MECHANICAL NET TORQUE OPTIMIZATION OF SUCKER-ROD PUMPING UNITS

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ABSTRACT:

The optimization of the gearbox net torque of a sucker-rod pumping unit is essential to maintain longer operating times. In this paper the four different torques acting on the gear reducer is detailed and calculated for an example case. After the actual net torque is compared with the optimal one. The calculation procedure presented includes the moment of inertia as well.

1 INTRODUCTION

The mechanical net torque acting on the gearbox of a sucker-rod pumping unit consists of four different torque components. The determination of these torques requires the interpretation of a dynamometer survey. As a result of the analysis of the current operating condition the net torque is determined throughout one pumping cycle. The optimization of the net torque has key importance achieving longer operating time of the pumping unit. After the determination of the different torque components the theory of optimization is presented alongside with the required considerations. A dynamometer survey taken on a C-640D-365-168 pumping unit is investigated in this paper and the torques were illustrated in Fig 2-4.

2 DETERMINATION OF THE MECHANICAL NET TORQUE

All four different torque components have to be determined in order to obtain the mechanical net torque. These torques are the rod torque, the counterbalance torque, the rotary moment of inertia and the articulating moment of inertia as seen in Fig. 1. All of these components can be determined from a dynamometer survey, which is a result of a measurement using dynamometers. The flowchart of the determination of the torque components is shown in Fig. 1.

Modern electronic dynamometers register the polished rod displacement and the load acting on the polished rod in time using a uniform time interval throughout the measurement. All the aforementioned torques are functions of the crank angle, which is not recorded in the dynamometer survey. This circumstance, however, necessitates the determination of the crank angles in time, which can be obtained

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from the measured polished rod displacements. To handle this problem a successive approximation was introduced by Kis and Koncz [1].

2.1 ROD TORQUE

For the calculation of the polished rod torque the knowledge of the torque factor – the imaginary lever arm between the crankshaft and the end of the horsehead – throughout the pumping cycle is required, as seen in Eq. 1, which is easily obtained from the crank angles using the linkage lengths of the pumping unit. The structural unbalance is given for every pumping unit by the manufacturer. [2]

\[ T_{\text{Rod}}(\theta) = TF(\theta) \cdot (F(\theta) - SU) \]  

2.2 COUNTERBALANCE TORQUE

The counterbalance torque is obtained using the previously calculated crank angles. The counterbalance torque vs time function is described by Eq. 2. The required maximal counterbalance moment in Eq. 2 can be determined from the specification of the sucker-rod pumping unit and the knowledge of the applied counterweights on the crank arm. The value of \( \tau \) is depending on the pumping unit specification and the applied counterweight position if they are asymmetrically placed on the crank.

![Diagram](Fig. 1)
The rod torque and counterbalance torque calculated for the example case are shown in Fig. 2.

2.3 MOMENT OF INERTIA

These torques are a result of the energy release and dissipation of the parts that are moving at varying speeds. Two different type of moment of inertia is distinguished. The moment of inertia is calculated for the example case is found in Fig. 3.

2.3.1 ARTICULATING MOMENT OF INERTIA

Since some parts of the pumping unit have an alternating movement during the pumping cycle – beam, horsehead, equalizer, etc. – the accelerations and decelerations introduce a new torque type, the articulating moment of inertia. This torque component exists during constant pumping speed. This torque component is directly proportional to the angular acceleration of the beam as seen in Eq. 3.

\[
T_{ia}(\theta) = \frac{12}{32.2} \cdot TF(\theta) \cdot \frac{I_b}{A} \cdot \frac{d^2 \theta_b}{dt^2}
\]

The beam angular acceleration can be obtained using two different methods as seen in Fig. 1. The first method involves the calculation of the crank angles first, then using the calculation procedure proposed by Svinos [3] to get the required beam acceleration vs. time function. The other method determines the beam acceleration by differentiating the measured polished rod displacements twice and then dividing...
them with the length of link A. The value of $I_b$ is only depending on the pumping unit specification. These methods are investigated in detail by Takács and Kis in [4].

2.3.2 **Rotary Moment of Inertia**

Unlike the other moment of inertia the rotary moment of inertia only exists if the crank is turning with a varying speed during the pumping cycle which is likely when a high slip or ultra-high slip prime mover is powering the pumping unit. This torque component is directly proportional to the crank angular acceleration as shown in Eq. 4, which is obtained from the calculated crank angle vs. time points a method using Fourier series. [2]

$$T_{ir}(\theta) = \frac{12}{32.2} \cdot I_s \cdot \frac{d^2 \theta}{dt^2}$$

![Graph showing Gearbox Torques](image)

**Fig. 3**

3 **Optimization of the Net Gearbox Torque**

The optimal gearbox torque is achieved by finding the counterweight configuration which equalizes the net torque better than any other solution. To find the best counterweight placement the effect of the moment of inertia has to be kept in mind. Also if the pumping speed varying during the pumping cycle the change in the counterweights type and position alters the value of $I_s$. Therefore during the calculation of the new counterbalance torque the rotary moment of inertia has to be determined with the new mass moment of inertia. This circumstance makes the process more complex than the previous optimization methods, which neglected the moment of inertia. The actual and optimal net gearbox torque is shown in Fig. 4 for
the example problem. The maximal torque of the gearbox in example case is 640 k in-lbs. At the conditions valid at the measurement the maximal net torque is 602 k in-lbs whereas its value is only 494 k in-lbs at the optimized condition. The life expectancy would be increased with this reduced maximal torque load.

The effect of the asymmetrically placed counterweights on the crank is affecting the counterbalance torque by changing the value of the phase angle. Further investigation is required to successfully describe the magnitude of this change on different pumping units. By taking this correction into account the optimization of these installations will be carried out more precisely.

4 SUMMARY

The theoretical background of the gearbox net torque determination was presented in detail providing solutions to the arising problems. After obtaining the gearbox net torque the method and the most important considerations of the torque optimization was presented to achieve the desired operating condition. A higher level of accuracy is achieved by taking the moment of inertia into account during the optimization procedure. The dependence of the rotary moment of inertia on the counterweight placement on the crank make the optimization more difficult. The actual and optimal conditions were compared for the sample case.
LIST OF SYMBOLS

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Meaning</th>
<th>Unit</th>
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<tbody>
<tr>
<td>A</td>
<td>distance between the saddle bearing and the polished rod</td>
<td>in</td>
</tr>
<tr>
<td>$F(\theta)$</td>
<td>polished rod load at crank angle $\theta$</td>
<td>lbs</td>
</tr>
<tr>
<td>$I_b$</td>
<td>mass moment of inertia of the beam, horsehead, equalizer, bearings, and pitmans, referred to the saddle bearing</td>
<td>lbm ft$^2$</td>
</tr>
<tr>
<td>$I_s$</td>
<td>mass moment of inertia of the cranks, counterweights, and slow-speed gearing, referred to the crankshaft</td>
<td>lbm ft$^2$</td>
</tr>
<tr>
<td>SU</td>
<td>structural unbalance of the pumping unit</td>
<td>lbs</td>
</tr>
<tr>
<td>$T_{CB}(\theta)$</td>
<td>counterbalance torque at crank angle $\theta$</td>
<td>in lbs</td>
</tr>
<tr>
<td>$T_{CB_{max}}$</td>
<td>maximum moment of counterweights and cranks</td>
<td>in lbs</td>
</tr>
<tr>
<td>$T_{ia}(\theta)$</td>
<td>articulating moment of inertia on the gearbox at crank angle $\theta$</td>
<td>in lbs</td>
</tr>
<tr>
<td>$T_{ir}(\theta)$</td>
<td>rotary moment of inertia on the crankshaft at crank angle $\theta$</td>
<td>in lbs</td>
</tr>
<tr>
<td>$T_{net}(\theta)$</td>
<td>net torque on the gearbox at the crank angle $\theta$</td>
<td>in lbs</td>
</tr>
<tr>
<td>$T_{Rod}(\theta)$</td>
<td>rod torque at crank angle $\theta$</td>
<td>in lbs</td>
</tr>
<tr>
<td>$TF(\theta)$</td>
<td>torque factor at the crank angle $\theta$</td>
<td></td>
</tr>
<tr>
<td>$d^2\theta/dt^2$</td>
<td>angular acceleration of the crankshaft</td>
<td>1/sec$^2$</td>
</tr>
<tr>
<td>$d^2\theta/dt^2$</td>
<td>angular acceleration of the beam</td>
<td>1/sec$^2$</td>
</tr>
<tr>
<td>$\tau$</td>
<td>counterweight arm offset angle</td>
<td>degrees</td>
</tr>
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BIBLIOGRAPHY


