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ABSTRACT

Ground excavation constitutes a significant component of production costs in any surface mining operation. The excavation process entails material digging and removal in which the equipment motion is constrained by the workspace geometry. A major excavation problem is the variability of material properties, resulting in varying mechanical energy input and stress loading of shovel dipper-and-tooth assembly across the working bench. This variability has a huge impact on the shovel dipper and tooth assembly in hard formations.

With this in mind, the primary objectives of the project were to (i) provide the theoretical basis to develop the Intelligent Shovel Excavation (ISE) technology to solve the problems associated with excavation in material formations; (ii) advance knowledge and frontiers in shovel excavation through intelligent navigation; and (iii) submit proposal for the design, development and implementation of the ISE technology for shovel excavation at experimental surface mining sites.

The mathematical methods were used to (i) develop shovel's kinematics and dynamics, and (ii) establish the relationship between shovel parameters and the resistive forces from the material formation during excavation process.

The ADAMS simulation environment was used to develop the hydraulic and cable shovel virtual prototypes. Two numerical examples are included to test the theoretical hypotheses and the obtained results are discussed.

The area of sensor technology was studied. Application of specific wrist-mounted sensors to characterize the material, bucket and frame assembly was determined. Data acquisition, display and control system for shovel loading technology was adopted. The concept of data acquisition and control system was designed and a shovel boom stresses were simulated.

A multi-partner collaboration between research organizations, shovel manufacturer, hardware and sensor technology companies, and surface mining companies is proposed to test design features, construct a field ready prototype and perform field testing at the surface mining sites. It is anticipated that 10 % in energy savings including electricity (cable shovel) and fuel (hydraulic shovel) will be achieved by implementation of ISE technology.

The research project on Novel Excavation Technologies for Efficient and Economic Surface Mining was sponsored by the DOE – Mining Industry of the Future (Award Number: DE-FG26-03NT41929). The project was completed by the researchers from the Pennsylvania State University (PSU), University of Alberta (UOA), and University of Missouri-Rolla (UMR) over the 18-month period.

TABLE OF CONTENTS

Abstract	3
Executive Summary	5
Experimental	6
Results and Discussion	8
Conclusions	27
References	28
Appendix #1	31
Appendix #2	37

EXECUTIVE SUMMARY

The key accomplishments of this research project include characterization of shovel-formation interaction, development of kinematics and dynamic of cable and hydraulic shovels, development of cable and hydraulic shovel simulation models, and determination of sensor technology, data acquisition, display and control system for shovel loading technology.

Theoretical models for cable and hydraulic shovels, and numerical model of shovel-formation interaction were developed. An analysis of the shovel excavators was performed applying the Newton-Euler equations and Kane's method. Using exponential formulation, a framework for kinematics modeling and analysis was carried out. Based on the kinematics results, dynamic equations of shovel were developed. The shovel bucket and its interactions with hoist and crowd forces were also modeled and studied thoroughly to understand the active forces in different positions, times and operating planes. A detailed mathematical modeling of cable and hydraulic shovels, and their interaction with material formations is given in Appendices 1 and 2.

The shovel virtual prototypes were designed in ADAMS simulation environment, and three-dimensional (3D) cable and hydraulic solid models were developed. Examples are included to illustrate the application of the models in surface mining applications. These models can be used to simulate the performance of shovels during digging operation, including energy consumptions under different digging profiles and the efficient profile for energy saving. These models are critical for the evaluation of the performance of shovels during excavation and its optimization of digging profiles. The main novelty of this study is the use of new modeling environment to simulate the performance of shovels based on given digging paths and to define efficient digging profiles based on energy consumption. Once the shovel dynamic environment is defined and modeled, it allows fast, efficient and accurate dynamic simulation for optimized shovel operating profiling.

In the next phase of the project, the area of sensor technology was studied. Application of specific wrist-mounted sensors to characterize the material, bucket and frame assembly was determined. It is concluded that position sensor and pressure transducer can be configured and applied for shovel technology. The concept of data acquisition and control system was designed and stresses of shovel boom were simulated.

The Principal Investigators propose design, development and implementation of the novel technology for shovel excavation through the DOE Mining Industry of the Future: Area of Interest – Energy Efficient Alternatives to Current Technologies in Extraction. We would test design features, construct a field ready prototype and perform field testing at the surface mining sites. Multi-partner collaboration would include research organization, shovel manufacturer, hardware and sensor technology companies, and surface mining companies. The Pennsylvania State University and University of Missouri Rolla will invite P&H Mining (shovel manufacturer), Thunderbird Mining Systems (shovel monitors, hardware and software), Admotec, Inc. and Intertechnology,

Inc. (sensor technology), and North American Coal Corporation and Hanson Aggregates (industry sites) to participate in the project.

It is anticipated that 10 % in energy savings including electricity (cable shovel) and fuel (hydraulic shovel) will be achieved by implementation of ISE technology. The energy savings are due to increased job efficiency where real-time information and precise knowledge of the material properties provide on-the-job decisions by shovel operators and mine production engineers to guide shovel excavation. The ISE technology assists operators to navigate shovel excavation based on the established correlation between material properties and dipper excavation.

Two manuscripts, as a direct result of this research, will be published in refereed journals including SME Transactions, by Society for Mining, Metallurgy, and Exploration, and AusIMM Transaction – Section A: Mining Technology, by the Australasian Institute of Mining and Metallurgy.

EXPERIMENTAL

The first step in this project was to develop shovel's kinematics and dynamics, and to establish the relationship between shovel parameters and the resistive forces from the material formation during excavation process.

Prior to this research project, there has been some research on the relationship between hydraulic forces or other performance parameters of hydraulic shovel excavators and the resistive forces from the environments during excavation process. Most of the work treats shovel excavators in a form similar to the equations of motion of robotic manipulators and develops the kinematics models first based on the geometrical relations among the links (the boom, the stick and the bucket). This will be followed by a dynamic analysis of the shovel excavator either by applying the Newton-Euler equations to each link in succession or by Lagrange equations to the entire system, which results in a set of differential equations. The set of generally nonlinear equations governs the relationship among the cylinder forces, joint forces between the links, resistive forces from environments, and geometrical and kinematics parameters (Araya et al, 1988, Shishaev and Mochalov, 1989, Koivo, 1994, Koivo et al, 1996, Frimpong, Hu and Szymanski, 2002, Frimpong, Hu and Chang, 2003). If the resistive forces from the environments are given, then the hydraulic forces inside the cylinders and interactive forces at the joints can be determined for shovel excavators. This modeling philosophy is general and popular, but the associated nonlinear differential equations are difficult to solve, especially with the inclusion of the joint frictions and hydraulic fluid properties. Scaled or full-scale shovel excavators have also been used to measure directly the resistive forces experienced during the excavation and the hydraulic forces inside the cylinders using sensors embedded inside the shovel components and the formation or muck pile (Imanishi et al, 1987, Luengo et al, 1998). Experimental investigations have the advantage of simulating excavation operations in real-time and realistically, however, they are expensive and time-consuming. Caterpillar and Komatsu have also advanced shovel navigation systems for monitoring bucket position,

accurate load in bucket and accurate mill and waste dump destinations (Wadell and Maier, 1998). These research and technology initiatives have assisted mine production with accurate information on bucket location, and load monitoring systems. However, they do not have means of measuring stress wave propagation through the shovel linkages to provide real-time excavation information in primary digging conditions, which is critical toward the solution of variable hard digging and quality problems.

Cable shovels are widely used in surface mining operations for materials excavation and loading of dump trucks. They have larger capacities and breakout forces for excavation and loading than hydraulic shovels. Research on machine kinematics and dynamics is a key to understanding and improving their operating performance, as outlined by previous researchers (Murray et al, 1993; Craig, 1986; Tafazoli et al, 1999; Daneshmend et al, 1993). Daneshmend et al. (1993) applied the Newton-Euler method to build a cable shovel simulation model. Tafazoli et al. (1999) developed a method to identify inertial parameters for excavator arms. Craig (1986) introduced a basic method to describe the position and orientation of components and to analyze the kinematics of the associated mechanism. Murray et al (1993) formulated the exponential method for robotic motion. This method allows for a comprehensive description and analysis of the kinematic configuration in a global frame system. Kane and Levinson (1985) developed the Kane method and explained its advantages over other methods for dynamic analysis. Other researchers have used this method to analyze the dynamics of different systems such as automobiles, robotic and production machinery. Through the Kane's method, a reasonable number of equations can be obtained to capture the system dynamics while avoiding many derivative processes. In addition, this method can also deal with non-holonomic constrained problems. Frimpong, Hu and Szymanski (2002) and Frimpong, Hu and Chang (2003) have advanced cable shovel dynamics to simulate the shovel boom-dipper-teeth interactions with in-situ formation and muck-pile.

Analysis of the shovel excavators was performed applying the Newton-Euler equations and Kane's method. With the advent of computer applications, a new technology called virtual prototyping is receiving wide application in various industries. It can simulate realistically the full-motion behavior of complex mechanical systems and provide quick analysis for multiple design variations toward an optimal design. This reduces the number of costly physical prototypes, improves design quality, and dramatically reduces product development time. Additionally, the solutions to above mentioned complex equations and the corresponding analysis can easily be carried out in a virtual simulation environment.

A hydraulic and cable shovels were modeled using ADAMS simulation environment. Only the front-end assembly including handle and bucket of the cable shovel is modeled because this research is only concerned with the digging section of a duty cycle. Therefore, the rotation of the upper structure is not modeled and the handle and bucket assembly only moves on the vertical plane. Also, no joint friction between the hoist cable and the sheave and between the handle and the crowd pivot is considered.

In the next phase of the project, the area of sensor technology was studied in great detail. Successful excavation process control requires sensor feedback from the excavation tool to precisely determine the formation diggability and quality. Thus, the excavator needs a non-conventional control approach that uses force/torque feedback to deal with uncertainties in the real environment. The dynamic and unstructured mining conditions dictate a control architecture that can combine data from multiple sensors to evaluate and monitor the environment. They must also perform high-level reasoning for task planning, and execute and control real-time machine components and actions. In addition to force/torque sensors, multi-sensors or hybrid sensory system was studied.

RESULTS AND DISCUSSION

The basic components of a hydraulic shovel established through this project are shown in Figure 1. The hydraulic shovel consists of an upper structure, an undercarriage, and a three-linkage assembly, i.e., the boom, sticks, and bucket assembly. The three-linkage assembly is housed in the upper structure, which is supported by the undercarriage. The boom is connected to the upper structure with pin joints having horizontal rotation. The boom, stick and bucket assembly is also pi-jointed and operated with the use of hydraulic cylinders. The upper structure and the assembly together can swing against the undercarriage. The whole hydraulic shovel can propel forward or backward. The attached coordinate frames indicate the possible motions of the components of the hydraulic shovel. Figure 2 shows a free body diagram of a stick during excavation, while Figure 3 shows bucket-formation interaction. A detailed mathematical modeling of hydraulic shovel is given in Appendix #1.

Figure 1. Hydraulic Shovel Structural Components

Figure 2. Free Body Diagram of the Stick

Figure 3. Bucket-Formation Interaction

In order to study the effect of different parameters on the performance of a hydraulic shovel, we developed a parameterized virtual model in ADAMS simulation environment. By examining the effect of different parameters and different values for each parameter, we can find the optimized combination for the hydraulic shovel performance.

Three different initial conditions are considered to start and run the simulator. They are:

- a. *Force conditions:* The hydraulic force inputs to the cylinders and the digging force to the tip of the bucket are applied and the time response of the system is then obtained by measuring the displacement, velocity and acceleration of each joint and the extension displacement, extension velocity and extension acceleration of each cylinder.
- b. *Displacement condition:* A digging trajectory at the bucket tip is prescribed as input. The model solves for the required displacement input for each cylinder to achieve that trajectory. It also output other parameters such as joint angles, angular velocity and acceleration.
- c. *Mixed condition:* The extension displacement of each cylinder combined with a prescribed resistive force at the bucket tip is prescribed as the input. With the time dependence of the extension displacement of each cylinder and the resistive force known, the hydraulic force inside each cylinder and joint force between two links can be evaluated.

Figure 4 shows a hydraulic shovel simulator developed in ADAMS environment. Figure 5 shows a virtual prototype, while figure 6 shows 3D solid hydraulic shovel model. Only the front-end assembly including boom, stick and bucket of the hydraulic shovel is modeled for the reason described in the previous section. Other assumptions are (i) there is no joint friction between the upper structure and the assembly and between the links within the assembly; and (ii) the hydraulic cylinders are ideal, i.e., no frictional losses.

Figure 4. A Hydraulic Shovel in ADAMS Simulation Environment

Figure 5. A Mechanical Simulation System – Virtual Prototype (backhoe)

Figure 6. 3D Solid Hydraulic Shovel Model

An example is include to illustrate the performance of hydraulic shovel based on given digging paths and to define efficient digging profiles based on energy consumption.

The kinematics and dynamic simulations of hydraulic shovel are illustrated by prescribing an assumed trajectory at the bucket tip when excavating a formation or muck-pile (as illustrated in Figure 7). The formation or muck-pile has a slope of 50 degrees. The trajectory is designed such that at the end of its execution, the volume of the cut material is equal to the bucket capacity of the shovel.

Figure 7. A Trajectory for the Bucket tip of a Hydraulic Shovel

The bucket tip moves at constant speed and it takes 7.5 seconds to execute the entire trajectory. The main geometry data for the shovel simulated is listed in Table 1, while the parameters for determining the resistive force are shown in Table 2.

Table 1. Main Hydraulic Shovel Data

	Length ¹ (m)	Mass (kg)	Inertia Moments ² ($kg \cdot m^2$)
Boom	7.682	36420	1.850E+005
Stick	5.334	21310	4.810E+004
Bucket	3.950	40800	4.567E+004
Bucket capacity 5.625			

¹ Length between two joints for boom and stick and between joint and bucket tip for bucket

² Moment of inertia about the gravitational center.

Table 2: Data for Excavated Material Properties (Koivo et al, 1996)

k_p	1.005	$\mathcal{E}$	$55,000 \text{ kg}/(\text{m}^2/\text{s}^2)$
k_s	5,500	γ_{soil}	$1,921.8 \text{ kg}/\text{m}^3$
μ	0.1	b	4.8m
N	$1 \text{ kg} \cdot \text{m}/\text{s}^2$		

k_p and k_s are specific resistances in digging material; constants b is width of cut slice of the material, μ is the friction coefficient of the bucket and the cutting material; N is the pressure force of the bucket with the cutting material; and ε is the coefficient of resistance experienced in filling the bucket during the movement of the prism of soil.

Given the geometrical and physical parameters for the shovel and the cutting material, the evolution of the joint angles and cylinder forces with time are shown in Figures 8 and 9 based on the given trajectory curve. Figure 9 shows that the cylinders experience three phases during the digging part: a gradual increase, reaching maximum, and gradual decrease. The forces reaches the maximum around the middle of the trajectory.

Figure 8. Joint Angles with Time

Figure 9. Cylinder Forces with Time

It is often required in surface mining industry that the duty cycle is shortened in order to improve production efficiency. This is particularly important for large-scale operations

where shovels are overtrucked. Even reducing the execution time of a single cycle of an operation by a few seconds can translate into a large savings over the entire job. Here the same trajectory as shown in Figure 7 is finished within three different digging times: 5, 7.5 and 10 seconds. The power consumption for the three cases are simulated as shown in Figure 10. It can be seen that shortening the digging time consumes more energy. The peak power required almost doubles when the digging time reduces half. Therefore, there is a need for a trade-off between energy consumption and saving duty cycle. This is very important, because energy consumption is directly related to the forces exerted by the front-end components (boom, stick and bucket), and more severe working environments for these parts are expected with shortened duty cycle. Note the horizontal axis in Figure 10 is non-dimensionized for comparative purpose.

Figure 10. Trajectory Power Consumption for 3 Different Digging Time (5 s, 7.5 s, 10 s)

Digging trajectory is a very important factor to influence the performance of a hydraulic shovel. As indicated by Hendricks et al (1989), the shovel response may be more heavily dependent on the position of the bucket in the material rather than the material characteristics. Three different trajectories are considered with the end point on each trajectory being reached at the same time of 7.5 seconds and with full bucket loading capacity. The digging trajectories are shown in Figure 11 and the corresponding power consumption for the trajectories are shown in Figure 12 (case 1 corresponds to the trajectory with short but deep digging, case 3 with long but shallow digging and case 2 in between the first two cases). It can be seen that a short but deep trajectory (case 1) requires more energy consumption than a long but shallow one. A shallow trajectory make digging easy. The peak power required decreases about 50% when the shallow digging trajectory is executed instead of the deep one.

Figure 11. Three Trajectories for Bucket Tip of a Hydraulic Shovel

Figure 12 Power Consumption for 3 Trajectories with the Same Digging Time

Figure 13 illustrates the key components of a cable shovel excavator, which include undercarriage, upper structure and attachment. The attachment for cable shovels consists of the boom, crowd machinery, handle and bucket. The digging path of the dipper is produced by the extension/retraction of the handle (crowd) and by the cable hosting action. Hoisting of the dipper is accomplished by means of cables attached to the dipper which pass over sheaves at the boom point and spool on a deck mounted powered drum. The crowd action is produced either by cables or a direct rack and

pinion gear drive. Figure 14 shows geometrical structure of a cable shovel. A detailed mathematical modeling of cable shovel is given in Appendix #2.

Figure 13. Elements of Cable Shovel

Figure 14. Geometrical Structure of a Cable Shovel

Figure 15 shows a cable shovel simulator developed in ADAMS environment, while Figure 16 shows a 3D solid shovel model. Only the front-end assembly including handle and bucket of the cable shovel is modeled because this research is only concerned with the digging section of a duty cycle. Therefore, the rotation of the upper structure is not modeled and the handle and bucket assembly only moves on the vertical plane. Also, no joint friction between the hoist cable and the sheave and between the handle and the crowd pivot is considered.

Figure 15. A Cable Shovel in ADAMS Simulation Environment

Figure 16. 3D Solid Cable Shovel Model

An example is included to illustrate the performance of cable shovel based on given digging paths and to define efficient digging profiles based on energy consumption.

An assumed trajectory as shown in Figure 7 is proposed to simulate the digging path inside a formation or muck-pile. The formation or muck-pile has a slope of 50 degrees. The trajectory is designed such that at the end of its execution, the volume of the cut material is equal to the bucket capacity of the shovel. The bucket tip moves at constant speed and it takes different times to execute the entire trajectory. The interaction between the bucket and the excavated material is a complex problem and there are different models, which are applicable to different materials (Blouin et al, 2001). Length of crowd is 14.3 m, mass is 24,494 kg and Inertia moments is $1.179 \times 10^6 \text{ kgm}^2$.

Here the same trajectory as for hydraulic shovel (Figure 7) is completed within three different digging times: 5, 7.5 and 10 seconds. The power consumption for the three cases is simulated as shown in Figure 17. It can be seen that shortening the digging time causes higher power consumption peak. The peak power consumption increases 100% when the digging time reduces half. Therefore, there is a need for a trade-off between energy consumption and saving duty cycle. This is very important, because energy consumption is directly related to the forces exerted by the front-end components (handle and bucket), and more severe working environments for these parts are expected with shortened duty cycle. Note the horizontal axis in Figure 17 is non-dimensional for comparative purpose.

Figure 17. Power Consumption for the Same Trajectory with 3 Different Digging Times

Digging trajectory is a very important factor to influence the performance of a cable shovel. As indicated by Hendricks et al (1989), the shovel response may be more heavily dependent on the position of the bucket in the material rather than the material characteristics. Three different trajectories are considered with the end point on each

trajectory being reached at the same time of 7.5 seconds and with full bucket loading capacity. The digging trajectories are shown in Figure 18 and the corresponding power consumption for the trajectories is shown in Figure 19 (case 1 corresponds to the trajectory with short but deep digging, case 3 with long but shallow digging and case 2 in between the first two cases). It can be seen that the total work done for the three trajectories is no much difference for the current three trajectories. However, the peak power consumption is different with the middle trajectory (case 2), indicating the most severe working conditions under the current working environments.

Figure 18. Three Trajectories for Bucket Tip of a Cable Shovel

Figure 19. Power Consumption for 3 trajectories with the Same Digging Time Duration

One of the big concerns for cable shovels is the frequent replacement of the hoist cables due to cyclic fatigue loading and the friction between hoist cable and sheave. The working environment of the handle may be not so severe compared with hoist cable. However, some critical locations such as the contact points between the handle and saddle block may experience severe working conditions due to stress concentrations. Therefore, a simulation of the forces inside the hoist cable and the handle are usually essential for optimized design and maintenance of these critical components. Figure 20 shows the hoist force and handle force for the digging trajectory as shown in Figure 7 with a digging time of 7.5 seconds. It can be seen that the cable force is different from that of handle. The hoist cable force experiences its peak around its middle range while the force inside the handle experiences a positive (tensile) force at the beginning, with the force becoming negative (compressive) due to high digging force from the material formations. The handle force becomes positive near the end of the trip due to the material weight inside the bucket.

Figure 20. Cable forces change with time

Based on previous results, we were able to simulate the subsequent stress and/or performance analysis of the shovel components during excavation. Figure 21 shows typical display of stress values for shovel boom during excavation process.

Figure 21. Stress Display for Cable Shovel Boom During Excavation Process

The concept of data acquisition, display and control system for shovel loading technology is shown in Figure 22. The data acquisition system needs to be used to gather data on the wired instrumentations on both the cable and the hydraulic shovel components. A computer terminal needs to be connected to the acquisition setup for graphic imaging and representation.

Figure 22. Data Acquisition, Display and Control System for Shovel Technology

We propose the modular sensory system for shovels that (i) will ensure accurate and repeatable measurements; (ii) a reliable and durable structure, preferably rugged, that will meet the changing harsh conditions of the mining environment; and (iii) a cost effective solution in which the composite of sensors greatly enhances the value and performance of the entire shovel mining system. Force torque sensors collect information within the shovel's working envelope and transmit the data to the operator or Intranet system for online access. Adequate information can be gathered to characterize the material, equipment and process. These sensors could also be attached to a secured periphery of the shovel boom-bucket assembly. The ISE technology synthesizes the captured longitudinal or axial stress waves in the boom-bucket train via receiver probes on the boom and synthesizes the flux in digital form. After detailed study of different types of sensors we adopted the position sensor and pressure transducer shown in Figures 23 and 24.

Figure 23. EK600 position sensor by Admotec Inc.

Position sensor EK600 by Admotec, Inc. is available for shaft diameter from 2 to 140 mm (1/16" to 5.5"); temperature range: -40°C to 115°C (-40°F to 240°F); resolutions of 3,000 cycles (pulses) per revolution; and channel frequencies up to 500 kHz. It provides up to 10 times the resolution of comparable gear-tooth sensors in the same applications. They can be installed quickly and easily without the delicate alignment associated with optical encoder kits.

Ashcroft Model V2 Pressure Transducer by Intertechnology Inc has a 1% interchangeability from unit to unit - laser trimmed technology, fused silicon for superior

linearity and repeatable performance, extremely high proof and pressure tolerance, and wide operating range of -40°C to 121°C (-40° to +250° F).

Figure 24. Ashcroft Model V2 Pressure Transducer by Intertechnology Inc.

The suggested system requires software to interface the hardware with the computer(s). This software should be able to provide easy data acquisition, data analysis and presentation. Also, the software should be able provide control of the shovel motion. We adopted LabView 7.1 for this purpose. In addition, this software would provide easy means of configuring the hardware to work with the computer(s).

The obtained research results described above enabled us to achieve the first two objectives of the project: (i) provide the theoretical basis to develop the ISE technology to solve the problems associated with excavation in material formation; and (ii) advance knowledge and frontiers in shovel excavation.

We propose design, development and implementation of the novel technology for shovel excavation through the DOE Mining Industry of the Future: Area of Interest – Energy Efficient Alternatives to Current Technologies in Extraction. We would test design features, construct a field ready prototype and perform field testing at the surface mining sites. Multi-partner collaboration would include research organization, shovel manufacturer, hardware and sensor technology companies, and surface mining companies. The Pennsylvania State University and University of Missouri Rolla will invite P&H Mining (shovel manufacturer), Thunderbird Mining Systems (shovel monitors, hardware and software), Admotec, Inc. and Intertechnology, Inc. (sensor technology), and North American Coal Corporation and Hanson Aggregates (industry sites) to participate in the project.

The project would be done over the 5-year period and would include the following three phases:

- (i) Phase I: The researchers from the PSU and UMR will provide an engineering design of shovel prototype.
- (ii) Phase II: PSU and UMR will invite P&H to manufacture a field-ready shovel prototype based on previous design, and Thunderbird Mining Systems, Admotec, Inc, and Intertechnology to provide monitors, hardware, software and sensors for field testing.
- (iii) Phase III: Technology will be tested at coal surface mine (North American Coal Corporation) and stone quarry (Hanson Aggregates).

Energy Benefits

This project has formulated the theory and the technical concepts, with mathematical and simulation modeling techniques, for developing an intelligent shovel excavation (ISE) technology (Phase I). The next phase of the project (Phase II) should provide design, development and implementation of the ISE technology for shovel excavation at experimental surface mining site.

It is anticipated that 10 % in energy savings including electricity (cable shovel) and fuel (hydraulic shovel) will be achieved by implementation of ISE technology. The energy savings are due to increased job efficiency where real-time information and precise knowledge of the material properties provide on-the-job decisions by shovel operators and mine production engineers to guide shovel excavation. The ISE technology assists operators to navigate shovel excavation based on the established correlation between material properties and dipper excavation.

According to the P&H MinePro Services studies (2003), a single cable shovel with bucket of 38.2 m³ (50 yd³), over the period of 7,000 hours, requires an average electricity use of 8,050 megawatt hours. Over the same period, a hydraulic excavator with the same bucket size, consumes 4,352,750 liters (1,150,000 gallons) of fuel. Therefore, by applying ISE technology, the electricity savings for the cable shovel of 50 yd³ dipper size will be 805 megawatt hours over the 7,000 hours, while the fuel savings for the hydraulic shovel of the seam size and over the same period will be 115,000 gallons.

The following describes the shovel energy consumption summary estimates for three commodities including coal, copper and stone. Table 3 shows estimated energy savings for single shovel. It should be noted that the energy per ton produced for current technology was derived from the DOE's methodology on the OIT-mining website.

Table 3. Estimated energy savings per ton produced for single unit

Hydraulic Shovel			
Commodity	Current technology - Single Unit (BTU/ton) A	Savings per Single Unit (BTU/ton) B	Proposed technology - Single Unit (BTU/ton) C
Coal	3,861	386.1	3,474.9
Stone	5,141	514.1	4,626.9
Cable Shovel			
Commodity	Current technology - Single Unit (BTU/ton) A	Savings per Single Unit (BTU/ton) B	Proposed technology - Single Unit (BTU/ton) C
Coal	2,494	249.4	2,244.6
Copper	1,214	121.4	1,092.6

Table 4 shows the energy savings for four hypothetical surface mines with the following assumptions:

- Coal surface mine; annual production AP = 2,000,000 t; and principal equipment: hydraulic shovel.
- Stone surface mine; annual production AP = 5,000,000 t; and principal equipment: hydraulic shovel.
- Coal surface mine; annual production AP = 5,000,000 t; and principal equipment: cable shovel.
- Copper surface mine; annual production AP = 42,000,000 t; and principal equipment: cable shovel.

Table 4. Estimated annual energy savings for single unit

Hydraulic Shovel			
Commodity	Current technology - Single Unit (BTU/year) A1	Proposed technology - Single Unit (BTU/year) B1	Savings per Single Unit (BTU/year) C1
Coal	7,722,000,000	6,949,800,000	772,200,000
Stone	25,705,000,000	23,134,500,000	2,570,500,000
Cable Shovel			
Commodity	Current technology - Single Unit (BTU/year) A1	Proposed technology - Single Unit (BTU/year) B1	Savings per Single Unit (BTU/year) C1
Coal	12,470,000,000	11,223,000,000	1,247,000,000
Copper	50,988,000,000	45,889,200,000	5,098,800,000

The following formulas have been used to estimate annual energy consumption for single shovel:

$$A1 = AP \times A$$

$$B1 = AP \times C$$

$$C1 = A1 - B1$$

where

A1 – estimated annual energy consumption with current technology (BTU/year)

AP – annual production (ton/year)

A – estimated energy consumption per ton produced with current technology (BTU/ton)

B1 - annual energy consumption with proposed technology (BTU/year)

C - estimated energy consumption per ton produced with proposed technology (BTU/ton)

C1 – annual energy savings per single shovel (BTU/year)

It can be seen from the Table 4 that proposed technology for hydraulic shovel could save 772,200,000 BTU/year per single shovel in coal surface mine with annual production of 2 million tons. The same technology enables energy savings for the single shovel at stone mine, with annual production of 5 million tons, at 2,570,500,000 BTU/year.

The proposed technology for cable shovel will provide energy savings of 1,247,000,000 BTU/year per single shovel in coal surface mine with annual production of 5 million tons. The same technology enables energy savings for the single shovel at copper surface mine, with annual production of 42 million tons, at 5,098,800,000 BTU/year.

CONCLUSIONS

This project advances knowledge and frontiers in shovel excavation, and provide a basis for the Intelligent Shovel Excavation (ISE) technology to solve the problems associated with material excavation.

Analysis of the shovel excavator was performed applying the Newton-Euler equations and Kane's method. In this project, transformation matrix method is used to describe the motion of machine and analyze the kinematics relationship; then Kane method is applied to build static and dynamic model in closed form.

Both hydraulic and cable shovel simulators for surface mining were developed using ADAMS simulation environment. The parameterized characteristics of the simulator make it easy to investigate the effect on shovel performance under varying working conditions. The simulator captures the kinematics and dynamics of the shovel within its operating environment. This parameterized simulator provides a powerful tool for performance monitoring, excavation process designs and structural optimization of shovels. It also enables a virtual prototyping environment for complex mechanical systems and performing detailed analysis before detailed design and/or manufacturing to minimize the use of costly physical prototypes. The virtual prototyping technique may also be useful in improving design quality and dramatically reducing product

development time. The research results of this study are extremely important, since it can influence further technology development in the mining field.

Based on research results of this study, it can be concluded that changes in parameters such as digging time or trajectory influence the performance of hydraulic shovels, as indicated by different power consumptions for different digging times and different digging trajectories. A shortened digging time increases the power required for the operation and deteriorates the working environments of the shovel assembly. The peak power required almost doubles when the digging time reduces half. The simulation on the effect of different digging trajectories also shows that a shallow trajectory makes it easier to dig than a deep trajectory when the two trajectories are traveled within the same time duration. The peak power required decreases about 50% when the shallow digging trajectory is executed instead of the deep one. Simulation model of cable shovel shows that the forces inside the hoist cable and the handle are usually essential for optimized design and maintenance of these critical components. It can be concluded that the hoist cable force experiences its peak around its middle range while the force inside the handle experiences a positive (tensile) force at the beginning, with the force becoming negative (compressive) due to high digging force from the material formations. The handle force becomes positive near the end of the trip due to the material weight inside the bucket.

The area of sensor technology was studied. Application of specific wrist-mounted sensors to characterize the material, bucket and frame assembly was determined. Data acquisition, display and control system for shovel loading technology was adopted. The concept of data acquisition and control system was designed and the stress of shovel components was simulated. It is determined that position sensor and pressure transducer can be configured and applied for shovel technology.

It is anticipated that 10 % in energy savings including electricity and fuel will be achieved by implementation of ISE technology. This project was a paper study on ground Breaking Innovative Technology Concepts for Mining. Therefore this report indicates estimates on energy savings if proposed technology would be applied. In order to get an accurate numbers, the prototype technology should be developed and tested in the field.

REFERENCES

1. Alekseeva, T.V., Artemev, K.A., Voitsekhouskii, R.I., and Ulyanov, N.A., 1985. Machines for earthmoving work, theory and calculations. Amerind Publishing Co., New Delhi, India.
2. Araya, H., Kakuzen, M., Kimura, N. and Hayashi, N., 1988. Automatic control system for hydraulic shovels. Proceedings of the USA-Japan Symposium on Flexible Automation - Crossing Bridges: Advances in Flexible Automation and Robotics. Minneapolis, MN, USA, p.695-701.

3. Blouin, S., Hemami, A. and Lipsett, M., 2001. Review of resistive force models for earthmoving processes. *Journal of Aerospace Engineering*, Vol.14, No.3, p.102-111.
4. BSC Incorporated, Appendix: Data quality and methodology for calculating energy consumption estimates. <http://www.eere.energy.gov/industry/mining/analysis.html>
5. Craig, J.J. 1986. *Introduction to Robotics: Mechanics and Control*. © Addison-Wesley Publishing Company, Reading.
6. Daneshmend, L., C.Hendricks, S.Wu, and M.Scoble. 1993. "Design of a Mining Shovel Simulator". *Mine Design for 21st Century*, Bawden& Archibald (Ed.). © Balkema, Rotterdam.
7. Frimpong, S., Y.Hu and J.Szymanski. 2002. "Dynamic Shovel Excavation Simulator in Surface Mining Operations". *SCSC 2002 Proc.* San Diego, CA, July 14-19 (CD-ROM).
8. Frimpong, S., Y.Hu and Z.Chang. 2003. "Hydraulic Shovel Simulator in Surface Mining Excavation Engineering". *12th MPES*, Kalgoorlie, West Australia, April 23-25 (CD ROM).
9. Haug, E.J. 1989. *Computer Aided Kinematics and Dynamics of Mechanical System, Volume I: Basic Methods*. © Allyn and Bacon, Boston.
10. Hendricks, C., Scoble, M.J., and Peck, J., 1989. Performance monitoring of electric mining shovels. *Transactions of the Institution of Mining and Metallurgy, Section A: Mining Industry*, Vol.98, September-December, p.A151-A159.
11. Imanishi, E., Zui, H., Yoshimatsu, H. and Niwata, K., 1987. Study on simulation of linkage mechanism. *Nippon Kikai Gakkai Ronbunshu, C Hen*, Vol.53, No.492, p.1711-1719.
12. Kane, T.R. and D.A.Levinson. 1985. *Dynamics: Theory and Applications*. © McGraw-Hill, NY.
13. Koivo, A.J., 1994. Kinematics of excavators (backhoes) for transferring surface materials. *Journal of Aerospace Engineering*, Vol.7, No.1, p.17-32.
14. Koivo, A.J., Thoma, M., Kocaoglan, E. and Andrade-Cetto, J., 1996. Modeling and control of excavator dynamics during digging operation. *J. Aerospace Eng.*, Vol.9, No.1, p.10-18.
15. Luengo, O., Singh, S. and Cannon, H., 1998. Modeling and identification of soil-tool interaction in automated excavation. *Proceedings of the 1998 IEEE/RSJ*

International Conference on Intelligent Robots and Systems, Victoria, BC, Canada, p.1900-1906.

16. Murray, R.M., Z.Li and S.S.Sastry. 1993. *A Mathematical Introduction to Robotic Manipulation*. © CRC Press, Boca Raton, USA.
17. Novak, A.J. and Larson, C., 1991. A computer simulation of backhoe type excavators. International Off-Highway and Power plant Congress and Exposition, SAE SP-884, Milwaukee, Wisconsin, USA, p.41-48.
18. P&H MinePro Services, 2003. Peak Performance Practices: Excavator Selection. Harnischfeger Corporation, pp. 1-69.
19. Shishaev, S.V. and Mochalov, E.A., 1989. Evaluating the characteristics of the working process of a hydraulic power shovel with an active dipper. Soviet Mining Science, Vol.24, No.5, p.452-457.
20. Tafazoli, S., P.D.Lawrence, and S.E.Salcudean. 1999. "Identification of Inertial and Friction Parameters from Excavator Arms". *IEEE Trans. on Robotics and Autom.*, 15(5): 966~971.
21. Wadell, C. and Maier, 1998, GPS Ore Control at Phelps Dodge Morenci; © Society for Mining, Metallurgy and Exploration Inc., Preprint 98-166: 2.

Appendix #1 - Hydraulic Shovel

Newton-Euler method

Between the two methods for describing the kinematics and dynamics of shovel excavators, Newton-Euler method is preferred over Lagrange one because it provides detailed information on all links and joints, which will be useful in the subsequent stress and/or performance analysis of the components during excavation. The attention here is restricted to the boom, stick and bucket assembly of the hydraulic shovels because this research is only concerned with the digging section of a duty cycle. Therefore, the rotation of the upper structure is not modeled and the bucket, stick and boom assembly only moves on the vertical plane. In Newton-Euler method, the velocities and accelerations are first computed iteratively for each link from boom to stick and then to bucket based on the kinematics of rigid bodies.

$$\begin{aligned}
 \vec{\omega}_C &= \vec{\omega}_B + (\dot{\theta}_3 \vec{k}) \\
 \vec{\dot{\omega}}_C &= \vec{\dot{\omega}}_B + (\ddot{\theta}_3 \vec{k}) + \vec{\omega}_B \times (\dot{\theta}_3 \vec{k}) \\
 \vec{v}_C &= \vec{v}_B + (\dot{\theta}_3 \vec{k}) \times \vec{r}_{BC} \\
 \vec{\dot{v}}_{C_3} &= \vec{\dot{v}}_B + \vec{\dot{\omega}}_C \times \vec{r}_{BC_3} + \vec{\omega}_B \times (\vec{\omega}_B \times \vec{r}_{BC_3}) \\
 \vec{\dot{v}}_C &= \vec{\dot{v}}_B + \vec{\dot{\omega}}_C \times \vec{r}_{BC} + \vec{\omega}_B \times (\vec{\omega}_B \times \vec{r}_{BC})
 \end{aligned} \tag{1}$$

Figure 2 shows a free body diagram of a stick during excavation. Under these assumptions, the kinematics equation for the stick is given as equation (1). $\vec{\omega}_B$, $\vec{\dot{\omega}}_B$, $\vec{\omega}_C$ and $\vec{\dot{\omega}}_C$ are the angular velocity and angular acceleration of the previous link (boom) and the stick, respectively. $\vec{v}_B$, $\vec{\dot{v}}_B$, $\vec{v}_C$ and $\vec{\dot{v}}_C$ are the translational velocity and translational acceleration at joint point B and C of the stick, respectively. $\dot{\theta}_3$ and $\ddot{\theta}_3$ are the angular velocity and angular acceleration of the local coordinate system of the stick relative to that of the boom. The inertial force, ${}^3\vec{F}_3$, and moment, ${}^3\vec{M}_3$, acting on the stick can then be determined by applying the Newton-Euler's equation. The mathematical model in the local coordinate frame (about the gravitational center of the stick) is given by equation (2). m_3 and I_3 are the mass and inertial moment of the stick, respectively. From the free body diagram of the stick, all forces and moments are balanced resulting in the balance equation (3).

$$\begin{aligned}
 {}^3\vec{M}_3 &= I_3 \vec{\dot{\omega}}_C + \vec{\omega}_C \times (I_3 \vec{\omega}_C) \\
 {}^3\vec{F}_3 &= m_3 \vec{\dot{v}}_{C_3}
 \end{aligned} \tag{2}$$

$$\begin{aligned}
 \vec{M}_B + \vec{M}_C + \vec{r}_{C_3B} \times \vec{F}_B + \vec{r}_{C_3B} \times \vec{F}_C + \vec{r}_{C_3I} \times \vec{F}_S &= {}^3\vec{M}_3 \\
 \vec{F}_B + \vec{F}_C + \vec{F}_S &= {}^3\vec{F}_3
 \end{aligned} \tag{3}$$

$\vec{F}_B$ and $\vec{F}_C$ are the forces acting on the stick from the boom and bucket, respectively, and expressed in the local coordinate frame. $\vec{F}_s$ is the hydraulic force exerted on the stick by the stick cylinder. $\vec{M}_B$ and $\vec{M}_C$ are the moments acting on the stick from the boom and bucket, respectively, and expressed in the local coordinate frame. The force balance equations can be established for boom and bucket as well. The complete set of the equations for describing the kinematics and dynamics of the boom, stick and bucket assembly involves near fifty equations, which can be solved for unknown parameters based on prescribed inputs. For example, with a given digging trajectory and the digging force at the tip of the bucket, it is possible to calculate joint forces between links and hydraulic forces inside three cylinders, therefore, establishing the relations among the digging force, hydraulic forces and other factors.

The interaction between the bucket and the excavated material is a complex problem and there are different models, which are applicable to different materials (Blouin et al, 2001). The resistive force, F_r , as shown in Figure 3 was calculated using the model by Alekseeva et al (1985):

$$F_r = k_p \left[k_s b h + \mu N + \varepsilon \left(1 + \frac{V_s}{V_b} \right) b h \sum_i \Delta x_i \right] \quad (4)$$

k_p and k_s are specific resistances in digging material; constants b and h are width and thickness of cut slice of the material, respectively; μ is the friction coefficient of the bucket and the cutting material; N is the pressure force of the bucket with the cutting material; ε is the coefficient of resistance experienced in filling the bucket during the movement of the prism of soil; V_s and V_b are volumes of the prism of cutting material and the bucket, respectively; and Δx_i is the increment along the horizontal axis. θ_b defines the angle between the foregoing plane that contains the bottom of the bucket and the X_4 -axis. The digging angle, θ_{dg} , is defined as the tangential angle of the proscribed trajectory with the horizontal line. With the time dependence of the trajectory known, the digging angles with time can be determined for each instant during the excavation process. F_t and F_n are the tangential and normal components of the resistive force.

Shovel performance is defined as power consumption of all cylinders during digging operation. The power delivered to each cylinder is the product of the force applied by that cylinder and the extension rate of this cylinder (assuming ideal hydraulic cylinders). Novak and Larson (1991) estimates the power delivered to the boom cylinder in equation (5).

$$P_b = \left| \vec{F}_b \cdot \vec{V}_b \right| \quad (5)$$

$\vec{F}_b$ and $\vec{V}_b$ are the hydraulic force and extension rate of the boom cylinder, respectively. The total power required to drive the excavator is simply the sum of these individual powers:

$$P_{total} = |\vec{F}_b \cdot \vec{V}_b| + |\vec{F}_s \cdot \vec{V}_s| + |\vec{F}_{bu} \cdot \vec{V}_{bu}| \quad (6)$$

$\vec{F}_s$ and $\vec{V}_s$ and $\vec{F}_{bu}$ and $\vec{V}_{bu}$ are the respective hydraulic forces and extension rates of the stick and bucket cylinders.

Kane method

The basic components of a hydraulic shovel are shown in Fig. 1. The global frame is A -X₁Y₁Z₁. We also define three local frame: the frame of A-X₂Y₂Z₂ is fixed to boom in point A; The frame of B-X₃Y₃Z₃ is fixed on stick in point B; and the coordinate system of C- X₄Y₄Z₄ is fixed to bucket in point C. From the structure shown in Fig 1, the transformation matrix of boom, stick, and bucket to base frame A -X₁Y₁Z₁ in origin A can be obtained as ${}^1_2A$, ${}^2_3A$, ${}^3_4A$ respectively (Craig, 1986). They are listed in follow.

$${}^1_2A = \begin{bmatrix} \cos \theta_1 & -\sin \theta_1 & 0 & 0 \\ \sin \theta_1 & \cos \theta_1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (1a)$$

$${}^2_3A = \begin{bmatrix} \cos \theta_2 & -\sin \theta_2 & 0 & l_1 \\ \sin \theta_2 & \cos \theta_2 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (1b)$$

$${}^3_4A = \begin{bmatrix} \cos \theta_3 & -\sin \theta_3 & 0 & l_2 \\ \sin \theta_3 & \cos \theta_3 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (1c)$$

According to (1a~c), transformation matrix between each component to base frame can be obtain. And the ${}^1_3A$, ${}^1_4A$ present the matrix of boom, stick and bucket to base frame. They also are listed in following:

$${}^1_3A = {}^1_2A \cdot {}^2_3A = \begin{bmatrix} \cos(\theta_1 + \theta_2) & -\sin(\theta_1 + \theta_2) & 0 & l_1 \cos \theta_1 \\ \sin(\theta_1 + \theta_2) & \cos(\theta_1 + \theta_2) & 0 & l_1 \sin \theta_1 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (2a)$$

$${}^1_4A = {}^1_2A \cdot {}^2_3A \cdot {}^3_4A = \begin{bmatrix} \cos(\theta_1 + \theta_2 + \theta_3) & -\sin(\theta_1 + \theta_2 + \theta_3) & 0 & l_1 \cos \theta_1 + l_2 \cos(\theta_1 + \theta_2) \\ \sin(\theta_1 + \theta_2 + \theta_3) & \cos(\theta_1 + \theta_2 + \theta_3) & 0 & l_2 \sin \theta_1 + l_1 \sin(\theta_1 + \theta_2) \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (3a)$$

Where the l_1 , l_2 , l_3 are the length of boom, stick and bucket; $\theta_1, \theta_2, \theta_3$ are the angle of boom, stick and bucket as Figure 1. According to the above transmission matrix, we can describe the global coordinate of each point in different component. For example, to any point in bucket whose local coordinate value is (x_3, y_3) , the values of global coordinate is given by equation (4a).

$$\begin{Bmatrix} X_3 \\ Y_3 \\ 0 \\ 1 \end{Bmatrix} = \begin{bmatrix} \cos(\theta_1 + \theta_2 + \theta_3) & -\sin(\theta_1 + \theta_2 + \theta_3) & 0 & l_1 \cos \theta_1 + l_2 \cos(\theta_1 + \theta_2) \\ \sin(\theta_1 + \theta_2 + \theta_3) & \cos(\theta_1 + \theta_2 + \theta_3) & 0 & l_2 \sin \theta_1 + l_1 \sin(\theta_1 + \theta_2) \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{Bmatrix} x_3 \\ y_3 \\ 0 \\ 1 \end{Bmatrix} \quad (4a)$$

Similarly, the global coordinate of point in boom or stick can be obtained by same method. Based on above transformation matrix, inverse kinematics and direct kinematics can be studied. Direct kinematics analysis is to obtain trajectory of bucket tip or other points in components according to the angle of each joint in joint space. On the other hand, if the trajectory of bucket tip is known, the processing of solving joint angle is called inverse kinematics. In the two processes, the direct kinematics is simpler than inverse kinematics. In inverse kinematics analysis, solution of algebra equation is necessary. In most case, the algebra equations are nonlinear. In hydraulic shovel, if the motion of tract is neglected, we can use equation (4a) to do inverse kinematics analysis. In equation (4a), there are two equations. If only the trajectory of tip is given, the angle of each joint is not unique. So another constraint of orient of bucket should be added. It means that if the angle of $(\theta_1 + \theta_2 + \theta_3)$ is known, we can determine the angle of $\theta_1, \theta_2, \theta_3$. If the trajectory of bucket is given, the two degrees of angle and displacement is fixed. So the hydraulic shovel is more flexible than cable shovel. It can dig special area in operations.

In hydraulic shovel, hydraulic cylinders are actuators to overcome the resistant force and drive components. They form three kinematics constraint relationships as follow:

$$c_1^2 = (h_{1x} \cos \theta_1 - h_{1y} \sin \theta_1 - h_{0x})^2 + (h_{1x} \sin \theta_1 + h_{1y} \cos \theta_1 - h_{0y})^2 \quad (5a)$$

$$c_2^2 = (l_1 + h_{21x}c_2 - h_{21y}s_2 - h_{12x})^2 + (h_{21y} \sin \theta_2 + h_{21x} \cos \theta_2 - h_{12y})^2 \quad (5b)$$

$$c_3^2 = (l_1 + l_2c_2 + h_{31x}c_{23} - h_{31y}s_{23} - h_{13x})^2 + (l_2s_2 + h_{31x}s_{23} + h_{31y}c_{23} - h_{13y})^2 \quad (5c)$$

c_1 is the length of cylinder of boom; c_2 is the length of arm cylinder; c_3 is the length of bucket cylinder; h_{0x} and h_{0y} , the coordinate value of point of boom cylinder in

undercarriage; h_{1x} and h_{1y} , the coordinate value of point of boom cylinder in undercarriage; h_{21x} and h_{21y} , the coordinate value of point of stick cylinder in stick; h_{12x} and h_{12y} , the coordinate value of point of stick cylinder in boom; h_{31x} and h_{31y} , the coordinate value of point of bucket cylinder in bucket; h_{13x} and h_{13y} , the coordinate value of point of bucket cylinder in boom. Based on transformation matrix, the velocities and accelerations can individually be derived. To the points in the bucket, velocities and accelerations are

$$\begin{bmatrix} X \\ Y \\ 0 \\ 1 \end{bmatrix}' = \begin{bmatrix} -\sin \theta_{123} \dot{\theta}_{123} & -\cos \theta_{123} \dot{\theta}_{123} & 0 & -l_1 \sin \theta_1 \dot{\theta}_1 - l_2 \sin \theta_{12} \dot{\theta}_{12} \\ \cos \theta_{123} \dot{\theta}_{123} & -\sin \theta_{123} \dot{\theta}_{123} & 0 & l_1 \cos \theta_1 \dot{\theta}_1 + l_2 \cos \theta_{12} \dot{\theta}_{12} \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ 0 \\ 1 \end{bmatrix} \quad (6a)$$

$$\begin{bmatrix} X \\ Y \\ 0 \\ 1 \end{bmatrix}'' = \begin{bmatrix} -\sin \theta_{123} \ddot{\theta}_{123} - \cos \theta_{123} \dot{\theta}_{123}^2 & -\cos \theta_{123} \ddot{\theta}_{123} + \sin \theta_{123} \dot{\theta}_{123}^2 & 0 & a_1 \\ \cos \theta_{123} \ddot{\theta}_{123} - \sin \theta_{123} \dot{\theta}_{123}^2 & -\sin \theta_{123} \ddot{\theta}_{123} - \cos \theta_{123} \dot{\theta}_{123}^2 & 0 & a_2 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ 0 \\ 1 \end{bmatrix} \quad (6b)$$

$$a_1 = -l_1 (\sin \theta_1 \ddot{\theta}_1 + \cos \theta_1 \dot{\theta}_1^2) - l_2 (\sin \theta_{12} \ddot{\theta}_{12} + \cos \theta_{12} \dot{\theta}_{12}^2) \quad (6c)$$

$$a_2 = l_1 (\cos \theta_1 \ddot{\theta}_1 - \sin \theta_1 \dot{\theta}_1^2) + l_2 (\cos \theta_{12} \ddot{\theta}_{12} - \sin \theta_{12} \dot{\theta}_{12}^2) \quad (6d)$$

In this section, Kane method is used to develop and analyze the structure of the static and dynamic problems associated with the hydraulic shovel. The process begins with defining the general speeds $u_1, u_2, \dots, u_n$ first; then the forces acting on the system are calculated, including external forces $(F_i)_j$, torques $(T_i)_j$, inertial force, $(F_i)_j^*$ and inertial torques $(T_i)_j^*$. Then, the dynamics motion equations are formulated as follow:

$$F_r + F_r^* = 0 \quad (7a)$$

$$F_r^* = \sum_{i=1}^m [(F_i^*)_r \frac{\partial V_i}{\partial u_r} + (T_i^*)_r \frac{\partial W_i}{\partial u_r}]; \quad F_i^* = -m_i a_i; \quad T_i^* = -I_i \varepsilon_i; \quad \text{and} \quad F_r = \sum_{i=1}^m [(F_i)_r \frac{\partial V_i}{\partial u_r} + (T_i)_r \frac{\partial W_i}{\partial u_r}]$$

The general speeds are given as: u_1, u_2, u_3 , where $u_1 = \dot{\theta}_1$, $u_2 = \dot{\theta}_2$ and $u_3 = \dot{\theta}_3$. Then we can solve the partial velocity for different velocity to general speed. For example, to the points in bucket, the partial velocities are shown as follow:

$$\frac{\partial v_3}{\partial \dot{\theta}_1} = \begin{bmatrix} -\sin \theta_{123} & -\cos \theta_{123} \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} + \begin{bmatrix} -\sin \theta_{12} & -\cos \theta_{12} \end{bmatrix} \begin{Bmatrix} l_2 \\ 0 \end{Bmatrix} + \begin{bmatrix} -\sin \theta_1 & -\cos \theta_1 \end{bmatrix} \begin{Bmatrix} l_1 \\ 0 \end{Bmatrix} \quad (8a)$$

Differentiate of constriction equation (5a~c), we can obtain the relationship between the velocities of cylinder and linkage angle as below:

$$\frac{\partial \dot{c}_1}{\partial \theta_1} = \frac{1}{c_1} [h_{0x}(h_{1x}s_1 + h_{1y}c_1) - h_{0y}(h_{1x}c_1 - h_{1y}s_1)] \quad (9a)$$

$$\frac{\partial \dot{c}_2}{\partial \theta_2} = \frac{1}{c_2} [(l_1 - h_{12x})(-h_{21x}s_2 - h_{21y}c_2) - h_{12y}(h_{21y}c_2 - h_{21x}s_2)] \quad (10a)$$

$$\frac{\partial \dot{c}_3}{\partial \theta_2} = \frac{1}{c_3} [(l_1 - h_{13x})(-l_2s_2 - h_{31x}s_{23} - h_{31y}c_{23}) + (-h_{13y})(l_2c_2 + h_{31x}c_{23} - h_{31y}s_{23})] \quad (11a)$$

h_{21x} and h_{21y} are coordinate values of point of arm cylinder in stick;

All the external forces incident on the shovel are included in developing the system static model. These forces include the hydraulic force from cylinders, gravity of the handle, the payload for shoveling resistance and gravity of material. The equations for static force are:

$$T_1 \frac{\partial \dot{c}_1}{\partial \theta_1} + \vec{F} \frac{\partial \vec{s}_1}{\partial \theta_1} + M_1 g(x_1 \cos \theta_1 - y_1 \sin \theta_1) + M_2 g(l_1 \cos \theta_1 + x_2 \cos \theta_{12} - y_2 \sin \theta_{12}) + M_3 g(l_1 \cos \theta_1 + l_2 \cos \theta_{12} + x_3 \cos \theta_{123} - y_3 \sin \theta_{123}) = 0 \quad (12a)$$

$$T_2 \frac{\partial \dot{c}_2}{\partial \theta_2} + T_3 \frac{\partial \dot{c}_3}{\partial \theta_2} + \vec{F} \frac{\partial \vec{s}}{\partial \theta_2} + M_2 g(x_2 \cos \theta_{12} - y_2 \sin \theta_{12}) + M_3 g(l_2 \cos \theta_{12} + x_3 \cos \theta_{123} - y_3 \sin \theta_{123}) = 0 \quad (12b)$$

$$T_3 \frac{\partial \dot{c}_3}{\partial \theta_3} + \vec{F} \frac{\partial \vec{s}}{\partial \theta_3} + M_3 g(x_3 \cos \theta_{123} - y_3 \sin \theta_{123}) = 0 \quad (12c)$$

According to above static equation, some physical significance of items can be studied. The three equations are corresponding to three general speed u_1 , u_2 and u_3 . Therefore,

$T_1 \frac{\partial \dot{c}_1}{\partial \theta_1}$, $T_2 \frac{\partial \dot{c}_2}{\partial \theta_2} + T_3 \frac{\partial \dot{c}_3}{\partial \theta_2}$ and $T_3 \frac{\partial \dot{c}_3}{\partial \theta_3}$ are the driving torques to θ_1 , θ_2 and θ_3 . For torque

$T_1 \frac{\partial \dot{c}_1}{\partial \theta_1}$, $\frac{\partial \dot{c}_1}{\partial \theta_1}$ is arm of the force. Similarly, we can analyze the contribution on whole system of different forces as resistant force, gravity of material and components.

Appendix #2 - Cable Shovel

Kinematics Analysis of Cable Shovel

In this section, exponential formula based on the screw theory is used to analyze the kinematics of cable shovel. Using this method, the kinematic equations can be developed and analyzed within a global coordinate system, as illustrated in Figure 25.

Figure 25. Spatial Coordinates of the Exponential Formulation

In this figure, the velocity of a vector rotating about an axis through the origin is given by equation (a1).

$$\dot{q}(t) = \omega \times q(t) = \hat{\omega} \cdot q(t) \quad (a1)$$

The displacement vector can be obtained via the exponential formulation in equations (a2) and (a3).

$$q(t) = e^{\hat{\omega}\phi} q(t_0) \quad (a2)$$

$$e^{\hat{\omega}\theta} = I + \hat{\omega} \sin \phi + \hat{\omega}^2 \cos \phi \quad (a3)$$

This vector can also be considered as the transformation matrix in the motion. According to the screw theory, any rigid motion can be represented as a rotation about the axis and a translation along the same axis. Therefore, in a general case, the rotation and translation motions about the same axis can be obtained as equations (a4), (a5) and (a6) from Murray et al (1993).

$$q(t) = e^{\hat{\xi}\theta} q(t_0) \quad (a4)$$

$$\hat{\xi} = \begin{bmatrix} \hat{\omega} & -\omega \times q + h\omega \\ 0 & 1 \end{bmatrix} \quad (a5)$$

$$e^{\hat{\xi}\theta} = \begin{bmatrix} e^{\hat{\omega}\theta} & (1 - e^{\hat{\omega}\theta})q + h\theta\omega \\ 0 & 1 \end{bmatrix} \quad (a6)$$

In equation (a4), ξ is a parameter that includes information of a rotation vector and a point in the axis of rotation. Thus, if the axis vector plus any point in the vector space and the values of rotation and translation are known, their transformation matrix can be easily obtained from the exponential method.

The structural frame of a cable shovel boom-dipper assembly is illustrated in Figure 14. The global coordinate system is $O_0X_0Y_0Z_0$; the local coordinate system of handle in the initial position is $O_{10}X_{10}Y_{10}Z_{10}$; local coordinate system of handle in motion is shown as $O_1X_1Y_1Z_1$. The axis of rotation motion of handle is $(0,0,1)^T$; the appointed point in axis is given as $[l_0 \cos \alpha_0 \quad l_0 \sin \alpha_0 \quad 0]^T$. Angular displacement of motion is θ ; the displacement of translation is l_x . If the bucket only rotates in the position 1, the transformation matrix is given by equation (a7) according to exponential formula.

$${}^1_2A = \begin{bmatrix} \cos \theta & \sin \theta & 0 & l_0 \cos \alpha_0 \\ \sin \theta & \cos \theta & 0 & l_0 \sin \alpha_0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (a7)$$

If the bucket only translates along the X_1 coordinate axis, the transformation matrix is given by equation (a8).

$${}^2_3A = \begin{bmatrix} 1 & 0 & 0 & l_x \cos \theta \\ 0 & 1 & 0 & l_x \sin \theta \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (a8)$$

Thus, the total transformation matrix including rotation and translation is given by equation (a9).

$${}^1_3A = \begin{bmatrix} \cos \theta & \sin \theta & 0 & l_0 \cos \alpha_0 + l_x \cos \theta \\ \sin \theta & \cos \theta & 0 & l_0 \sin \alpha_0 + l_x \sin \theta \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \quad (a9)$$

Therefore, the coordinate values of any position in the handle can be obtained by the exponential formula as equation (a10).

$$P = \begin{bmatrix} \cos \theta & -\sin \theta & 0 & l_0 \cos \alpha_0 + l_x \cos \theta \\ \sin \theta & \cos \theta & 0 & l_0 \sin \alpha_0 + l_x \sin \theta \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ 0 \\ 1 \end{bmatrix} = \begin{bmatrix} x \cos \theta - y \sin \theta + l_0 \cos \alpha_0 + l_x \cos \theta \\ x \sin \theta + y \cos \theta + l_0 \sin \alpha_0 + l_x \sin \theta \\ 0 \\ 1 \end{bmatrix} \quad (a10)$$

The vector of velocity and acceleration are shown in equation (a11) and (a12).

$$V = \begin{bmatrix} -\sin \theta \dot{\theta} & \cos \theta \dot{\theta} & 0 & \dot{l}_x \cos \theta - l_x \sin \theta \dot{\theta} \\ \cos \theta \dot{\theta} & -\sin \theta \dot{\theta} & 0 & \dot{l}_x \sin \theta + l_x \cos \theta \dot{\theta} \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} x \\ y \\ 0 \\ 1 \end{bmatrix} \quad (a11)$$

$$a = \begin{bmatrix} -\sin \theta \ddot{\theta} - \cos \theta \dot{\theta}^2 & -\cos \theta \ddot{\theta} + \sin \theta \dot{\theta}^2 & 0 & \ddot{l}_x \cos \theta - 2\dot{l}_x \sin \theta \dot{\theta} - l_x \cos \theta \ddot{\theta} - l_x \sin \theta \dot{\theta}^2 \\ \cos \theta \ddot{\theta} - \sin \theta \dot{\theta}^2 & -\sin \theta \ddot{\theta} - \cos \theta \dot{\theta}^2 & 0 & \ddot{l}_x \sin \theta + 2\dot{l}_x \cos \theta \dot{\theta} - l_x \sin \theta \ddot{\theta} + l_x \cos \theta \dot{\theta}^2 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} x \\ y \\ 0 \\ 1 \end{bmatrix} \quad (a12)$$

In the triangle O_1AQ in Figure 14, the constraint equation is given by equation (a13).

$$l_r^2 = (l_0 + l_x)^2 + l_2^2 - 2l_2(l_0 + l_x)\cos(\alpha - \theta) \quad (a13)$$

Dynamic Modeling of Cable Shovel using Kane's Method

Now, the Kane's method is used to develop and analyze the structure of the static and dynamic problems associated with the cable shovel.

In this method, the process begins by defining the general speeds $u_1, u_2, \dots, u_n$ first; then the active force and inertia force, F_i and F_i^* on i th component of the system are calculated as equation (a14) and (a15)

$$F_i^* = -m_i a_i \quad (a14)$$

$$T_i^* = -I_i \varepsilon_i \quad (a15)$$

Then, the general active force $(F_i)_r$ and general active torque $(T_i)_r$ in i th component to general speed u_r are shown in equation (a16), (a17)

$$(F_i)_r = F_i \frac{\partial V_i}{\partial u_r} \quad (a16)$$

$$(T_i)_r = T_i \frac{\partial W_i}{\partial u_r} \quad (\text{a17})$$

Similarly, the general inertia force $(F_i)_r^*$ and general inertia torque $(T_i)_r^*$ in i th component to general speed u_r are shown in equation (a18) and (a19)

$$(F_i^*)_r = (F_i^*) \cdot \frac{\partial V_i}{\partial u_r} \quad (\text{a18})$$

$$(T_i^*)_r = (T_i^*) \cdot \frac{\partial W_i}{\partial u_r} \quad (\text{a19})$$

In above equations, $\frac{\partial V_i}{\partial u_r}$ and $\frac{\partial W_i}{\partial u_r}$ are called the partial velocity of V_i and W_i to general speed u_r .

Summate all general forces to same general speed as equation (a20),(a21)

$$F_r = \sum_{i=1}^m [(F_i)_r + (T_i)_r] \quad (\text{a20})$$

$$F_r^* = \sum_{i=1}^m [(F_i^*)_r + (T_i^*)_r] \quad (\text{a21})$$

The dynamic equation of whole system can be obtain by equation (a22)

$$F_r + F_r^* = 0 \quad (i = 1, 2 \dots n) \quad (\text{a22})$$

In cable shovel model, we assume the general speeds are u_1 and u_2 given by equations (a23) and (a24).

$$u_1 = \dot{\theta}_1 \quad (\text{a23})$$

$$u_2 = \dot{l}_x \quad (\text{a24})$$

Then, the partial derivatives of velocities with respect to u_1 and u_2 in the point with coordinate value of (x, y) in the handle system are given by equations (a22) and (23).

$$\frac{\partial \vec{v}}{\partial u_1} = \begin{bmatrix} -x \sin \theta - y \cos \theta - l_x \sin \theta \\ x \cos \theta - y \sin \theta + l_x \cos \theta \end{bmatrix} \quad (\text{a25})$$

$$\frac{\partial \vec{v}}{\partial u_2} = \begin{bmatrix} \cos \theta \\ \sin \theta \end{bmatrix} \quad (\text{a26})$$

The partial derivatives of the cable moving velocity with respect to the general speeds are also given by equations (a26) and (27).

$$\frac{\partial \dot{l}_r}{\partial \dot{\theta}} = \frac{-l_2(l_0 + l_x) \sin(\alpha - \theta)}{l_r} \quad (\text{a27})$$

$$\frac{\partial \dot{l}_r}{\partial \dot{l}_x} = \frac{(l_0 + l_x) - l_2 \cos(\alpha - \theta)}{l_r} \quad (\text{a28})$$

Motion Equations in Static and Dynamic Conditions

When the handle is rest or moves in low speed, static model can be applied to analyze the system. The static model of cable shovel is shown as equation (a29) and (a30).

$$T_h \frac{-l_2(l_0 + l_x) \sin(\alpha - \theta)}{l_r} + F_x(-x_r \sin \theta - y_r \cos \theta - l_x \sin \theta) + F_y(x_r \cos \theta - y_r \sin \theta + l_x \cos \theta) + mg(x_g \cos \theta - y_g \sin \theta + l_x \cos \theta) = 0 \quad (\text{a29})$$

$$T_h \frac{(l_0 + l_x) - l_2 \cos(\alpha - \theta)}{l_r} + F_x \cos \theta + F_y \sin \theta + N_c + mg \sin \theta = 0 \quad (\text{a30})$$

In the dynamic model, velocity and acceleration of bucket and handle is considered, and the inertia force and moment should be included in the system of equations. Equations on the inertia force and inertia moment are shown in equations (a31) and (a32).

$$M^* = -I\ddot{\theta} \quad (\text{a31})$$

$$\begin{aligned} \vec{F}^* &= -ma = -m \begin{bmatrix} \ddot{x} \\ \ddot{y} \end{bmatrix} \\ &= -m \begin{bmatrix} \ddot{\theta}(-x \sin \theta - y \cos \theta - l_x \sin \theta) + \dot{\theta}^2(-x \cos \theta + y \sin \theta - l_x \cos \theta) - 2\dot{l}_x \sin \theta \dot{\theta} + \ddot{l}_x \cos \theta \\ \ddot{\theta}(x \cos \theta - y \sin \theta + l_x \cos \theta) + \dot{\theta}^2(-x \sin \theta - y \cos \theta - l_x \sin \theta) + 2\dot{l}_x \cos \theta \dot{\theta} + \ddot{l}_x \sin \theta \\ 0 \\ 1 \end{bmatrix} \\ &= -m \ddot{\theta} \begin{bmatrix} -x \sin \theta - y \cos \theta - l_x \sin \theta \\ x \cos \theta - y \sin \theta + l_x \cos \theta \\ 0 \\ 0 \end{bmatrix} - m \dot{\theta}^2 \begin{bmatrix} -x \cos \theta + y \sin \theta - l_x \cos \theta \\ -x \sin \theta - y \cos \theta - l_x \sin \theta \\ 0 \\ 0 \end{bmatrix} - 2m \dot{l}_x \dot{\theta} \begin{bmatrix} -\sin \theta \\ \cos \theta \\ 0 \\ 0 \end{bmatrix} + \ddot{l}_x \begin{bmatrix} \cos \theta \\ \sin \theta \\ 0 \\ 0 \end{bmatrix} \end{aligned} \quad (\text{a32})$$

Substituting (a31) and (a32) to the Kane equation (a22), dynamic model is obtained as (a33) and (a34).

$$\begin{aligned} T \frac{-l_2 \sin(\alpha - \theta)}{l_r} + M^* + F_x^* (-x_g \sin \theta - y_g \cos \theta - l_x \sin \theta) + F_y^* (x_g \cos \theta - y_g \sin \theta + l_x \cos \theta) \\ + F_x (-x_r \sin \theta - y_r \cos \theta - l_x \sin \theta) + F_y (x_r \cos \theta - y_r \sin \theta + l_x \cos \theta) \\ + mg(x_g \cos \theta - y_g \sin \theta + l_x \cos \theta) = 0 \end{aligned} \quad (a33)$$

$$T \frac{(l_0 + l_x) + l_2 \cos(\alpha - \theta)}{l_r} + F_x^* \cos \theta + F_y^* \sin \theta + F_x \cos \theta + F_y \sin \theta + N + mg \sin \theta = 0 \quad (a34)$$

Thus, matrix form of dynamic model of the shovel is shown as the equations (a35).

$$\begin{bmatrix} I + m(x^2 + y^2 + l_x^2) & m(-y) \\ m(-y) & m \end{bmatrix} \begin{Bmatrix} \ddot{\theta} \\ \ddot{l}_x \end{Bmatrix} = F + H(\theta, \dot{\theta}, l_x, \dot{l}_x) \quad (a35)$$

Identification of Shovel Structural Parameters

On the base of dynamic model and static model we can estimate some parameters of system. From the static equation, vector of the unknown parameters is given by equations (a36).

$$q_e = \{mx_o, my_o, m\}^T \quad (a36)$$

Based on static model, we can obtain the equation (a37), (a38) and (a39)

$$A_e q_e = \tau_e \quad (a37)$$

$$A_e = \begin{bmatrix} gc\theta & -gs\theta & l_x c\theta \\ 0 & 0 & -gs\theta \end{bmatrix} \quad (a38)$$

$$\tau_e = \begin{bmatrix} -F \frac{\partial S_F}{\partial \theta} + T \frac{l_2(l_0 - l_x) \sin(\alpha + 2\pi - \theta)}{r} \\ -N - F \frac{\partial S_F}{\partial l_x} - T \frac{l_2 \cos(\alpha + 2\pi - \theta)}{r} \end{bmatrix} \quad (a39)$$

Similar to the static analysis, the unknown parameter vectors in the dynamic equations can be given by equations (a40) to (a43).

$$q_e = \{mx_o, my_o, m, I + m(x^2 + y^2)\}^T \quad (a40)$$

$$A_e q_e = \tau_e \quad (\text{a41})$$

$$A_e = \begin{bmatrix} gc\theta - 2\dot{l}_x\dot{\theta} - l_x\ddot{\theta} & -gs\theta + \ddot{l}_x - l_x\dot{\theta}^2 & gl_xc\theta & \ddot{\theta} \\ \dot{\theta}^2 & \ddot{\theta} & -gs\theta - \ddot{l}_x + ml\dot{\theta}^2 & 0 \end{bmatrix} \quad (\text{a42})$$

$$\tau_e = \begin{bmatrix} -F \frac{\partial S_F}{\partial \theta} + T \frac{l_2(l_0 - l_x) \sin(\alpha + 2\pi - \theta)}{r} \\ -N - F \frac{\partial S_F}{\partial l_x} - T \frac{l_2 \cos(\alpha + 2\pi - \theta)}{r} \end{bmatrix} \quad (\text{a43})$$

Power Calculation for Cable Shovel

Based on the definition by Novak and Larson (1991), the total power required to drive the cable shovel is:

$$P_{total} = \left| \vec{F}_h \cdot \vec{V}_h \right| + \left| \vec{F}_c \cdot \vec{V}_c \right| \quad (5)$$

$\vec{F}_h$ and $\vec{V}_h$ and $\vec{F}_c$ and $\vec{V}_c$ are the respective draw forces and motion velocities of the hoist and crowd cables.

The total work done for the entire trajectory is defined as:

$$P_{work} = \int_0^{t_0} P_{total} dt$$

t_0 is the time duration for executing an entire trajectory.

List of acronyms and abbreviations

q	coordinate value of any point in axis
ϕ	angular displacement; h is motion displacement.
ξ	parameter that includes information of a rotation vector and a point in the axis of rotation.
m	mass of handler
I	inertia of moment to mass center
x, y	the coordinates of mass center in body fixed frame
l_r	distance between A and pivot O
l_0	initial distance between pivot and mass center
α	angle of boom in frame angle
l_x	translation displacement of handle l_2
θ	rotation angle of handle in global frame
l_2	distance between A and suspension point Q
$O_0X_0Y_0Z_0$	global coordinate system
$O_{10}X_{10}Y_{10}Z_{10}$	local coordinate system under static condition
$O_1X_1Y_1Z_1$	local coordinate system under dynamic condition
$(0,0,1)^T$	axis of rotation motion of dipper handle
$[l_0 \cos \alpha_0 \quad l_0 \sin \alpha_0 \quad 0]^T$	selected point in the axis of rotation
$[l_x \cos \theta \quad l_0 \sin \theta \quad 0]^T$	motion transformation
T_h	hoist force of cable
N_c	crown force of handle
M	inertia torque of handle
F_x, F_y	resistance force in x and y direction
${}^m_n A$	transformation matrix of frame m relative to frame n
V	general speeds in Kane equation
v	velocity of points in handle
a	acceleration of points in handle
W	angle velocity of handle
q_e	unknown parameters in equation of parameter estimation.
A_e	coefficient matrix in equation of parameter estimation.
τ_e	force vector in equation of parameter estimation.
k_p	gains coefficient of position in PID control
k_v	gains coefficient of velocity in PID control
k_i	gains coefficient of integral in PID control
q_d, q	desired position and actual position in shovel control scheme
f_d, f	desired force and actual force in shovel control scheme