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Large Mobile Mining Equipment Operating On Soft Ground

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ABSTRACT: The Canadian oil sand deposits of Northern Alberta, operated as large scale bulk handling surface mining operations provide harsh mining conditions in terms of climate and geological characteristics. The ground may be as hard as competent sandstone in winter and as soft as weak clay in summer. The toll on large mobile mining equipment such as > 327 tonne capacity haulers and > 46 m^3 electric and hydraulic shovels is high with structural lives often half that expected. With a move to ever bigger equipment, under the notion that "'bigger is better", the life expectancy of these units is of particular concern. Soft ground conditions cause high rack occurrences that lead to predominantly fatigue failure scenarios, poor stability and incorrect payload evaluations derived from on-board strut pressure information analyzers. This is not solely a concern for mines in the Canadian oil sands, but there exists far reaching implications for surface mining equipment in any soft or weatherable ground operating environments. This paper focuses around the evaluation of payload as affected by soft ground reaction and the effect on frame life for large mobile units, a discussion on the extension of the process to .shovel undercarriage and carbody is made.

I INTRODUCTION

Joseph (2002) introduced the oil sand - equipment interactions program (OsEIP), a joint undertaking by university and consultant researchers, oil sand mining companies and manufacturers of large mobile mining equipment. The focus of the program is an improved understanding of changing oil sand behaviour and associated equipment reactions under static and dynamic conditions. Field tests conducted with > 327 tonne capacity haulers and > 46 m' shovels, including measurement techniques such as passive seismic ground reaction, gave correlations of ground behaviour with equipment duty cycle motion (Joseph, 2002 and Joseph and Hansen, 2002).

In evaluations of truck payload reported by on-board information systems, which report to the mine data management system, which in turn report production Figures to an awaiting plant facility, it was noted that an independent calculation of payload via the sum of forces derived from pressure sensors located at each of the four suspension cylinders (.struts) of the truck were substantially different. Recognizing that the on-board system determines payload at the transition from first to second gear, it was suspected that the effect of soft ground conditions may provide an inadequate stable bearing to allow a consistent measurement of production. Futhermore, on inspection of gear changes, the critical transition was found to commonly occur in the operating pit somewhere between the shovel and pit exit, where little if no road material is added to improve surface characteristics, due to cascading effects resulting in downstream plant interference.

It has been verified by Wohlgemuth (1997), Joseph (2001) and Trombley (2001) that the critical condition for truck frames may be expressed in terms of rack, defined by the difference of the sum of diagonally opposite strut pairs, such that: If L_H, R_F, L_R and R_R denote left front (position 1), right front (position 2), left rear (position 3) and right rear (position 4) struts respectively, then:

$$\text{Rack}= ((L_F + R_R)-(R_F + L_R)j \qquad (1)$$

The life of a truck frame or any rigid mechanical structure such as a shovel carbody or sideframe may be determined in terms of the number of fatigue cycles. Wohlgemuth (1997) and Trombley (2001) expressed an adverse rack cycle as one that exceeded a magnitude of 16 MPa.

2 UNIT CONSIDERATIONS

Although the work conducted by Wohlgemuth (1997) and Trombley (2001) allowed an evaluation of frame life, akin to the procedure adopted by the manufacturer for on-board reporting, it has been recognized that the units of pressure used should be replaced by those of force, due to the front and rear strut pairs being of different active cross sectional area. To enable a similar evaluation of shovel carbody or sideframe performance, force loading at near corner locales is derivable from duty cycle loading; due to the power draw of hoist and crowd systems in the case of cable units (Joseph and Hansen, 2002), or hydraulic cylinder pressure loading In the case of hydraulic shovels. An alternative and more direct approach is to install accelerometers at the corner locales, allowing rack to be determined in acceleration units. Revisiting the suggested units of rack for truck frames, if the load in the truck body is known, then rack may also be determined in acceleration units.

Comparing the use of pressure, force or acceleration units, shows that an adverse rack cycle trigger of 16 MPa is approximately equivalent to an acceleration rack of 1.5g, where g is 9.81 m/s^2. The magnitude of g level will vary, depending on the load in the truck body, however 1.5g is commensurate for an adverse occurrence, with the unit under nominal load. Given values of 16 MPa versus 1.5g, the latter is a more universal descriptor of adverse motion as a performance indicator, recognized by a broader spectrum of people across the industry and may be applied to both truck and shovel structures, and is thus the convention units adopted here.

3 STATIC AND DYNAMIC STRUT RESPONSE

Large tonnage class rear dump haul trucks are typically equipped with four suspension cylinders or struts. The two cylinders at the front of the unit are of a slightly larger diameter than the rear to facilitate greater steering control of the unit, and are designed to accommodate one sixth of a full load each. The rear two cylinders are designed to accommodate one third of a full load each. The front single tire and the rear dual tire arrangements thus allow each tire to be loaded equally under static load on a level bearing surface.

If the bearing surface is level and motion of the unit ensues it is reasonable to expect the unit to pitch (front to back motion depending on whether the unit is accelerating or braking) but not to roll (side to side motion) or rack as described in Equation I.

Where the bearing surface is not level, but undulated as is expected in rough or soft terrain, pitch, roll and rack effects will result.

At rest, any one strut may be overly compressed or extended depending on the ground condition directly below its designated tire set. However as the total load is distributed over all struts regardless of the ground condition and whether one strut is taking its full share of the load, the sum of the load on the struts will remain constant, and be equivalent to the total load. If this sum is determined for the unit at rest, where the truck body is empty, then the tare weight above the struts is determined. Subtracting this from the case where the body is loaded at rest yields the payload.

During motion, where the ground condition is undulated, any one strut may become extended or compressed. In this case the effect of gravity on the load is reduced or enhanced causing a drop or rise in strut pressure. Payload then varies commensurate with this effect and may not give the optimum value triggered by a transition from 1^M to 2^{nd} gear.

4 REVISITING NEWTON'S 2^{NU} LAW

When the truck is at rest, i.e. v = 0 and the truck body is empty, then the tare weight of the unit, FTARE. ^{ma}y be determined via the sum of the weight reactions at each of the 4 struts, as gravity is neither enhanced or reduced due to motion:

$$F_{TARE} = g\sum_{i=1}^{4} m_i \tag{2}$$

It should be noted that FTARE only accounts for the weight directly impinging on the struts and does not represent the true tare weight of the unit, as the weight of tires, rims etc. are not included. This exclusion does not affect the determination of payload as they are below the point of reference.

Similarly, when the truck is at rest and the body contains a load, the loaded weight, F_{LO}AD. of the unit may be determined in the same fashion:

$$F_{LOAD} = g\sum_{i=1}^{4} Mi \tag{3}$$

The payload of the unit may then be determined as a difference between the two values:

$$\sum_{i=1}^{4}(M_i - m_i) = \frac{F_{LOAD} - F_{TARE}}{g} \tag{4}$$

In the case where the truck is not at rest, such that v > 0, whether the unit is empty or loaded, it is unlikely that the dynamic weights are the same as the static determinations. In considering Newton's 2^{nc} law applied to the loaded case:

$$F_{DYNAMIC} = \sum_{i=1}^{4} M_i(g + a_i) \tag{5}$$

The mass, M,, contributing to each strut of the unit effectively does not change, therefore we are witnessing a variation in acceleration, a,, enhancing or reducing the static gravitational constant, g at each strut location. Flexure in the frame of the unit gives rise to different values of a, at each strut on the same unit. The flexure, in turn is also a reflection of the ground conditions giving rise to the phenomenon. FDYNAMIC i* therefore inappropriate for use in determination of the payload.

$$\sum_{i=1}^{4}(M_i - m_i) \neq \frac{F_{DYNAMIC} - F_{TARE}}{g} \quad (6)$$

Given that the zero velocity condition can be identified and that during this period the strut pressures can be monitored to identify equilibrium values. Equation 4 may then be used to establish the payload of the unit.

For any strut, the impinging mass, m, or M, may be found from the v = 0 condition, as per Equation 2 or 3 respectively, depending on whether the focus is the unloaded or loaded case. Applying this then allows (g + a,) to be determined under dynamic conditions as described by Equation 5.

At any instance, the dynamic strut pressure data can be used to describe the rack to which the system is subjected, Equation 1, expressing the individual strut response values in number of g's:

$$\# g_{STRUT\,i} = \frac{(g + a_i)}{g} = \frac{F_{DYNAMIC\,i}}{F_{LOAD\,i}} \quad (7)$$

5 VALUE OF D1MENSIONLESS g UNITS

If the convention of Equation 7 is applied to Equation 1, Equation 8 is defined:

$$Rack = \frac{1}{g}[(a_1 + a_4) - (a_2 + a_3)] \quad (8)$$

The algebraic difference définition for rack resulting in Equation 8 causes the datum value of g present in Equation 7 to be canceled. This is also the case in determining pitch, roll and bounce, Equations 9, 10 and 11:

$$Pitch = \frac{1}{g}[(a_1 + a_2) - (a_3 + a_4)] \quad (9)$$

$$Roll = \frac{1}{g}[(a_1 + a_3) - (a_2 + a_4)] \quad (10)$$

$$Bounce = \frac{1}{g}\sum_{i=1}^{4} a_i \quad (11)$$

In the latter case, bounce is defined as the sum of the difference between the dynamic and static loading states for all frame suspension elements. Some elements may have a reducing and others an enhancing contribution, such that it is possible for individual struts to be adversely extended or compressed, but overall with a near zero bounce effect.

Thus, the value of the dimcnsionless unit is a measure of the relative g effect in rack, pitch roll or bounce, as may be determined.

6 PAYLOAD

A sample of data was acquired as representative ol typical operation for 44 truck cycles in three sets of operating conditions; firm ground, medium ground and soft ground; judged by the degree of rutting and ground deformation that developed during lhe course of the test periods. For the purposes of this paper, this relative description of ground response is correlated to the stiffness - deformation ground response for oil sand developed by Joseph (2002), reproduced in Figure I.

Utilizing the data acquisition system installed by the manufacturer, the collected data consisted of a time log, strut pressures, unit velocity and payload as determined by the on-board system. The front and rear strut cross sectional areas were also supplied as general information from the manufacturer.

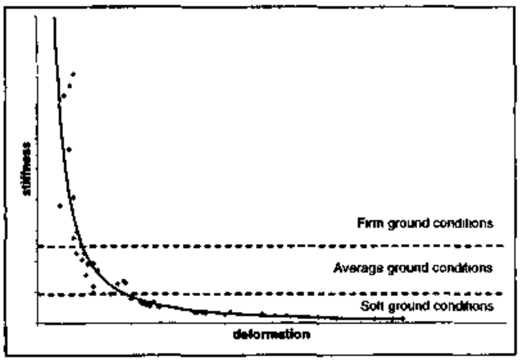

Figure 1 Oil sand ground stiffness - deformation behaviour, after Joseph (2002)

Following the procedure outlined in section 4, the zero velocity condition was identified for both loaded and unloaded states of the unit, allowing the tare weight, F_{TARE}, the loaded weight, F_{LOAD}. and the payload to be determined. This was compared to that reported by the on-board system as illustrated in Figure 2:

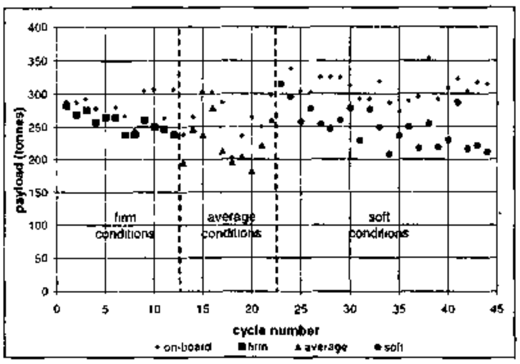

Figure 2 Comparison of on-board to calculated pay load for various operating conditions.

Two trends are immediately apparent from Figure 2. Firstly, that the on-board system consistently over-estimates the payload regardless of the operating conditions and secondly that the softer the ground, the greater the discrepancy between the on-board system and the calculated values. It should be noted that the cycles were recorded consecutively in each of the three operating conditions. This is important when observing the trend of the firm condition data. As the number of cycles advances, the greater the difference between the payload values is observed. This agrees with the earlier work of Joseph (2002). in describing the strain softening behaviour of oil sand with increasing number of cycles. The inability of the truck to record the correct payload on transition from P^1 to 2^{ml} gear, due to the undulating nature of the softer ground is clearly visible. Thus in accordance with Figure 1, oil sand may be primarily described as firm, but with increasing cycles may follow the stiffness - deformation trend, through average to soft behaviour as it is worked by loading.

7 FRAME LIFE

About two thirds of the haul trucks within the 44 cycle data set were observed under adverse operating conditions, such that it was suspected that the frame life may become compromised. To facilitate an example, it was suggested that 1 million adverse cycles exceeding 1.5 g in rack may cause structural failure. The design and operating specifications of the units are such that it is expected that the frames should be good for about 10 years under nominal conditions.

For a typical single truck duty cycle. Figure 3 shows the sum of in-line forces at the struts, allowing the empty, loading and loaded portions of the duty cycle whilst operating in-pit to be clearly seen. A typical duty cycle time is about 20 minutes, with two thirds of the time on poor ground conditions in-pit and the remainder on well constructed roads from pit ramp to either dump or crusher locales. Since the in-pit reactions are the most severe, the analysis is restricted to this area of concern, a 13 minute period.

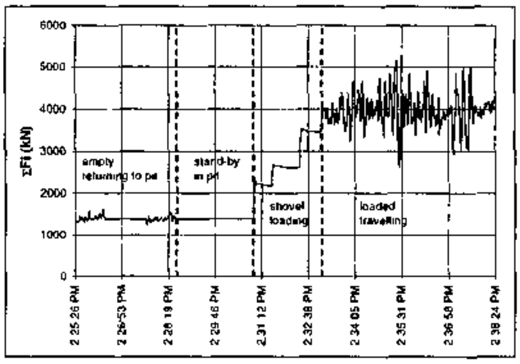

Figure 3 Sum of forces impinging on struts.

The procedure outlined in section 4 was employed to determine strut response in terms of g, with the rack value then determined and illustrated in Figure 4. A comparison of Figures 3 and 4 using the common time base shows that adverse rack is most prevalent when the unit is loaded and in motion.

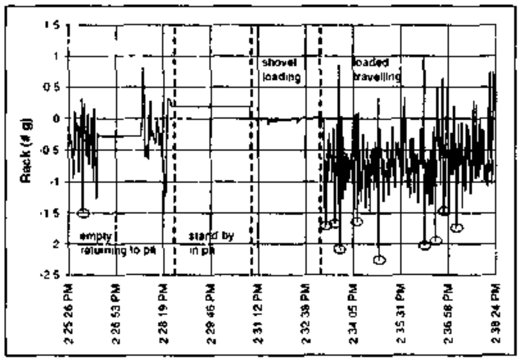

Figure 4 Rack expressed in number of g.

We can define the frame life in simple terms of unit availability (A), utilization (U), rack cycles to failure (F), average number of rack cycles per min (N) and the operating hours per annum (H), given by Equation 12:

$$L = \frac{F}{HAUN} \tag{12}$$

Figure 4 indicates the rack events exceeding 1.5g, totaling 10 events in a period of 13 minutes. If we take typical availability and utilization values of 80% for the unit, for a 350 day operating year, this corresponds to a frame life of:

$$L = \frac{1000000}{350(24)60(0.8)0.8(\frac{10}{(13+7)})} = 6.2\,years$$

If the expected life of the frame is 10 years, then the frame liie estimate is at 62% of nominal.

8 APPLICATION TO SHOVELS

Joseph and Hansen (2002), evaluated the duty loading cycle for a shovel. Further work by the author confirms that duty motions cause the weight distribution on the tracks to adversely rack the carbody when operating at an angle other than perpendicular to (pitch effects) or parallel to (roll effects) the face. The most detrimental rack position was found to occur when the dipper is positioned to dig opposite to, or passes over a sideframe corner in route to or Irom the truck to be loaded.

$$F_{C,H} = \frac{V_{C,H} i_{C,H} \eta_{C,H}}{v_{C,H}} \qquad (13)$$

The extent of loading may be determined from the power draw at the hoist and crowd motors, given that the velocity of motion of the dipper is known, as expressed by Equation 13; where the subscripts C and H refer to the crowd or hoist components, V and i are the voltage and current draws by the hoist or crowd motors, r\ is the efficiency of the system from motor to dipper primarily affected by the reduction gears in the arrangement, and v is the velocity of motion at the dipper. The translation from forces acting at the dipper to the effect at the undercarriage is then merely an exercise in machine geometry relative to snapshots within the duty cycle. However, the short duration of motion for the shovel passing over a corner location, relative to the speed of data acquisition at the hoist and crowd motors, may not permit an adequate evaluation of the load contributions to the undercarriage and a subsequent evaluation of equivalent rack. It is only when the shovel is positioned for some time in the act of corner digging that a clear picture of rack is evident.

Given that the evaluation of rack for trucks, as previously described in this paper was achieved in terms of acceleration units, it seems appropriate that the same units be used to evaluate the rack effect on a shovel undercarriage. As an evaluation of force loading, although simple in principle, may not be practical due to the nature of the machine operation, the least of which is identifying a zero velocity state, a more direct approach for realizing acceleration values is suggested.

The use of accelerometers mounted at machine frame corner locations, such that the rack on the carbody may be evaluated directly for potential damaging occurrences is suggested. Furthermore, if the instruments are located on the sideframes, then an evaluation of the life of these units, which are highly subject to fatigue cracking is made possible.

9 CONCLUSION

It has been shown that firm ground conditions provide a good correlation between on-board payload determination and the evaluation of payload by the method outlined in this paper. However, as ground conditions become softer the difference between the values increases, suggesting that the payloads reported when operating on soft ground may not be sufficiently accurate. In fact, Figure 2 suggests that the difference between reported and actual values may be as much as 100 tonnes, a significant error of up to 45%. It is thus suggested that payload recording while the unit is in motion on soft ground may not be the most reliable reporting of payload.

An approach for estimating frame life, regardless of whether a truck or shovel is being considered has been suggested. An example, using field data from a > 327 tonne unit in operation on soft ground conditions has shown that the life expectancy of a frame can be markedly reduced. In the example, the life of the frame was reduced by 38%.

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