Methods for Assessing Dynamic Performance of Shovels

MICHAEL G. LIPSETT

Equipment specifications tend to quote highest achievable performance; but this specification does not tell the whole story of machine behaviour. In the case of shovels, many criteria are used for selecting new equipment, generally related to capital and operating cost, capacity, digging forces, reach, loading rate, and propel rate. Some are competing objectives, such as high loading capacity and low energy consumption. In some circumstances, special criteria become important, such as low ground bearing pressure and operating temperature range. Access to spare parts and after-sales service may also be a consideration if the machine is intended for primary production and must have high availability. Energy consumption and emissions are increasingly important in some jurisdictions. This paper describes a simple framework for assessing different shovel designs, including dynamic performance and emissions. Examples are given for face shovels for surface mining excavation.

Keywords: mining shovels, excavators, performance, system identification

1. Introduction

Mobile mining equipment specifications tend to quote highest achievable performance. While this approach is not incorrect, such information does not tell the whole story of machine behaviour. In the case of face shovel for surface mining excavation, many criteria are used for selecting new equipment: technical specifications, economic factors, safety features, & environmental impacts.

Some are competing criteria: power for one task affects available power for other tasks. In operating conditions where there is limited power, an analysis of the task dynamics can be used to assess the achievable performance of the machine.

In the case of mining shovels, the task specification at the highest level is quite simple. A shovel has a modest number of tasks to perform. It digs dirt, and it travels to different locations to dig dirt.

The type of power plant affects the equipment design. Hydraulic machines have actuators on the shovel attachment. Electric cable shovels have most of the motors and transmission elements on the revolving frame, with cables transmitting power for the hoisting motion.

It is important to ensure compatibility with other systems, for example, matching bucket capacity to haul truck size so that a shovel does not have to take a partial bucket to top up a truck.

1.1 Critical Specifications and Design Considerations

The specifications for a shovel must be clear enough to allow the selection of a machine that will deliver the required performance at an acceptable cost with high physical availability throughout the useful life of the machine. The operating performance specifications include not only fragmentation and loading but also equipment relocation. Other specifications include production rate, capital cost, production cost per hour of operation (including maintenance), mechanical availability vs. expected utilization, payload per bucket (matched to an integer number of loads to fill a truck), and energy usage (kW/ton or litres/ton). Travel speed affects mine design and short-range planning, if the machine has to relocate often to different locations in a mine.

Specifications that affect physical availability include periodic servicing for fuel (or power cable relocation), and replacement of consumables such as teeth, as well as maintenance outages. There may be other costs for infrastructure, such as support equipment for clean-up or ripping frozen top layers of a bench in winter operation.

Key design considerations for an excavating shovel to meet the performance and reliability specifications are based primarily on kinematics. The kinematics of the shovel attachment determines the maximum reachable height, and the maximum reach to the bank to do work (which determines the maximum bench height in mine design and production planning). Geometric and inertial parameters of the shovel

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Corresponding author: Dr. M.G. Lipsett, Department of Mechanical Engineering, University of Alberta, Edmonton, T6G-2G8, Phone: (780) 492-9494, Fax: (780) 492-2200, Email: michael.lipsett@ualberta.ca
mechanism plus actuator forces and masses of structural components yield the static forces that the machine can deliver as penetrating or cutting force at different shovel poses, break-out force, or a combination. Kinematic mechanism analysis can be used to calculate these specifications.

Dynamic performance is the force that can be delivered at the ground-engaging tool when the machine is moving at speed. This is more important for machines that have to rely on accelerating into the soil in order to achieve the desired depth of cut. Most machines in service do not depend on such a motion, for two reasons: the motion is difficult to control, and the impact can cause damage to the machine. Analysis of the machine dynamics requires knowledge of nongeometric parameters such as friction in joints, the dynamic behaviour of the actuators and transmission, and dynamic behaviour of interaction with the ground (soil-tool interaction, which entails earthmoving models and knowledge of soil parameters, and, for soft ground, the dynamics of soft foundations). Performance is usually summarized as digging cycle time and maximum mass of material moved per unit time, e.g. tons per hour, based on computer-aided design and performance testing of prototype machines.

Other design considerations that influence dynamic performance indirectly are the choice of structural materials (typically steel for cost), the type of actuators and transmission elements, the control system, even cab location and layout is important, as the viewable workspace can affect operator awareness of the task and potential obstacles in the work area, such as support equipment.

The effectiveness of a shovel’s mechanical performance depends on three things:

- the power available to actuate the shovel;
- the power required to penetrate the ore face, break out the material, and swing; and
- the efficiency with which power is converted during operation.

The focus of this paper is the efficiency of hydraulic shovel attachments and metrics for assessment. The Terex O&K RH 400 is analyzed as a case study; reachable workspace, mobility during digging (also called manipulability), and achievable cutting forces are presented with some simplifying assumptions for the dynamics of the machine. Methods for determining the parameters of the models are discussed.

2. Shovel Attachment Efficiency

2.1 Model-Based Analysis

Forward kinematics gives the location of the orientation of the bucket lip as a vector $X$, as a function of the displacements of the joints of the shovel attachment $\theta$:

$$X = f(\theta).$$  \hspace{1cm} (1)

Inverse kinematics solves the joint displacements necessary to achieve a desired bucket position and orientation, if the pose is possible:

$$\theta = f^{-1}(X).$$  \hspace{1cm} (2)

Rigid-body dynamic models of excavators have been formulated for a number of machine types, particularly backhoes [1,2,3]. Dynamic machine models require estimates of the kinematic parameters and inertial parameters of the machine links, as well as constitutive relationships for actuators, friction in joints, etc. [4]. Furthermore, the model requires the joint displacements, velocities, and accelerations to calculate the forces and torques that move the mechanism.

While kinematic parameters such as the distance between joints on a link are easily measured, other parameters such as the moment of inertia of a massive shovel structure such as the boom can be difficult to estimate accurately. It is difficult to measure the joint motions of a working machine. For these reasons, using the full equations of motion is impractical in most operating situations.

Rather than identify the complete rigid-body dynamics equations of motion, it is sufficient for design purposes to consider a partial model of forces on the shovel attachment. The actuators overcome the soil resistance to cutting $F_c$ at the bucket tip, where $F_c$ is the vector of cutting and breakout forces. The actuators also oppose the gravity load of the mechanism and payload, resist friction, and accelerate the machine to change its motion. Superposition of actuator torques for each load yields the total torque $\tau_i$ at joint $i$:

$$\tau_i = \tau_{gi} + \tau_{ai} + \tau_{fi} + \tau_{ci},$$  \hspace{1cm} (3)
where \( \tau_g \) is gravity torque, \( \tau_a \) is acceleration torque, \( \tau_f \) is friction torque, and \( \tau_c \) is cutting torque. (Here the term torque is used to describe the effort produced by an actuator. In some cases, such as a hydraulic cylinder, an actuator actually produces a linear force.)

In a quasi-static scenario, the acceleration of the machine is ignored. This is often a reasonable assumption, because most of the torque available to drive a joint is taken up by the gravity load and friction forces. Static force-balance analysis is used to find the vector of torques \( \tau_g \) to support the gravity load. Friction torque can be estimated as a linear function of joint velocity, with zero friction when there is no motion. Given a particular shovel pose - that is, a set of joint displacements that articulates the shovel attachment in a particular position – the actuator torque vector for cutting \( \tau_c \) is

\[
\tau_c = \tau - \tau_g .
\]

and the achievable cutting force vector \( F_c \) is

\[
F_c = [J^T]^{-1} \tau_c ,
\]

where \( J \) is the Jacobian matrix describing the relationship between joint motions and motions of the bucket. Cutting forces for hydraulic shovels are calculated in the region of the workspace that is reachable at a horizontal angle of attack. While it is not necessary to penetrate the ore face horizontally, this is the preferred approach to maximize the volume of bucket that is filled when the bucket is pulled out of the bank.

2.2 Manipulability

Manipulability is a measure of ease of motion, that is, how effectively can a machine arbitrarily change the position and orientation of the bucket at the end of the shovel attachment, or alternatively how effectively can the machine exert a force onto its environment. Manipulability is based on the eigenvalues of the Jacobian solved at a given set of joint angles [5].

To illustrate the concept, consider a two-link planar serial mechanism with revolute joints, such as a simple robot arm [5]. The forward kinematics for the position of the end of the mechanism is

\[
x = l_1 \cos \theta_1 + l_2 \cos(\theta_1 + \theta_2) \\
y = l_1 \sin \theta_1 + l_2 \sin(\theta_1 + \theta_2) .
\]

(6)

The Jacobian of this two-degree-of-freedom mechanism is a two-by-two matrix that relates the joint velocities to the velocity of the end of the mechanism:

\[
J = \begin{bmatrix}
\frac{\partial x}{\partial \theta_1} & \frac{\partial x}{\partial \theta_2} \\
\frac{\partial y}{\partial \theta_1} & \frac{\partial y}{\partial \theta_2}
\end{bmatrix} ,
\]

(7)

which, for the two-link mechanism, is

\[
J = \begin{bmatrix}
-l_1 \sin \theta_1 - l_2 \sin(\theta_1 + \theta_2) & -l_2 \sin(\theta_1 + \theta_2) \\
l_1 \cos \theta_1 + l_2 \cos(\theta_1 + \theta_2) & l_2 \cos(\theta_1 + \theta_2)
\end{bmatrix}
\]

(8)

The eigenvectors of \( J \) can be visualized as the axes of an ellipsoid that represents how readily a unit motion of the joints translates into motion of the end of the mechanism, and so it is a sensitivity analysis. Figure 1 illustrates the manipulability ellipsoid for the two-link mechanism.
Manipulability measures at a particular pose of the machine may include the volume of the ellipsoid (or area in the case of two dimensions), the minimum eigenvalue, and the ratio of the maximum eigenvalue to the minimum eigenvalue. When there are three degrees of freedom for moving in a plane (as is the case of a shovel attachment, which has boom, stick, and curl resulting in planar movement), then manipulability is assessed from the square roots of the eigenvalues of the matrix product $J^T J$. Eigenvalues and corresponding eigenvectors can be found using singular value decomposition, a standard matrix operation that can be done numerically [6].

### 2.2 Manipulability force ellipsoid and dynamic manipulability

The force manipulability is the complement of the kinematic manipulability, which means that a direction with good motion responsiveness has poor force generation capability. When machine dynamics are fully known, then a dynamic manipulability assessment can be made from the equations of motion [5], which include terms for mass, centres of gravity, moments of inertia, and friction. While inertial terms can be estimated for a rigid-body dynamic model [4,7], in the case of shovels, when accelerations are fairly low and there is uncertainty in the inertial parameters, a quasi-static formulation based on kinematic manipulability is simpler and yet sufficient for assessing machine performance.

### 3. Case Study: O&K RH 400

The RH series of excavators differs from most other front shovels, in that it employs an additional mechanism. The TriPower is an intermediate cam connection on the boom designed to generate a nearly horizontal bucket trajectory when the stick cylinders are stroking with boom hydraulics released and curl cylinders locked. This feature keeps the bucket angle of attack nearly horizontal as the shovel digs forward, and allows additional flow of hydraulic fluid to the stick cylinders at system pressure. In this way, more power can be directed into penetrating the soil than a comparable machine that has to actuate multiple cylinders to push horizontally [8].

#### 3.1 Forward Kinematics

Forward kinematics yields the location and orientation of the bucket lip, given the three joint displacements. Figure 2 shows the kinematic parameters of the RH 400 shovel attachment. Inverse kinematics produces the joint displacements for a given shovel pose. A skilled operator can achieve the desired bucket motion with good angle of attack, provided that visibility is good. But as face shovels increase in size, it becomes increasingly difficult for the operator to see whether the bucket is digging exactly where the bucket lip is. New operators require extensive training with senior operators to develop this skill. The TriPower mechanism simplifies the operator action (but complicates the inverse kinematics calculations compared to standard hydraulic shovels that have direct actuation of each joint).
Static kinematic parameters for the RH 400 shown in Figure 2 are $A$, $B$, $D$, $E$, $EL$, $F$, $G$, $H$, $J$, $L_o$, $T_1$, $T_2$, $T_3$, $T_4$, $T_5$, $T_6$, $M$, $N$, $Q$, $B_k$, $B_{kt}$, $\phi_{bo}$, $\phi_{bkt}$. Calculated constants are:

$$
\mu = \arccos \left( \frac{T_1}{2} + \frac{T_4}{2} - \frac{T_5}{2} \right) / (2T_1T_4),
$$

$$
\eta = \arccos \left( \frac{T_1}{2} + \frac{T_5}{2} - \frac{T_4}{2} \right) / (2T_1T_5),
$$

$$
\phi_f = \arctan \left( \frac{F}{G} \right),
$$

$$
F_g = \sqrt{F^2 + G^2},
$$

$$
\phi_l = \arccos \left( \frac{T_5}{2} + \frac{T_6}{2} - \frac{T_3}{2} \right) / (2T_5T_6).
$$

Figure 2 - RH 400 Kinematics

Measured variables are the joint angles: the boom angle $\theta_b$, the length of the hydraulic cylinder to actuate the stick $L_s$, and the actuator for the curl cylinder $L_c$. In these expressions, angles are given in radians. The following expressions yield the stick angle $\theta_2$:

$$
\theta_2 = \theta_1 + \beta,
$$

$$
\kappa = \theta_2 + \phi_f,
$$

$$
C^2 = F_g^2 + B^2 - 2F_gB \cos(\kappa),
$$

$$
\delta = \arccos \left( \frac{L_t^2 + C^2 - T_3^2}{2L_tC} \right),
$$

$$
\Delta = \arccos \left( \frac{F_g^2 + C^2 - B^2}{2F_gC} \right),
$$

$$
\Phi_s = \pi + \delta - \Delta,
$$

$$
\phi_1 = \Phi_s - \phi_f,
$$

$$
\phi_{l4} = \arctan \left( \frac{L_4 \sin(\phi_1) - B \sin(\theta_b)}{L_4 \cos(\phi_1) - B \sin(\theta_b) + G} \right),
$$

$$
\phi_{l1} = \phi_{l4} - \mu,
$$

$$
\theta_s = \phi_{l1} - \theta_b - \eta.
$$
\[
\theta_1 = \arccos\left[ \frac{R_1^2 + M^2 - L_1^2}{(2R_1 M)} \right], \quad \theta_2 = \theta_1 + \theta_r.
\]

Having solved the relative angle between boom and stick, the next set of expressions is used to find the bucket curl angle:

\[
U_x = A \cos(\theta_1) - B \cos(\theta_b) - T_s \cos(\theta_b + \theta_r + \phi_s + \pi), \quad \phi_s = \arctan(U_x/|U_x|),
\]

\[
U_y = A \sin(\theta_1) - B \sin(\theta_b) - T_s \sin(\theta_b + \theta_r + \phi_s + \pi),
\]

\[
U = \sqrt{U_x^2 + U_y^2},
\]

\[
m = \sqrt{U_x^2 + U_y^2 - 2NU \cos(\theta_1 + \gamma - \phi_s - \theta_1)},
\]

\[
\Phi_\mu = \arccos\left[ \frac{m^2 + N^2 - U_x^2}{(2mN)} \right] + \arccos\left[ \frac{m^2 + Q^2 - L_c^2}{(2mQ)} \right],
\]

\[
\phi_b = \Phi_\mu + \theta_1 + \theta_2 + \gamma;
\]

\[
\theta_{\text{tooth}} = \phi_b + \phi_{\text{bu}}.
\]

The manipulability of the machine at that pose with curl cylinders locked is given by:

\[
J = \begin{bmatrix}
\frac{\partial \phi_1}{\partial \theta_1} & \frac{\partial \phi_1}{\partial \theta_2} \\
\frac{\partial \phi_2}{\partial \theta_1} & \frac{\partial \phi_2}{\partial \theta_2} 
\end{bmatrix},
\]

where

\[
\frac{\partial \phi_1}{\partial \theta_1} = -A \sin(\theta_1) - N \sin(\theta_1 + \pi + \theta_2 + \gamma) - B_s \sin(\Phi_\mu + \theta_1 + \theta_2 + \gamma + \phi_{\text{bu}}),
\]

\[
\frac{\partial \phi_1}{\partial \theta_2} = -N \sin(\theta_1 + \pi + \theta_2 + \gamma) - B_s \sin(\Phi_\mu + \theta_1 + \theta_2 + \gamma + \phi_{\text{bu}}),
\]

\[
\frac{\partial \phi_2}{\partial \theta_1} = A \cos(\theta_1) + N \cos(\theta_1 + \pi + \theta_2 + \gamma) + B_s \cos(\Phi_\mu + \theta_1 + \theta_2 + \gamma + \phi_{\text{bu}}),
\]

\[
\frac{\partial \phi_2}{\partial \theta_2} = N \cos(\theta_1 + \pi + \theta_2 + \gamma) + B_s \cos(\Phi_\mu + \theta_1 + \theta_2 + \gamma + \phi_{\text{bu}}),
\]

with small changes in \( \Phi_\mu \) neglected to simplify the solution. The derivatives can also be approximated numerically. Fortunately, for machine types that do not have an intermediate linkage (TriPower), the analysis is considerably simpler.

From these expressions, differential motions and cutting forces can be calculated in the region of the workspace reachable at a horizontal angle of attack. While it is not necessary to penetrate the ore face horizontally, this is the preferred approach to maximize the volume of bucket filled, after which the bucket may be tilted to prevent the charge of ore or soil from spilling.

For the manipulability analysis, static kinematic and dynamic parameters were found from supplier information or estimated.

The method first establishes the horizontal reachable workspace. The workspace is divided with uniform spacing of the boom and stick joints. At each location of the proximal joint of the bucket, the algorithm attempts to position the bucket horizontally. If the range of curl motion is insufficient to pose horizontally, then the angle closest to horizontal is returned. The tooth location is calculated for the subset of points where horizontal angle of attack is achievable. The software package Matlab [9] was used to perform the
calculations and to plot the results.

Figure 3 shows the reachable workspace of the RH 400 with the bucket horizontal, and a set of ellipses. These ellipses show how a small set of motions of the joints would result in a motion of the bucket teeth. The major axis of an ellipse indicates the most responsive direction; and each bucket location has a different dominant direction. In the middle of the workspace the machine can move easily in any direction in the plane, as shown by the almost circular manipulability ellipsoid.

![Figure 3 - Kinematic Manipulability of the RH 400 with Horizontal Bucket (Distances in mm)](image)

In this case, the ability to generate force horizontally at the outer limit of the reachable horizontal workspace is quite good, with the compromise of not being able to push horizontally very fast. In the middle of the workspace, the ability to generate bucket tip forces is good in all directions.

4. Field Performance Assessment

Operations can track the number of bank cubic metres produced by a given shovel through tonnage recorded by the payload monitoring system on each haul truck. This information is logged through the dispatch system, to give a coarse estimate of average bucket payload by assuming proper three-pass or four-pass loading. The system also shows how long it takes to fill a truck, but it cannot tell where any lags appear. Cycle time is timed by observation and a stopwatch. Utilization can be recorded well through a dispatch system provided that operators code the machine status accurately and in a timely manner. Excavator payload is a mining production parameter that is difficult to measure. In surface mines, tonnage is tracked either by payload monitoring on haul trucks, or weightometers on conveyors. Production tonnage is often estimated from truck payload monitoring systems, which are difficult and expensive to maintain accurately.

4.1 On-Board Monitoring of Field Performance

Shovel performance can also be measured on the machine itself. The advantage of on-board monitoring is the opportunity to learn more about the operation, and to give that information to the operator so that he or she can work the machine more effectively. A payload monitor on a shovel would require fewer units than truck-based monitors, with the advantage of giving the shovel operator immediate feedback on bucket fill and required cutting force. Estimates of cutting force would be useful for validating numerical models for next-generation bucket design engineering tools.
There are two basic approaches to estimating payload and cutting forces: instrument the bucket and assume that motion errors are small, or instrument the machine and calibrate its measurements for standard motions.

Monitoring the bucket location and angle is challenging, because sensors must be very rugged. Linear string potentiometers are adequate for field tests but not for production monitoring. These sensors measure joint displacement at two of the actuating cylinders, here designated $L_s$ for the stick and $L_c$ for curl. Since boom cylinder length $L_b$ is a function of $\theta_1$, a polynomial approximation can be used to evaluate $\theta_1$ if the length of $L_b$ is measured instead of the joint angle. Alternatively, internal sensors can be used in hydraulic cylinders, although they are expensive; or resolvers can be used directly at the pin joints. Absolute angle of a structural link can also be estimated using inclinometers with piezoresistive accelerometers.

Direct methods for estimating the mass in an excavator bucket are based on parameter estimation. Two model types are used. A quasi-static method assumes that machine accelerations can be neglected as disturbances. A quasi-static method estimates mass and centre-of-gravity of material in the links of the shovel attachment. For simplicity, minor links associated with actuator cylinders may be neglected, with their mass lumped divided between the links to which they connect. Measurement transients can be filtered to compensate for effects such as pressure fluctuations due to acceleration of shovel attachment because of rigid-body vibration of machine by rocking of the tracks on compliant ground. Cylinder forces are more easily based on pressures in the cylinders themselves rather than making direct measurements using strain gauges. On a testbed backhoe excavator measurements are taken of pressures in hydraulic cylinders, and joint angles for the shovel attachment pose, which are then used to estimate bucket payload (mass and centre of gravity) using a least-squares estimate of parameters in quasi-static equations of rigid-body motion. This method is subject to errors from actuator dynamics, ground softness, and changes in inertia tensors.

A dynamic method may be more accurate, but it requires a validated rigid-body dynamical model of the machine, with accurate inertial parameters and friction constitutive relationships at joints, and using measurements of joint angles, velocities, acceleration, and estimates of external forces on the shovel attachment links using electroresistive strain gauges. Recursive methods [7] allow the model to be updated automatically, neglects minor linkages, and uses pressure measurements at each end of cylinder to estimate cylinder force (including a model of cylinder friction). Any unmodeled dynamics of hydraulic system are considered to be minor sources of error.

5. Conclusions

Performance assessment of shovels can be done at the design stage using model-based analysis. Sensitivity analysis techniques such as manipulability can lead to insights into machine behaviour over the operating workspace. Field measurements of dynamic performance can be done on the machine, but care must be taken to use rugged sensors and data acquisition systems. The current generation of payload monitoring systems are designed in this way. Other performance metrics such as average production rate will remain valuable indicators of performance, especially for machines that can not be instrumented.

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7. References


