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PTB´S 16.5 MN HYDRAULIC AMPLIFICATION MACHINE AFTER MODERNIZATION

R. Kumme and F. Koehler

Physikalisch-Technische Bundesanstalt (PTB), Braunschweig, Germany

Abstract: This paper describes the 16.5 MN hydraulic amplification machine of PTB after modernization. The control system of the hydraulic amplification machine is described. The amplification factor and the temperature influence of the machine are determined by comparisons with deadweight machines.

Keywords: force standard machine, hydraulic amplification, amplification factor.

1. INTRODUCTION

The 16.5 MN force standard machine with hydraulic amplification was installed in PTB already in 1974 [1]. In 1991 PTB has developed and realized a new control principle for this machine which was running over more than 20 years [2]. After so many years there was the problem that the computer system, the electronic and hydraulic components have to be replaced by new components which are not more available and that the machine should be operated in a full automatic mode. The electronic, hydraulic and computer systems have totally changed so that a new system has to be installed. The new system was installed in 2011. Because of the change of the whole control system the machine was investigated in detail to verify the uncertainty of the machine.

2. PRINCIPLE OF HYDRAULIC AMPLIFICATION

The principle of hydraulic amplification is well known, but there are many influences which have to be taken into account in the hydraulic system and in the control system of the machine to achieve lowest uncertainties of 0.01 %. The general principle of a hydraulic amplification machine is shown in Fig.1. A hydraulic amplification machine is working like a pressure balance. On the deadweight side dead-weights of mass m generate in the gravity field a force F_g which is given by

$$F_g = m \cdot g \cdot (1 - \rho_l / \rho_m) \qquad (1)$$

where
F_g is the force generated in the gravity field in N
m is the mass of the deadweight in kg,
g is the local gravity in m/s^2 ,
ρ_l is the density of the air in kg/m^3 and
ρ_m is the density of the deadweight in kg/m^3.

The gravity force F_g is in equilibrium with a hydraulic force F_1 which is given by:

$$F_1 = p_1 \cdot A_1 \qquad (2)$$

where
F_1 is the force generated by the piston cylinder system
p_1 is the hydraulic oil pressure in Pa which is acting on the piston cylinder system on the deadweight side and
A_1 is the effective area of this piston cylinder system in m^2.

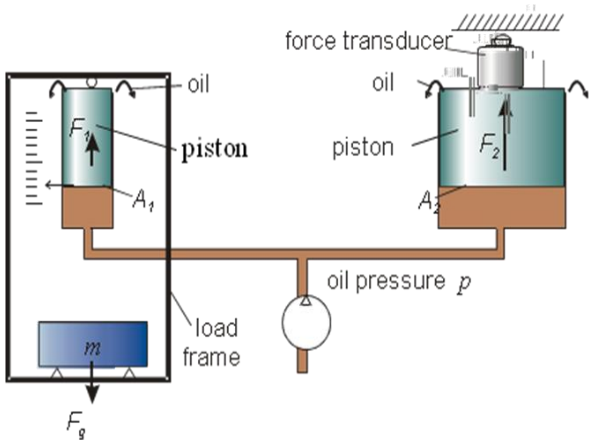

Fig. 1: Principle of a hydraulic amplification force standard machine

By a pressure balance the force F_1 is amplified to a larger force F_2 which is generated on the working side by a larger piston cylinder system with an effective area A_2. The force F_2 is acting on the force transducer to be calibrated in the hydraulic force standard machine and is given by:

$$F_2 = p_2 \cdot A_2 \qquad (3)$$

where
$F_2$is the force generated by the piston cylinder system,
p_2 is the hydraulic oil pressure in Pa which is acting on the piston cylinder system on the working side and
A_2 is the effective area of this piston cylinder system in m^2.

Very important is that the pressure on the deadweight side p_1 is equal to the pressure on the working side p_2 so that the same pressure p can be used on both sides of the pressure balance:

$$p_1 = p_2 = p \tag{4}$$

Only under these conditions the equation for the hydraulic amplification

$$F_1/A_1 = p_1 = p = p_2 = F_2/A_2 \tag{5}$$

is valid and it follows the equation:

$$F_2 = A_2/A_1 \cdot F_1 = Q \cdot F_1 \tag{6}$$

where

Q is the amplification factor given by the ratio of the effective areas of the piston cylinder systems A_2/A_1.

$$Q = A_2/A_1 \tag{7}$$

If F_1 is equal to F_g the following well known equation for a hydraulic amplification machine is valid:

$$F_2 = Q \cdot m \cdot g \cdot (1 - \rho_l/\rho_m) \tag{8}$$

To obtain these conditions the construction of the hydraulic amplification machine and the hydraulic control system has to be realized in such a way that the approximations mentioned before are valid.

Very important is that the pressure on the deadweight and on the working side is equal. To realize this it has to be taken into account that the pressure loss on the pipes on the working side and on the deadweight side is the same. This can be realized by two different methods.

One method is to design the hydraulic system in such a way that the pressure loss on the working side and on the deadweight side is the same. This can be realized by the dimension of the pipes and by the calculation of the pressure loss which has to be verified by pressure measurements. This method was used in the past in PTB´s hydraulic amplification machine [2].

The second method is to reduce the oil flow in the pressure balance to a minimum so that the pressure loss can be neglected. This method is used in the new control system described in this paper.

The main problem in equation 8 is the determination of the amplification factor Q. The classical method is the calculation of the amplification factor from the ratio of effective areas A_2/A_1 of the acting piston cylinder systems. Therefore the dimensions of each piston cylinder system are measured and the effective area for each piston cylinder system is calculated. In a first approximation the effective area A_i of each piston cylinder system i can be calculated as mean of the piston area A_{ip} and cylinder area A_{ic}.

$$A_i = (A_{ic} + A_{ip})/2 \tag{9}$$

The result is the effective area without the consideration of the change of the piston cylinder dimensions depending on the temperature and the pressure. In general the effective area of each piston cylinder system is a function of the temperature T and the pressure p in the piston cylinder system:

$$A_i = A_i(T, p) = = (A_{ic}(T, p) + A_{ip}(T, p))/2 \tag{10}$$

As a consequence also the effective area is a function of the temperature T and the pressure p:

$$Q = Q(T,p) \tag{11}$$

The calculation of the temperature and pressure dependence can be performed by numerical methods like the finite element method (FEM). The experimental verification of the deformation of the piston and the cylinder as a function of temperature and pressure is very difficult.

Easier and more practical is the method to determine the change of the amplification factor by a comparison of the force standard machine with a deadweight machine by using force transfer standards like described in this paper.

3. PTB´S HYDRAULIC AMPLIFICATION MACHINE

Fig. 2 shows the modernized 16.5 MN force standard machine of PTB. Large forces are generated by 4 piston cylinder systems on the right side of the machine (the working side). The force on the working side is realized by the amplification of the force generated by deadweight on the left side of the machine (dead weight side) (Fig. 3).

Fig. 2: PTB’s 16.5 MN force standard machine

Fig. 3: Deadweight side of the 16.5 MN fsm

The principle of the hydraulic control system is shown in Fig. 4. The control is based on the principle described in [2] but there are additional improvements. One improvement is the reduction of the oil flow in the pressure balance to a minimum so that the pressure loss can be neglected. For an explanation this method can be compared in electricity with the well known advantages of the 6 wire technique used for strain gauge bridges compared to the old 4 wire technique. The advantage of the 6 wire technique is that the voltage drop is reduced on the sensing wires because the sensing wires are separated from the supply wires and on the sensing wires is a negligible voltage drop. The same principle is used here for a hydraulic system. The main oil flow is on the working side and therefore there is also the main pressure loss over the hydraulic pipes. To reduce this influence of pressure losses to a minimum in the new hydraulic system of the 16.5 MN machine the oil is directly supplied to the large piston cylinder systems on the working side by an additional pipe connection and the oil flow over the pipe connection used in the pressure balance as a connection between the piston on the deadweight side and the pistons on the working side is reduced to a minimum so that there is a negligible pressure loss on these pipes.

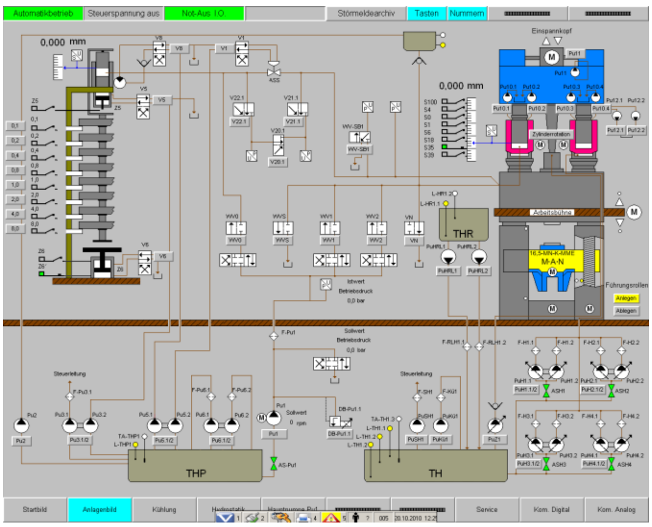

Fig. 4: Hydraulic control system

Another very important point is the control of the position of the piston in the cylinder on the deadweight side. The hydraulic force F_1 is only equal to the gravity force F_g if there is a static equilibrium between the hydraulic force and the deadweight force. To obtain this the position of the piston in the cylinder has to be controlled so that there are no accelerations on the piston cylinder system which will generate additional forces. This is obtained if the velocity of the piston in the cylinder is constant or better zero. In the new control system this is obtained by a high resolution position measurement of the piston position. To obtain this a capacitive displacement sensor is used and a high resolution control circuit with a resolution of 1.000.000 digit.

Further improvements of the electronic control of the new system are that all components of the hydraulic system are monitored on the display. This includes the whole temperature distribution in the machine. The amplification factor of the machine depends not only on the dimensions of the piston cylinder systems at one temperature. Also the temperature change influences the effective area of the piston cylinder systems and contributes to the amplification factor. This was investigated in detail by systematic investigations of the influence of the change of the temperature distribution to the change of the amplification factor of the machine. By a heating and cooling system the temperature of the piston cylinder system on the deadweight side could be increased and decreased and the change of the amplification factor was measured by a comparison with force transducers calibrated in the 2 MN deadweight machine of PTB.

4. DETERMINATION OF THE AMPLIFICATION FACTOR BY COMPARISON WITH DEADWEIGHT MACHINES

After the change of the hydraulic system and the control system of the machine it was necessary to verify and to determine the amplification factor of the machine again. The classical method for the determination of the amplification factor is based on the determination of the amplification factor from the dimensions of the piston cylinder systems by calculating the effective area of the piston cylinder system like described before. The theory can be very complicated and many influences have to be taken into account [3]. Not included in most calculations is the temperature and pressure dependency of the effective area which also contributes to the change of the effective area. This effect was now investigated by comparisons between the hydraulic amplification machine and the 2 MN deadweight machine of PTB. It was found that the temperature control of the machine is very important for the amplification factor and that a temperature change can result in changes of the amplification factor of up to the order of the uncertainty of the machine which is 0.01 %. Also the reproducibility of the machine depends of the temperature control.

The method which was used to determine the amplification factor was as follows. First the masses of the more then 20 years old deadweight stack of the machine were recalibrated and the force on the deadweight side was calculated according equation 1.

With the recalibrated mass values and with the gravity value in the height of the deadweight position which is g= 9,812494368 m/s^2 and with the density ρ_m = 7700 kg/m^3 of the deadweight used in the machine and with the mean air density ρ_l = 1.19 kg/m^3 the acting force F_g on the deadweight side was calculated according equation (1).

In the next step the force on the working side was measured with our best force transducers which are calibrated in the 2 MN deadweight machine with lowest uncertainty of 0.002 %. Therefore up to 2 MN the force F_2 on the working side of the 16.5 MN machine has a direct traceability to the 2 MN deadweight machine. The force measurements on the working side of the machine are performed under different temperature conditions to find the best operation temperature for the machine. It was found that best conditions and the best reproducibility is obtained if the temperature on the deadweight side and on the working side are at room temperature which is in general between 21.5 °C ± 0.5 K. The large piston cylinder systems on the working side are then in a thermal equilibrium with the room temperature and do not change because of the large mass of the big piston cylinder systems and the cooling of the oil temperature. On the measuring side the situation can be different. It was found that with increasing operation time of the machine the temperature can increase to larger values up to 25.5 °C ± 0.5 K. To reduce this temperature drift effect on the deadweight side different changes are performed. Hydraulic valves are exchanged by valves which have a lower temperature increase during operation and finally a oil tank which is in the measuring hall is used as a oil reservoir controlled to room temperature which serves in addition as a cooling system of the piston cylinder on the deadweight side. In this way the temperature on the deadweight side can be reduced to the same temperature 21.5 °C ± 0.5 K like on the working side.

Finally the amplification factor Q was calculated from the equation

$$Q = F_2 / F_1 \qquad (12)$$

where the force F_2 is measured on the working side of the machine by force transducers which are traceable to the 2 MN deadweight machine and the force $F_1 = F_g$ is traceable to the mass calibration and the gravity measurement.

For the determination of the amplification factor one transducer of 1 MN capacity and one transducer of 2 MN capacities were used and the amplification factor was determined for the 1 MN force step and the 2 MN force step.

Transducer	Force Step F_2	Amplification factor Q
1 MN transducer	1 MN	1029,471
2 MN transducer	1 MN	1029,454
2 MN transducer	2 MN	1029,489
Mean amplification factor	1 MN	1029,462
Mean amplification factor	1 MN & 2 MN	**1029,475**

Table 1: Measurement result of amplification factor

The results are shown in table 1. For the further evaluation the mean value of the amplification factor determined for the 1 MN and the 2 MN force step was used.

Q = **1029.475**

The table shows that the deviations from this mean amplification factor are in the order of ± 0.002 %.

Furthermore the dependency of the amplification factor as a function of temperature was evaluated. This was performed by switching of the new cooling system and in addition a heating system was installed on the deadweight side to increase the oil temperature so that on the deadweight side a temperature of 25.5 °C ± 0.5 K is obtained.

The change of the amplification factor was also measured with the force transducers calibrated in the 2 MN deadweight machine. And it was found that a temperature increase on the deadweight side of 4 K results in a reduction of the amplification factor by $1.2 \cdot 10^{-4}$. The reason is that the effective area A_1 on the deadweight side increases because of the thermal expansion of the piston and cylinder with increasing temperature. In summary the amplification factor Q decreases by $3 \cdot 10^{-5}$ by a temperature increase of 1 K on the deadweight side. Therefore it is very important to control the temperature distribution in the hydraulic amplification machine.

In addition based on the basic theory described in the beginning of this paper the effective areas of the piston cylinder systems was also calculated from the measurement data performed more then 20 year ago. In a first approximation the mean diameter of the piston was used 3 mm above the base surface of the piston and the mean cylinder diameter was used 70 mm from the top of the cylinder which is the position of the piston in the height of piston surface if it is in the cylinder. In the next step the effective area A_1 of the piston cylinder system on the deadweight side was calculated as arithmetic men from the piston and cylinder surface. For the working side the effective area A_2 of all 4 piston cylinder are calculated as an arithmetic mean. And finally the amplification factor was calculated according equation 7 from the effective areas. This value is in very good agreement with the experimental value.

	Diameter in m	Area in m^2
Piston A_{1p} (3 mm)	$2{,}897873 \cdot 10^{-02}$	$2{,}897873 \cdot 10^{-02}$
Cylinder A_{1c} (70 mm)	$2{,}899464 \cdot 10^{-02}$	$6{,}602756 \cdot 10^{-04}$
	Effective Area A_1:	$6{,}599133 \cdot 10^{-04}$
Piston A_{2p}	Diameter in m:	Area in m^2
1	$464{,}9853 \cdot 10^{-3}$	0,169811981
2	$465{,}0206 \cdot 10^{-3}$	0,169837765
3	$464{,}9882 \cdot 10^{-3}$	0,169814099
4	$465{,}0071 \cdot 10^{-3}$	0,169827904
Cylinder A_{2c}	Total Piston Area A_{2p}	0,679291749
1	$465{,}0399 \cdot 10^{-3}$	0,169851863
2	$465{,}0628 \cdot 10^{-3}$	0,169868591
3	$465{,}0409 \cdot 10^{-3}$	0,169852593
4	$465{,}0573 \cdot 10^{-3}$	0,169864574
	Total Cylinder Area A_{2c}:	0,679437621
	Effective Area A_2 :	0,679364685
	Amplification factor Q:	1029,4756

Table 1: Measurement result of amplification factor

Based on this former determination of the amplification factor to Q = **1029.475** it was decided to manufacture new deadweight of stainless steel and to adjust the mass to the force values according this amplification factor.

In the 16.5 MN hydraulic force standard machine 8 dead-weights are used with binary mass stack to achieve all forces from 100 kN up to 16.5 MN in steps of 100 kN. To obtain the force values in kN in the first step the necessary deadweight force $F_g = F_1$ is first calculated using the amplification factor Q determined before according the equation:

$$F_1 = F_2/Q \qquad (13)$$

In the next step the necessary deadweight mass (nominal value) was calculated considering the gravity value in the height of the deadweight position which is g= 9,812494368 m/s^2 and the density ρ_m = 7900 kg/m^3 of the new stainless steel dead-weights which will be used in future in the machine and with the mean air density ρ_l = 1.19 kg/m^3 according the equation:

$$m = F_g \,/\, (\, g \cdot (1 - \rho_l/\rho_m)) \qquad (14)$$

F_2 in kN	Q	F_g in kN	nominal value in kg	marking	mass in g	U (k=2) in g
100	1029,475	97,13689	9,90080	1	9900,79	0,03
200	1029,475	194,27378	19,80160	2	19801,58	0,06
400	1029,475	388,54756	39,60319	3	39603,19	0,12
800	1029,475	777,09512	79,20638	4	79206,42	0,24
1000	1029,475	971,36890	99,00798	5	99008,10	0,30
2000	1029,475	1942,7378	198,0160	6	198016,1	0,6
4000	1029,475	3885,4756	396,0319	7	396031,9	1,2
8000	1029,475	7770,9512	792,0638	8	792064,8	2,4

Table 2: Mass of the new stainless steal dead-weights

Finally the new dead-weights are adjusted to the nominal mass values calculated by equation 11 and the mass was determined by the working group mass of PTB with an uncertainty of $3 \cdot 10^{-6}$ (k = 2). The values are summarized in table 2.

5. CONCLUSION

PTB´s 16.5 MN force standard machine with hydraulic amplification was total modernized so that full automatic calibrations can be performed up to 16.5 MN. After the change of the control system the machine was investigated in detail by comparisons with the 2 MN deadweight machine and finally the amplification factor was determined by measurement. The paper presents the method for the amplification factor determination by force measurement. The advantage is that compared to the method calculating the effective area of the piston cylinder systems the method is easier and that other effects like temperature influences or pressure losses in the system are already taken into account.

6. OUTLOOK

In measurements which are performed to determine the amplification factor are performed with the old dead-weights which were recalibrated for this purpose. In the near future PTB has planned to exchange the old deadweight stack by the new stainless steal dead-weights. With the new mass stack again comparison with the 2 MN deadweight machine will be performed for the final validation.

7. REFERENCES

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