor engine subsystem by the wheel’s radius. With the tractive force determined, the draft force of the implement was subtracted creating a net force experienced by the wheel. This net force was then divided by the axleload, which equated to the belt’s acceleration, and then integrated to determine the velocity set point of the belt drive’s PID controller. This made it possible for the control circuit to simulate a tractor wheel’s torque and velocity simultaneously. Appendix A6 shows a breakdown of this algorithm.
An experiment was run with a 3,250 lb\(_f\) draft force introduced gradually after 30 seconds. This allowed the simulated tractor engine to speed up before lowering the simulated implement into the ground and introducing a draft force. The experiment was conducted in 8\(^{th}\) gear at 5.45 mph, which modeled the tractor’s greatest available tractive force situation.

Figure 24 and appendix B5 shows the results of this test. As seen in the tractor simulation forces graph the tractive force spiked and then dropped down when the travel speed was attained. Then, the tractive force increased when the draft force was introduced, where it lagged behind until it met the 3250 lb\(_f\) max draft force of the implement in the ground. With the introduction of the draft force, there was a corresponding drop in the engine speed and travel speed. This simulated the response of a tractor and thus represented the torque and velocity that the tractor’s wheel would experience.

Appendix C6 shows the results of the same test, except completed with the non-regenerative hydraulic circuit model. The same results were attained, except for a difference in supply flow and system pressure (Table 12). The non-regenerative circuit had a supply flow and system pressure of 26.3 gpm and 6290 psi, respectively. Whereas, the regenerative circuit had a supply flow and system pressure of 5.37 gpm and 5590 psi, respectively. The regenerative circuit (5.37 gpm) required 79.6% less flow then the non-regenerative circuit (26.3 gpm). The regenerative circuit was again found more efficient reiterating the point that this hydraulic circuit design was the best fit for the test stand. The tire and belt drive hydraulic machines of the
Figure 22. Regenerative Circuit, in Tractor Sim. Mode, performed at 5.45 mph, 3250 lbf Draft Force, Simulating an MT975B’s 8th Gear Engine Torque Curve.
regenerative circuit required 16.7 gpm and 11.3 gpm respectively. Whereas, the non-regenerative circuit’s tire and belt drive hydraulic machines required 16.7 gpm and 9.67 gpm respectively.

Another test was performed to show that the simulated engine will stall if too much draft force was present (Appendix B7). This tractor simulation operating mode was limited because it did not model the transmission of a tractor. If it were to model a transmission, the user would be able to see gear shifts as the model avoided a stall. However, for the purpose of this simulation, a single gear was sufficient as a proof of concept. Once the engine speed dropped below 800 rpm, the engine stalled and the simulation stopped. On the test stand itself the drive system would coast to a stop and report back that the simulated engine had stalled.

In all three control operating modes, the regenerative hydraulic circuit was found to outperform the non-regenerative circuit by requiring less supply flow. This allowed a reduction in

Figure 23. Tractor Engine Subsystem of Simulink Model

Figure 24. Implement Draft Subsystem of Simulink Model
size of the primary mover compared to the primary mover of the non-regenerative circuit. The power requirements for the regenerative circuit in tractor simulation mode was 17.5 hp. Whereas, the power requirements for the non-regenerative circuit in tractor simulation mode was 96.5 hp.

**Table 12. Predicted Steady State Torques, Speeds, Flows, and Pressures of Regenerative and Non-Regenerative Hydraulic Drive Circuits in Tractor Simulation Mode Performed at 13.5 mph and 3250 lb$_f$ draft force.**

<table>
<thead>
<tr>
<th>Tractor Sim. Mode Parameters at Steady State</th>
<th>Regenerative Circuit</th>
<th>Non-regenerative Circuit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Draft Force</td>
<td>3250 lb$_f$</td>
<td>3250 lb$_f$</td>
</tr>
<tr>
<td>Tractive Force</td>
<td>3250 lb$_f$</td>
<td>3250 lb$_f$</td>
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<tr>
<td>Individual Drive Torque</td>
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<td></td>
</tr>
<tr>
<td>Belt</td>
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<td>-2700 ft-lb$_f$</td>
</tr>
<tr>
<td>Tire</td>
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<td>8400 ft-lb$_f$</td>
</tr>
<tr>
<td>Individual Drive Travel Speed</td>
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<td></td>
</tr>
<tr>
<td>Belt</td>
<td>5.24 mph</td>
<td>5.24 mph</td>
</tr>
<tr>
<td>Tire</td>
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<td>5.24 mph</td>
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<tr>
<td>Engine Speed</td>
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</tr>
<tr>
<td>Supply Flow</td>
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<td>2020 rpm</td>
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<tr>
<td>System Pressure</td>
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</tr>
<tr>
<td></td>
<td>5590 psi</td>
<td>6290 psi</td>
</tr>
</tbody>
</table>
CHAPTER 4. CONCLUSION

The agricultural tire dynamometer test stand was designed and built over the course of two years. In hindsight, some minor elements could be improved on during that time.

The electric motor used to run the hydraulic system was undersized as noted in Chapter 2. This was done to protect the belts and discourage future operators from operating in slip control operating mode as described in Chapter 3. The belts are a few of the most expensive and integral parts contained within the test stand. They are also one of the first things installed with most of the test stand built around them. Thus, if the dynamometer had been designed with the ability to quickly change the belts, the slip control operating mode would have been a more feasible operating mode.

An attempt at a quick assembly and disassembly tool was fabricated. A belt spear was made to be able to lift the belts along with the sheave assemblies in order to switch out belts quickly (Appendix C). However, the spear had tolerances that were too tight to be able to slide out smoothly when the entire belt sheave assembly was placed. For the time being, a standard operating procedure was written to change out the belts with a combination of the belt spear, overhead crane, and forklift as needed (Appendix C). The belt spear was saved for future improvements and modifications if so desired. If a quick change system were created, the replacement of belts would not be such a strenuous endeavor. Leading to the option of increasing the size of the electric motor and being able to use the test stand in slip control operating mode. Although the belt life during this control operating mode would be in question.

Another concern if the slip control operating mode were to be used, was that the torque hub’s rated torque were equal to 39.3% of the torque needed to make a simulated MT975B’s wheel slip. Since the MT975B is one of the most powerful and heaviest tractors on the market the test
stand may not even be able to generate the torque needed to overcome the wheel’s friction force. Thus, the test stand’s torque inflicted on the wheel and belts should be monitored with great care if this operating mode were to be used.

Three regulators with one common shutoff valve between the air inlet and the regulators control the idler airbag assembly. When the air supply line is disconnected from the pneumatic system, the regulators proceed to leak air until no pressure is present. To improve on the system it is recommended to plumb three individual shutoff valves after the regulators to stop the slow leaks.

The test stand is meant to run a variety of experiments with an emphasis on exploring how the dimensions of a modern high-lug agricultural tire influences tractive capabilities. It is able to change the wheel load, draft, speed, and torque inflicted on the tire during operation. The contact area between the belt and tire can also be changed to represent different situations. With the ability to control this variety of parameters, the test stand is capable of many different experiments dealing with a wheel’s tractive capabilities. For example, by controlling the wheel load, speed, and torque of the tire, a field recording can be played on the test stand. This would provide insight on the forces experienced during normal field operation and how they influence tire characteristics. The test stand would simulate the same change in wheel load due to the topography of the land along with the torque necessary to traverse said land. All while providing an environment that minimizes uncontrollable variables and provides accurate and repeatable data.

There are two challenges faced when using the Wismer-Luth and Brixius models to compare tractive performance capabilities between tires, tire dimension representation and cone index variability. The study conducted by UPM reiterates this point by comparing the predicted motion resistance ratios of the two models with the actual motion resistance ratio.
Torque measurements available from the test stand will result in reliable, repeatable results when comparing different tires. The test stand eliminates one of the largest sources of error, the soil cone index, and focuses on the representation of tire dimensions. An empirical model could then be developed that better represents the tractive capabilities of individual tires, leading to a broader model that encompasses all agricultural tires. This would lead to a better understanding of how specific tire dimensions influence tractive capabilities. When numerous tests are completed with different sizes and models of tires, a new traction dynamic model that minimizes the variance influenced by tire dimension representation, could then be created to help tire manufacturers optimize tire designs and off-road equipment manufacturers find tire models that best meet their needs.
REFERENCES


APPENDIX A: SIZING AND DESIGN NOTES

1. Challenger MT975B Nebraska tractor test results

**NEBRASKA OECD TRACTOR TEST 1931 - SUMMARY 603 CHALLENGER MT975B DIESEL 16 SPEED**

**POWER TAKE-OFF PERFORMANCE**

<table>
<thead>
<tr>
<th>Power (hp)</th>
<th>Crank speed (rpm)</th>
<th>Engine Horsepower (hp)</th>
<th>Horsepower (hp)</th>
<th>Horsepower (hp)</th>
</tr>
</thead>
<tbody>
<tr>
<td>384.10</td>
<td>2100</td>
<td>32.92</td>
<td>106.18</td>
<td>15.39</td>
</tr>
<tr>
<td>264.22</td>
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<td>28.08</td>
<td>90.00</td>
<td>13.05</td>
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</table>

**MAXIMUM POWER AND FUEL CONSUMPTION**

<table>
<thead>
<tr>
<th>Engine Speed (rpm)</th>
<th>Power (hp)</th>
<th>Horsepower (hp)</th>
<th>Horsepower (hp)</th>
</tr>
</thead>
<tbody>
<tr>
<td>384.10</td>
<td>2100</td>
<td>32.92</td>
<td>106.18</td>
</tr>
<tr>
<td>264.22</td>
<td>1750</td>
<td>28.08</td>
<td>90.00</td>
</tr>
</tbody>
</table>

**VARYING POWER AND FUEL CONSUMPTION**

<table>
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<tr>
<th>Engine Speed (rpm)</th>
<th>Power (hp)</th>
<th>Horsepower (hp)</th>
<th>Horsepower (hp)</th>
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</thead>
<tbody>
<tr>
<td>384.10</td>
<td>2100</td>
<td>32.92</td>
<td>106.18</td>
</tr>
<tr>
<td>264.22</td>
<td>1750</td>
<td>28.08</td>
<td>90.00</td>
</tr>
</tbody>
</table>

**ENGINE OPERATING PARAMETERS:**

- Fuel rate: 289.4 - 284.7 lb/ft (99.0 - 105.1 kg/ft)
- High idle: 1730 rpm
- Turbo boost: 15.0 - 23.9 psi (105.9 - 165.4 kPa)

**CHASSIS:**

- Type: four wheel drive with triples
- Serial No.: AGCO07515847 (AGCO07515847)
- Wheelbase: 155.5" (3950 mm)
- Travel: 177.5" (4505 mm)
- Tread: 24.0" (610 mm)
- Speeds: 3.8 to 5.4 mph (6.1 to 8.7 km/h)
- Transmission: 16-speed, 3.8 to 5.4 mph (6.1 to 8.7 km/h)
- Clutch: wet multiple disc hydraulically actuated
- Steering: hydrostatic
- Power take-off: 600 rpm at 2000 engine rpm
### DRAWBAR PERFORMANCE

**Unballasted at 2100 RPM**

**MAXIMUM POWER IN SELECTED GEARS**

<table>
<thead>
<tr>
<th>Gear</th>
<th>Power (hp)</th>
<th>Drawbar pull (lbf)</th>
<th>Speed (mph)</th>
<th>Ground drawbar speed (mph)</th>
<th>Slip %</th>
<th>Foot Torque (psf)</th>
<th>Hp Lb-ft</th>
<th>Horsepower (% of 100)</th>
<th>Efficiency (% of 100)</th>
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<td>(13.30)</td>
<td>(64)</td>
<td>(79)</td>
<td>(97.21)</td>
</tr>
</tbody>
</table>

### REPAIRS AND ADJUSTMENTS

No repairs or adjustments.

**Note:** This tractor has a driveline protection system that limits the maximum engine torque in gears 1 through 7.

### REMARKS

All test results were determined from observed data obtained in accordance with official OTCDA and Nebraska test procedures. For the maximum power tests the fuel temperature at the primary fuel filter was maintained at 114°F (46°C). The performance figures on this summary were taken from a test conducted under the OTCDA Code II test procedure.

We, the undersigned, certify that this is a true and correct report of official Tractor Test No. 1981, Nebraska Summary 680, December 15, 2008.

Roger M. Hoy
Director

M. F. Kocher
V. I. Adamchuk
J. A. Smith
Board of Tractor Test Engineers

### TRACTOR SOUND LEVEL WITH CAB

<table>
<thead>
<tr>
<th>Frequency</th>
<th>dBA</th>
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<td>1000 Hz</td>
<td>560</td>
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<tr>
<td>2000 Hz</td>
<td>-</td>
</tr>
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### TIRES, BALLAST AND WEIGHT

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<th>Without Ballast</th>
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<td>Base Tires - No. in, ply &amp; pld (in)</td>
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</tr>
<tr>
<td>Ballast - Liquid (gallons)</td>
<td>None</td>
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<tr>
<td>- Cylindrical (gallons)</td>
<td>None</td>
</tr>
<tr>
<td>Front Tires - No. in, ply &amp; pld (in)</td>
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</tr>
<tr>
<td>Ballast - Cylindrical (gallons)</td>
<td>400 lbs (180.8 kg)</td>
</tr>
<tr>
<td>- Cylindrical (gallons)</td>
<td>400 lbs (180.8 kg)</td>
</tr>
<tr>
<td>Height of Drawbar</td>
<td>31.5 in (80 cm)</td>
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<tr>
<td>Static Weight with Operator - Rear</td>
<td>4000 lbs (1814 kg)</td>
</tr>
<tr>
<td>- Front</td>
<td>4000 lbs (1814 kg)</td>
</tr>
<tr>
<td>- Total</td>
<td>5000 lbs (2268 kg)</td>
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---
## DRAWBAR PERFORMANCE
(Undersballasted at 1750 RPM)

### MAXIMUM POWER IN SELECTED GEARS

<table>
<thead>
<tr>
<th>Power (hp)</th>
<th>Drawbar pull (lbs)</th>
<th>Speed (rpm)</th>
<th>Crankshaft speed (rpm)</th>
<th>Slip (%)</th>
<th>Fuel Consumption (gpm/kW)</th>
<th>Temp T°C</th>
<th>Air, in. (in. Hg)</th>
<th>Horsepower</th>
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<td>(105.05)</td>
<td>(105.05)</td>
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<tr>
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<tr>
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<td>(105.05)</td>
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</table>

## DRAWBAR PERFORMANCE
(Ballsallasted at 1750 RPM)

### MAXIMUM POWER IN SELECTED GEARS

<table>
<thead>
<tr>
<th>Power (hp)</th>
<th>Drawbar pull (lbs)</th>
<th>Speed (rpm)</th>
<th>Crankshaft speed (rpm)</th>
<th>Slip (%)</th>
<th>Fuel Consumption (gpm/kW)</th>
<th>Temp T°C</th>
<th>Air, in. (in. Hg)</th>
<th>Horsepower</th>
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<tbody>
<tr>
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<td>(308.46)</td>
<td>(90.45)</td>
<td>(144.41)</td>
<td>(144.41)</td>
<td>(144.41)</td>
<td>(144.41)</td>
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<td>(144.41)</td>
<td>(144.41)</td>
<td>(144.41)</td>
<td>(144.41)</td>
</tr>
</tbody>
</table>

### Footnotes
- Power is measured in horses (hp).
- Drawbar pull is in pounds (lbs).
- Speed is given in revolutions per minute (rpm).
- Crankshaft speed is also in rpm.
- Slip percentage indicates the slip between the engine and the drawbar.
- Fuel consumption is measured in gallons per minute per kilowatt (gpm/kW).
- Temperature is given in degrees Celsius (°C).
- Air pressure is in inches of mercury (in. Hg).
- Horsepower is calculated based on the load applied to the drawbar.

### Notes
- The data is based on the engine running at 1750 RPM.
- The calculations and measurements are provided for the maximum power output in the selected gears.
### HYDRAULIC PERFORMANCE

<table>
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<th>CATEGORY</th>
<th>NA</th>
<th>Quick-action NA</th>
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<tr>
<td>Minimum force exerted through whole range (psi):</td>
<td>NA</td>
<td>NA</td>
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<tr>
<td>Standard pump</td>
<td>3947 psi (273 kPa)</td>
<td>2913 psi (201 kPa)</td>
</tr>
<tr>
<td>High-flow pump</td>
<td>4577 psi (318 kPa)</td>
<td>3912 psi (269 kPa)</td>
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<tr>
<td>Pump delivery rate at minimum pressure and rated engine speed (gpm):</td>
<td>14.2 gpm (6.8 l/min)</td>
<td>10.6 gpm (4.8 l/min)</td>
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<tr>
<td>hydraulic power (hp):</td>
<td>66.4 HP (49.3 kW)</td>
<td>57.4 HP (42.6 kW)</td>
</tr>
<tr>
<td>Force (lbf):</td>
<td>87.0 lbf (39.5 kgf)</td>
<td>75.3 lbf (33.9 kgf)</td>
</tr>
</tbody>
</table>

---

**CHALLENGER MT975B DIESEL**

Institute of Agriculture and Natural Resources  
University of Nebraska-Lincoln
2. Shear and moment forces on shafts

*Figure 25. Main Sheave Shear and Moment Diagrams*
Figure 26. Idler Sheave Shear and Moment Diagrams
3. Hydraulic machine sizing calculations (Mathcad®)

Sizing for Hydraulic Machines

**ROTATIONAL SPEED**
*These max torques occur at a speed of around 5.6 mph*

\[ \text{Speed} = 5.5 \text{ mph} \quad \text{Time}_{\text{achieve \ speed}} = 20 \text{ s} \]

**BELT SIDE**

\[ \text{Speed}_{\text{sheave}} = \frac{\text{Speed}}{10 \text{ in.}} = 92.437 \text{ rpm} \]

\[ \text{Reduction Ratio} = 14.88 \quad \text{*(Ratio of Tire to Belt (62")/20")} \]

\[ \text{Speed}_{\text{pump}} = \text{Speed}_{\text{sheave}} \cdot \text{Reduction Ratio}_{\text{1}} = 1375.465 \text{ rpm} \]

\[ \alpha_{\text{belt}} = \frac{\text{Speed}_{\text{sheave}}}{\text{Time}_{\text{achieve \ speed}}} = 0.484 \frac{\text{rad}}{\text{s}^2} \]

**TIRE SIDE**

\[ \text{Speed}_{\text{tire}} = \frac{\text{Speed}}{31 \text{ in.}} = 29.818 \text{ rpm} \]

\[ \text{Reduction Ratio}_{\text{1}} = 47.6 \]

\[ \text{Speed}_{\text{pump}} = \text{Speed}_{\text{tire}} \cdot \text{Reduction Ratio}_{\text{1}} = 1419.358 \text{ rpm} \]

\[ \alpha_{\text{wheel}} = \frac{\text{Speed}_{\text{tire}}}{\text{Time}_{\text{achieve \ speed}}} = 0.156 \frac{\text{rad}}{\text{s}^2} \]

**TORQUES**

From Challenger MT9758 - Nebraska Test Laboratory

\[ \text{Max Drawbar total} = 32089 \text{ lbf} \quad \text{NumberOfTire} = 8 \]

\[ \text{Max Drawbar} = \frac{\text{Max Drawbar total}}{\text{NumberOfTire}} = 4011.125 \text{ lbf} \]

**BELT SIDE**

\[ \text{Torque of shaft} = \text{Max Drawbar} \cdot 10 \text{ in.} = 3342.604 \text{ ft} \cdot \text{lbf} \]

\[ \text{Inertial Belt} = 771902 \text{ lbf} \cdot \text{in}^2 \]
Inertial_torque_belt = Inertia_Belt \cdot a_{belt} = 967.655 \text{ in} \cdot \text{lbf}

\text{Total Torque Wheel} = \text{Torque at shaft} + \text{Inertial_torque_belt} = 41078.005 \text{ in} \cdot \text{lbf}

\text{Torque at pump} = \frac{\text{Total Torque Wheel}}{\text{Reduction Ratio}} = 2760.079 \text{ in} \cdot \text{lbf}

\text{TIRE SIDE}

\text{Torque at shaft} = \text{MaxDrawbar} \cdot 31 \text{ in} = 10382.073 \text{ ft} \cdot \text{lbf}

\text{Inertia Wheel} = 311764 \text{ lbm} \cdot \text{in}^2

\text{Inertial_torque_wheel} = \text{Inertia Wheel} \cdot a_{wheel} = 126.081 \text{ in} \cdot \text{lbf}

\text{Total Torque Wheel} = \text{Torque at shaft} + \text{Inertial_torque_wheel} = 124470.056 \text{ in} \cdot \text{lbf}

\text{Torque at pump} = \frac{\text{Total Torque Wheel}}{\text{Reduction Ratio}} = 2514.936 \text{ in} \cdot \text{lbf}

\text{Pump/Motor Sizing}

\text{TIRE (Max torque occurs at belt)}

\text{Max Torque} = \text{Torque at pump} = 2760.079 \text{ in} \cdot \text{lbf}

\text{Rated Max Output Speed} = 3400 \text{ rpm}

\text{Working Pressure} = 380 \text{ bar}

\text{Suggested Displacement} = \frac{\text{Max Torque} \cdot 2 \cdot \pi}{\text{Working Pressure}} = 51.574 \text{ cm}^3

\text{Suggested Flow} = \frac{\text{Suggested Displacement} \cdot \text{Rated Max Output Speed}}{2 \cdot \pi} = 48.323 \text{ gpm}

\text{Speed Check}

\text{Belt Circ} = 2 \cdot \pi \cdot 10 = 62.832 \text{ in}

\text{Shaft Speed} = \frac{\text{Rated Max Output Speed}}{\text{Reduction Ratio}} = 228.405 \text{ rev} \text{ min}^{-1}

\text{Belt Limited Speed} = \frac{\text{Shaft Speed} \cdot \text{Belt Circ}}{2 \cdot \pi} = 13.595 \text{ mph}
**EQUIPMENT SPECS**

Oerlikon Fairfield Torque Hub [11HBAOB13062028]

<table>
<thead>
<tr>
<th>Performance Data</th>
<th>Continuous</th>
<th>Intermittent</th>
</tr>
</thead>
<tbody>
<tr>
<td>5500 N·m</td>
<td>11000 N·m</td>
<td></td>
</tr>
<tr>
<td>48675 in-lbs</td>
<td>97350 in-lbs</td>
<td></td>
</tr>
</tbody>
</table>
4. Tire and belt inertia calculations

**BELT DRIVE INERTIA**

\[ J_{\text{belt,drive}} = J_1 + J_2 \]

\[ J_1 = J_2 = \frac{1}{2} \left( 3 \text{ sheaves} \left(169.7 \frac{\text{lbs}}{\text{sheave}} \right) + 1 \text{ shaft} \left(209.92 \frac{\text{lbs}}{\text{shaft}} \right) \right) \times (10^2 \text{ in}) \]

\[ J_{\text{1,2}} = 35951 \text{ lbs} \times \text{in}^2 \]

\[ J_{\text{belt,drive}} = (35951 \text{ lbs} \times \text{in}^2) + (35951 \text{ lbs} \times \text{in}^2) \]

\[ J_{\text{belt,drive}} = 71900 \text{ lbs} \times \text{in}^2 \text{ or } 5990 \text{ lbs} \times \text{ft}^2 (168 \text{ kg} \times \text{m}^2) \]

**TIRE DRIVE INERTIA**

\[ J_{\text{tire,drive}} = J_{\text{tire}} + J_{\text{shaft}} \]

\[ J_{\text{shaft}} = \frac{1}{2} (209.92 \text{ lbs}) \times (2 \text{ in})^2 = 419.84 \text{ lbs} \times \text{in}^2 \]

\[ J_{\text{tire}} = \frac{1}{2} (648 \text{ lbs}) \times (31 \text{ in})^2 = 311364 \text{ lbs} \times \text{in}^2 \]

\[ J_{\text{tire,drive}} = (311364 \text{ lbs} \times \text{in}^2) + (419.84 \text{ lbs} \times \text{in}^2) \]

\[ J_{\text{tire,drive}} = 311784 \text{ lbs} \times \text{in}^2 \text{ or } 2165.17 \text{ lbs} \times \text{ft}^2 (91.24 \text{ kg} \times \text{m}^2) \]
5. Complete non-regenerative hydraulic circuit diagram

Figure 27. Complete Non-Regenerative Hydraulic Circuit Diagram
6. Tractor simulation mode, Simulink model walkthrough

From the Nebraska Test Laboratory...
- At 2000 rpm in 8\textsuperscript{th} gear and at a travel speed of 5.45 mph the MT975B’s overall gear ratio is 385.5.
- Assumed rolling radius = 2.58 ft. = 31 in.

1. \(\omega_{\text{wheel}}^{(\text{rpm})}\) was read from the tire drive...

2. \(\omega_{\text{wheel}}^{(\text{rpm})} \times 385.5 = \omega_{\text{engine}}^{(\text{rpm})}\)

3. \(\omega_{\text{engine}}^{(\text{rpm})}\) was then inputted into the 1-D lookup table (Figure 28)...

![MT975B Assumed Torque Curve](image)

\[ \text{Figure 28. MT975B Assumed Engine Torque Curve} \]

...where \(T_{\text{engine}}^{(\text{ft-lbf})}\) was outputted.

4. \(T_{\text{engine}}^{(\text{ft-lbf})} \times \left(\frac{1}{8\text{ wheels}} \times 385.5\right) = T_{\text{wheel}}^{(\text{ft-lbf})}\)

5. \(T_{\text{wheel}}^{(\text{ft-lbf})} = F_{\text{T\text{ractive}}}^{(\text{lb})} \times \text{rolling radius} \Rightarrow F_{\text{T\text{ractive}}}^{(\text{lb})}\)

6. \(F_{\text{T\text{ractive}}}^{(\text{lb})} - F_{\text{Draft}}^{(\text{lb})} = F_{\text{Net}}^{(\text{lb})}\)

7. \(F_{\text{Net}}^{(\text{lb})} = m \times \text{acceleration (lbm) \times acceleration (ft}^2) \Rightarrow \text{acceleration (ft}^2)\)

8. \(\int \text{acceleration (ft}^2) = \text{velocity (ft)}\), then inputted into the belt drive PID control.
APPENDIX B: SIMULINK® SIMULATION EXPERIMENTAL RESULTS

1. Slip control mode, regenerative circuit, model outputs

Belt Leading; Slip: 0.08, Set Speed: 13.5 mph, Wheel-load: 5,200 lb

Figure 29. Regenerative Circuit, in Slip Mode, Model Outputs
2. Slip control mode, non-regenerative circuit, model outputs

Belt Leading; Slip: 0.08, Set Speed: 13.5 mph, Wheel-load: 5,200 lb

*Figure 30. Non-Regenerative Circuit, in Slip Mode, Model Outputs*
3. Speed control mode, regenerative circuit, model outputs

Belt Leading; Slip: 0.00, Set Speed: 13.5 mph, Wheel-load: 5,200 lb

Figure 31. Regenerative Circuit, in Speed Mode, Model Outputs
4. Speed control mode, non-regenerative circuit, model outputs

Belt Leading; Slip: 0.00, Set Speed: 13.5 mph, Wheel-load: 5,200 lb

*Figure 32. Non-regenerative Circuit, in Speed Mode, Model Outputs*
5. Tractor simulation mode, regenerative circuit, model outputs

Slip: 0.00, Set Speed: 5.45 mph, 3250 lb draft, Challenger MT975B: 8th gear

Figure 33. Regenerative Circuit, in Tractor Sim Mode, Model Outputs
6. Tractor simulation mode, non-regenerative circuit, model outputs

Slip: 0.00, Set Speed: 5.45 mph, 3250 lb$_f$ draft, Challenger MT975B: 8$^{th}$ gear

Figure 34. Regenerative Circuit, in Tractor Sim Mode, Model Outputs
7. Tractor simulation mode, regenerative circuit (stalled), model outputs

Slip: 0.00, Set Speed: 5.45 mph, 5000 lb ft draft, Challenger MT975B: 8th gear

![Diagram showing model outputs for tractor simulation mode and regenerative circuit in stalled condition.](image)

*Figure 35. Regenerative Circuit, in Tractor Sim Mode (Stalled), Model Outputs*
APPENDIX C: STANDARD OPERATING PROCEDURES (SOP)

1. Belt installation and removal

Notes

A belt spear was made to help install the belts, and sheave assemblies together. This would lead to an easier way of changing out belts. However, due to too close of tolerances it was found that using a combination of the belt spear, an overhead crane, and forklift that the job could be completed with greater ease.

Figure 36. Belt Spear
PROCEDURE

Step 1:

Using the forklift. Strap an idler sheave assembly to the bottom of the forks. Make sure to push both forks together as to keep it as compact as possible. Place each idler assembly onto their bearing blocks bolted to the test stand one at a time making sure to drape the three belts over them as you go. Place the center assembly first, then work your way out.

Step 2:

Strap the main sheave assemblies to the wings of the belt spear with ratchet straps. Make sure to run the strap underneath the 4” shaft while running it through the spokes of the outside sheaves.

Step 3:

Picking up the belt spear with a telehandler, place the main sheave assemblies over the test stand itself and get the main sheave assemblies as close as possible to the bearing blocks while also draping the belts over them. Then take the overhead crane and use a sling to cradle the main sheave assemblies, running the sling through the three main sheave’s spokes. Raising the crane will relieve the ratchet straps used to secure it to the belt spear, which subsequently may now be removed. Use the crane to gently lower and place the main sheave assembly closest to the test stand’s mast into its respective bearing block.

Step 4:

All sheave assemblies should now be in place except for the drive main sheave assembly or the assembly farthest away from the test stand’s mast. Cradle the last main assembly with the overhead crane, if the belt needs to be stretched to allow the assembly to be placed,
use the forklift and another sling to pull until it can be dropped and secured into the bearing blocks.

*Step 5:*

Secure all of the shafts into their respective pillow block bearings by tightening the four bolts of the bearing. Make sure that the square side of the brass gaskets are towards the outside of the bearing when reassembling.

*Step 6:*

If the removal of the belts is being considered, proceed with these steps in reverse order.