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BUCKET WHEEL DRIVE OF COMPACT BUCKET-WHEEL EXCAVATOR TYPE K650.3

POHON KOLESA KOMPAKTNÍHO KOLESOVÉHO RYPADLA K650.3

Abstract

Number of mining machines used in 1970s and 1980s in the middle and east Europe require reconstruction at the present time. Nowadays geologic conditions are so much different from the paste. That is why we look for other solutions with a priority to increase reliability and service life of bucket wheel excavators. Modernizing these machines it is suitable to utilise modern progressive methods.

Abstrakt

Celá řada těžebních strojů, které byly nasazeny v letech 1970 – 1980 ve střední a východní Evropě dnes vyžadují rekonstrukce. Současné geologické podmínky jsou natolik rozdílné, proto jsou hledána další řešení s cílem zvýšit spolehlivost a životnost velkostrojů. Proto je zapotřebí uplatnit současné progresivní technologie při jejich modernizaci.

1 INTRODUCTION

In 1998, company of HÄGGLUNDS DRIVES has used a hydrostatic drive wheel of the thrower machine that was solved using the principal of the height torque hydraulic motor Marathon MB 1600. Later in 2000, PRODECO a.s. company introduced a wheel thrower machine on tramline truck KSS 4000/3500 where the wheel drive had been solved as a direct hydro-drive with low speed hydraulic motor Marathon MB 1600. This company also introduced another wheel excavator KU 800.20 in 2002 that had been driven by two electro-motors over hydrodynamic clutches, cardan shafts and planetary gear-box.

Main aims of a drive wheel compact bucket wheel excavator K650.3 reconstruction are: to decrease mass on peak of the wheel boom arm (this will influence dynamic characteristics of the machine); to reduce the area for the machine drive; to reduce expenses for service and maintenance, etc. This problem was solved using knowledge and experience of hydraulic motors and also using data measured at the main construction point of the bucket wheel excavator.

Compact bucket wheel excavator K650.3 will be irrecoverable in the future. Due to good technical properties – good manoeuvrability, small dimensions and low weight - continuous miner will be used for mining works and also in case when quick transfer of the machine will be necessary.

2 EXECUTED MEASUREMENTS

Many measurements were executed within the frame of grant focused on compact bucket wheel excavator K650.3. Long-lasting measurements were executed on a shaft wheel (measurement of vibration on the compact bucket wheel excavator K650.3). Measurement has been organized and performed by Institute of Applied Mechanics Brno, Ltd. For final assessment, there were used the data measured in the days of 29.–30.6.2004. Four roses of strain gauges (tensiometers) were used for measurements. Each of them was placed along the periphery of a wheel shaft section – ones per angle of 90° (fig. 1). Strain gauges (sensibility ±0,35 %, i.e. ±0,42 Ω). Using the Hook law for planar stress and measured data of deformation, it is possible to acquire a time-dependent curve of the main strain:

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$$\sigma_1 = \frac{E}{1-\mu^2}(\varepsilon_1 + \mu \cdot \varepsilon_2),\ \sigma_2 = \frac{E}{1-\mu^2} \cdot (\varepsilon_2 + \mu \cdot \varepsilon_1) \tag{1}$$

where:

σ_1, σ_2 – main strain $[Pa]$,

ε_1, ε_2 – main relative deformation $[\mu m \cdot m^{-1}]$,

μ – Poisson number $[1]$, $\mu = 0{,}3$

E – Young's elastic modulus in tension $[Pa]$. $E = 2{,}06 \cdot 10^{11}\ Pa$

Determination of shear strain in a perpendicular level to the shaft axis:

$$\tau_0 = \frac{1}{2} \cdot (\sigma_1 - \sigma_2) \cdot sin(2\alpha) \tag{2}$$

where:

α – an angle of two main axes (i.e. main strain directions) and X-axis,

Calculation of the torque based on strain gauge measurement data applied for the shaft:

$$\tau_0 = \frac{M_k}{J_p}\frac{d}{2} = \frac{M_k}{W_k} \quad \Rightarrow \quad M_k = \tau_0 \cdot W_k = \tau_0 \cdot \frac{\pi \cdot d^3}{16} \tag{3}$$

where:

d – shaft diameter $[m]$, $d = 0{,}44\ m$

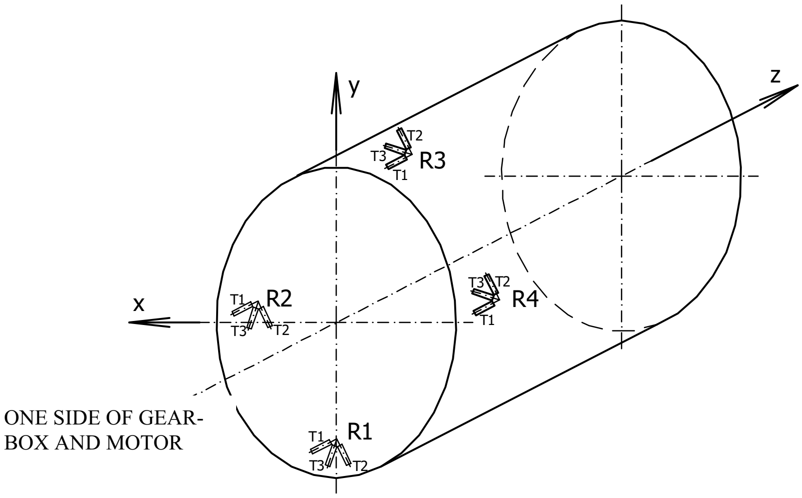

Fig. 1 Strain gauge roses orientation and location [2]

determined from strain gauges measurement

determined from the pressure measured in the hydraulic system

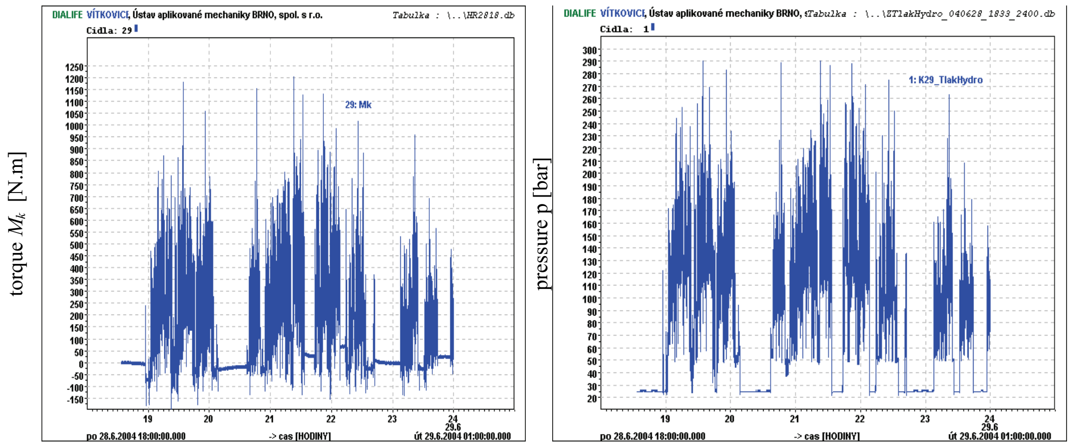

Fig. 2 The measured at 28. 6. 2004 from 18:00 to 24:00 (time-flow data) [6]

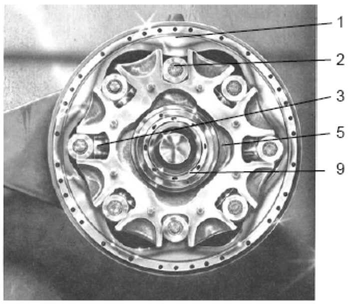

Fig. 3 Section of HÄGGLUNDS radial piston hydraulic motor [4] 1 - curvilinear path, 2 - generating roller, 3 – piston, 5 - rotating block piston, 9 - switchboard

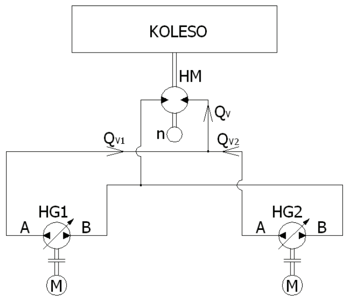

Fig. 4 Simplified schema

3 DIRECT WHEEL HYDRO-DRIVE OF THE EXCAVATOR K650.3

In next I created a new design of the wheel drive. Basic characteristics are:

diameter of the wheel	$d_k = 8{,}8\,m\ (r_k = 4{,}4\,m)$
number of revolutions (for the wheel)	$n_k = (0-4{,}6)\,min^{-1}$
number of buckets	$i = 14$
circumferential force	$F_0 = 217\,kN$

Design of driving hydraulic motor – design of geometric volume hydraulic motor

$$V_{HM} = \frac{2\cdot\pi\cdot M_K}{p_{HM}\cdot\eta_m} = \frac{2\cdot\pi\cdot 955\cdot 10^3}{25\cdot 10^6\cdot 0{,}95} = 0{,}2527\,m^3 = 252650\,cm^3 \tag{4}$$

I selected hydraulic motor MB4000, company of HAGGLUNDS with following characteristics [4] geometric volume $V_{HM} = 251323\,cm^3 \cdot ot^{-1}$

maximum pressure $p_{HM,max} = 350\,bar$

specific torque $M_s = 4000\,N \cdot m \cdot bar^{-1}$

maximum power $P_{HM,max} = 1580\,kW$

number of revolutions $n = 8\,min^{-1}$

weight $m_{HM} = 9800\,kg$

Design of driving hydraulic generator (pump, electromotor, case)

- pumps at parallel inclusion (fig. 4), because there is no single pump with sufficient characteristics to be used in this case,
- two uniform hydraulic generators are selected in this case, therefore it is necessary to buy all the case and equipments (that is why this variant would be more expensive),
- advantage is that the weight of hydraulic generator will be reduced.

I selected hydraulic generator SP355 (2 pieces) [4]

maximum geometric volume $V_{HG,max} = 355\,cm^3$

maximum number of revolutions $n_{max} = 2000\,min^{-1}$

maximum power $P_{HG,max} = 414\,kW$, near n_{max}

maximum torque $M_{HG,max} = 1976\,N \cdot m$, near $V_{HG,max}$, $\Delta p = 350\,bar$

weight $m_{HG} = 237\,kg$

total efficiency $\eta_{HG} = 97\%$

Electromotor

power $P_{ele} = (132 - 400)\,kW$

number of revolutions $n_{ele} = 1760\,min^{-1}$, near $Q = 625\,l \cdot min^{-1}$

Case PEC803

3 doors, 2 pumps at parallel inclusion

maximum volume of tank $V_{N,max} = 857\,l$

Liquid proposal

Company of HAGGLUNDS recommends to use a synthetic HFD oil (waterless synthetic liquid by ISO 6743), which is comparable to Shell Irus Fluids DU oil. Due to high price of Shell Irus Fluids DU oil, producer recommends another mineral oil HM (in accordance with ISO 6743), which is comparable to Shell Tellur S oil.

The main criterion for choice of oil is its price. That is the reason why I choose Shell Tellur S oil (the price is about 31% lower). Higher level of depreciation protection, long-time persistence, oil filterability, corrosion protection and high oxidative duration are guaranteed. Producer recommends value of viscosity 40 $cSt = 40.10^{-6}\,m^2.s^{-1}$ at temperature 40°C [4]. This criterion is satisfied by Shell Tellur S46 oil.

Variance in oil volume as a result of the liquid compressibility is $\Delta V_1 = 2081.10^{-6}\ m^3$, i.e. 0,625 % of total volume V. It means the variance of volume ΔV is 0,6 % of the liquid volume at 100 bars that is a recommender value by hydraulic motor producer [4]. Variance in oil volume as a result of the hosepipe extensibility is $\Delta V_2 = 1452.10^{-6}\ m^3$. Afterwards, the total volume variance is $\Delta V = 3533.10^{-6}\ m^3$.

Deflection angle of hydraulic motor (fig. 5.) appropriate to certain change of volume:

$$\varphi = 2\cdot\pi\cdot\frac{\Delta V}{V_{HM}} = 2\cdot\pi\cdot\frac{3533\cdot 10^{-6}}{252650\cdot 10^{-6}} = 0{,}088\,rad = 5{,}03° \quad (5)$$

Stiffness of a hydraulic circumference (fig. 6.):

$$M_{HM} = c\cdot\widehat{\varphi} \qquad \Rightarrow \qquad c = \frac{M_{HM}}{\widehat{\varphi}} = \frac{955\cdot 10^3}{0{,}088} = 10{,}852\cdot 10^6\,N\cdot m\cdot rad^{-1} \quad (6)$$

Natural frequency:

$$f = \frac{1}{2\cdot\pi}\cdot\sqrt{\frac{c\cdot(I_1+I_2)}{I_1\cdot I_2}} = \frac{1}{2\cdot\pi}\cdot\sqrt{\frac{10{,}852\cdot 10^6\cdot(2{,}662\cdot 10^5 + 2{,}722\cdot 10^6)}{2{,}662\cdot 10^5\cdot 2{,}722\cdot 10^6}} = 1{,}065\,Hz \quad (7)$$

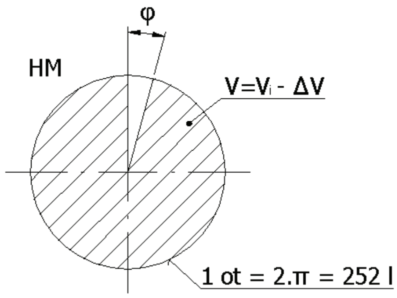

Fig. 5 Simplified schema of hydraulic motor displacement

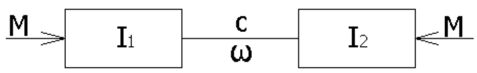

Fig. 6 Simplified schema of circumference

The value of natural frequency has influence on torque amplitude and wheel angular speed fluctuation. Inertia moment I_1 is given by summation of bucket inertia moment (14 pieces, empty), inertia moment of material in buckets, wheel inertia moment and inertia moment of hydraulic motor. The value is $I_1 = 266\ 169\ kg.m^2$. Inertia moment I'_2 is given by summation of inertia moment of hydroelectric generators (2 pieces), inertia moment of clutches (2 pieces) and inertia moment of electric motors (2 pieces). The value is $I'_2 = 21{,}08\ kg.m^2$. Resulting inertia moment: $I_2 = 2{,}722.106\ kg.m^2$.

4 CONCLUSIONS

One of the main advantages of the new design solution proposed in this paper is a 27% (i.e. weight of 4574 *kg*) reduction at the end of a wheel arm, comparing with a current indirect hydraulic drive. Direct hydraulic drive (without gearbox) compact bucket wheel excavator K650.3 is equipped by high-torque hydraulic motor Marathon MB4000 (Hägglunds company), which is installed directly on the shaft wheel. There will be placed rpm sensor on this hydraulic motor. Hydraulic liquid in the circuit (mineral oil Shell tellur S46) is supplied by two regulation hydrogenerators SP355 with maximum volume of 355 cm^3 which are connected with asynchronous electromotors with a power of 315 *kW*. Hydraulic aggregate PEC 803 includes electromotors, hydrogenerators, equipment and tank with capacity of 857 *l*. In hydraulic circuit, there are two accumulator blocks (accumulator with rubber bag) with capacity of 6 *l* to cover the pressure peak in the circuit. Considering the side of security, there is also a hydraulic controlled multi-plate park braking system.

In another point of this paper I deal with operating measurements of bucket wheel shaft deformation and measurements of oil pressure in a hydraulic system. The measured flow-time data were used for determination of torques and consecutive comparison. Comparing the torque determined by calculation based on measured wheel-shaft-deformation data and torque determined by calculation based on measured oil-pressure data I can say that both groups of results are identical (character their time's running) [6]. In next, inner forces and flexural torque of the wheel shaft were calculated in the region where measurements by tensiometers were executed. Considering the graphic presentation of the force to be moved, it is possible to say that it has been a periodical phenomenon that is related to purchase into a cut of each bucket.

I have to recommend recalculation of the natural frequency, because decreasing weight on the peak of wheel, I changed stiffness i.e. I am going to avoid negative resonance.

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