WEIGHT SAVING POTENTIALS OF PRESSURE INCREASE IN CYLINDERS OF MOBILE MACHINES KINEMATICS

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ABSTRACT

Many mobile machines operate primarily through the use of boom structures, which account for a large share of energy consumption. The purpose of this work is to investigate if and to what extent the selection of the system pressure level in hydraulically driven boom structures can contribute to the reduction of the moving weight and thus to the reduction of the overall energy demand. The paper focusses on the choice of optimal system-pressure with regard to the cylinder-weight. A calculation method is derived that allows for the analytical calculation of cylinder weight with regard to system pressure to profit from the lightermost cylinders taking into account all sizes of differential cylinders, material properties , safety-factors and cylinder-ratio. It is shown that by specifying the load force, cylinder material, safety factor and area ratio as input parameters the system pressure level that allows the lightest cylinders can be determined. Application of the method shows that there are ideal pressure levels for hydraulic cylinders that are almost independent of the cylinder force and therefore machine size. The results show that a proper pairing of cylinder size and system pressure may have big weight-saving potentials. Taking into account high-strength steel (tensile strength $\sigma = 500\ N/mm^2$) enormeous weight savings may be achieved with rising system pressures.

Keywords: mobile hydraulics, mobile machinery, energy efficiency, payload, boom design, hydraulic optimization, lightweight cylinder

1. INTRODUCTION

Energy optimization in the sector of mobile machines is an important issue. The population of 8..40 ton excacators worldwide is about 1.000.000 and needs more energy than the annually production of offshore windmills worldwide [1]. Together with wheel-loaders excavators have a share of around 60% in mobile construction machineries fuel need [2]. Especially the hydraulic-driven boom structures have –due to a bad payload-to-weight ratio- a big share of the energy consumption in these machines. The improvement of energy efficiency in the operation of boom structures is the subject of numerous research projects which may be classified to optimisations of the diesel motor [3, 4], the task-related design of the kinematics, structural optimisations regarding material and construction [5, 6] the use of compensators [7–10] and the hydraulic drivetrain [11–15]. Within the hydraulic drivetrain special attention is being paid to the efficient operation of the pump(s), the cirquit design and the proper adjustment of the cylinder size to the kinematics. For the optimisation of an energy-efficient system all aspects have to be considered because they are strongly linked.

In mobile machines there is a trend for lifting system pressures that can exemplarily be seen in hydraulic traction drives which today work with pressures up to 450 bar [16, 17]. Along with the idea to increase power density the question arises which are the most important restrictions and if lifting system pressure would be a good solution for making boom structures more powerful. In addition to the approaches stated above an improvement of the payload-weight ratio may be achieved by a proper choice of the system pressure in system design, which may allow for lighter cylinders.

After presenting the basic assumptions and design goals in chapter 2 the mathematical solution is presented in chapter 3. The possibilities and limitations of the methodology are discussed in chapter 4 before drawing a conclusion and giving an outlook.

2. BASIC ASSUMPTIONS AND DESIGN GOALS

One key benefit oy hydraulic actuation systems is the high power density which may further be improved with higher system pressures. Concerning the increase of system pressure the question arises if there are additional effects that may be lead to an increased energy efficiency of the overall system. Reducing the cylinder weight can be advantageous for:

- Reduction of energy consumption (less mass to be lifted + machine weight)
- Reduction of counterweights (cost + machine weight)
- Reduction of machine weight → more power in same weight-class possible

In the very price-sensitive market of mobile machines, these advantages can be exploited if production cost does not rise significantly, so changes in design and construction of hydraulic cylinders are not in the scope of this work.

The guiding question for this work is whether there is a lightest standard cylinder for accomplishing a work task and how this can be calculated depending on the sytem pressure level or, viceversa how system pressure may be appropriately chosen for the lightest (standard) cylinders possible.

2.1. Cylinder design

The design of boom structures is strongly driven by a high payload-to-weight ratio and the restrictions given by the working tasks (working area, movements, sensibility). The basic idea is to reduce the cylinder weight by optimizing the cylinder in stroke, size and rated pressure taking into account the applicable directives[18]. Even though many restrictions like the tasks to fulfil, the available design space, the strength of the mechanical linkage and many more limit the design space for cylinder size, the knowledge of a minimal weight-design may add a design-factor that could be considered in kinematics development.

Cylinders of different sizes may be compared by their work. Cylinders of equal work fulfill the equation:

$$F_c \cdot l_c = const. \qquad (1)$$

with the cylinder force F_c and the length l_c. In other words, a cylinder that is mounted closer to the joint has a shorter stroke but needs to supply more force, what in general leads to a bigger diameter. We will refer to different cylinder sizes fulfilling equation (1) as “equal-work cylinders” according to **Figure 1** (left).

Assuming that cylinder parts that are stroke-independent like hinges have a defined length, the cylinder can be calculated according to its position. The weight of the cylinder, which is the variable to be optimized here, is determined by the design of its components. In the following, we calculate performance-equivalent cylinders in order to find the lightest cylinder depending on system pressure and material for the given application. We will only consider parallel displacements of the cylinders according to Figure 1 (left) here. A change of the cylinder ratio α, which is defined as the ratio between rod and piston area, or any further degrees of freedom in the cylinder design is not considered in this work to get comparable results.

3. CALCULATION OF CYLINDER WEIGHT

Single-acting cylinders consist of a piston, a piston rod, a piston tube, as well as a piston head at both ends and mounting elements on the piston rod and piston bottom. These need be dimensioned during the design process. The cylinder weight then is calculated of its elements according to Figure 1 with the material constants $\rho = 7850\ kg/m\text{^}3$ (steel) and $\rho_{oil} = 880\ kg/m^3$(mineral oil).

$$m_c \cong \rho \cdot \pi \cdot (4 \cdot h \cdot r^2 + l \cdot (2 \cdot r \cdot s - s^2) + l \cdot r_s^2\ ) + oil\ fillling \tag{2}$$

with the cylinder mass m_c, and nomenclature according to Figure 1 (right). Hereby it is considered that the cylinder rod and the end caps have the same height h and the stroke of the cylinder can be approximized as $x = l - h$.

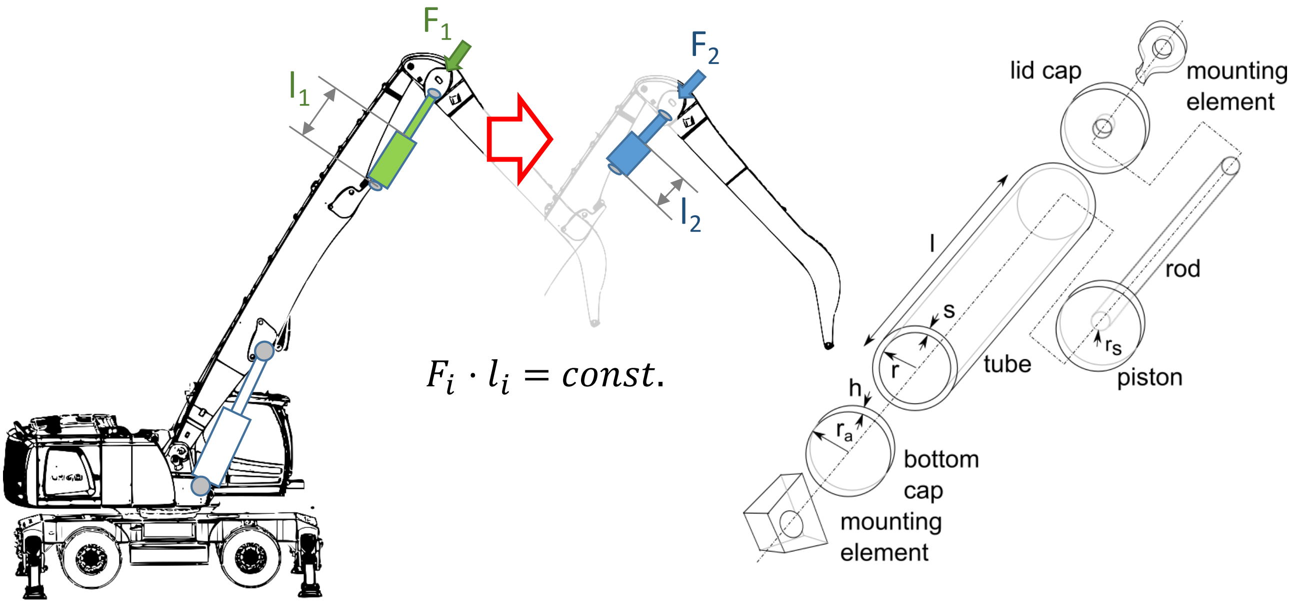

Figure 1: cylinders with $F \cdot l = const$ (left) and cylinder elements (right)

The cylinder rod with radius r_s can be calculated on the basis of the maximum stroke and the acting force F_c on buckling with buckling factor $\frac{l_k}{l} = 0{,}7$ whereas:

$$r_s = \sqrt[4]{\frac{F_c \cdot 4 \cdot \frac{l_k}{l} \cdot sf \cdot l_{0,9}^2}{E \cdot \pi^3}} \tag{3}$$

whereas $l_{0,9}$ is 90% of the rod length and the elastic modulus of the material: E=2.1$E10\ N/m^2$, the security factor which is assumed to be $sf\ = 1{,}5$ for hydraulic cylinders in mobile applications and $l_k/l = 0{,}7$ for the Euler-buckling factor.

The cylinder inner radius r then is calculated for a given maximum system pressure p and force F_c via:

$$r = \sqrt{\frac{F_c}{p \cdot \pi}} \tag{4}$$

Complying the restriction of a given aspect ratio α there may be the need to choose a bigger radius for the cylinder according to:

$$r = \sqrt{\frac{\alpha \cdot r_s^2}{\alpha - 1}} \tag{5}$$

Note: Formulae (4) and (5) show that choosing the right cylinder ratio α may reduce the cylinder diameter without any further optimisation.

From the sizing of rod and inner cylinder diameter the whole cylinder can be calculated for all pressure levels. Dimensioning the piston tube according to DIN 2413:202-04 is used if the ratio of outer and inner diameter is $d_a/d_i < 2$. This may not be applied for higher pressures, therefore an unequal distribution of tension along the wall thickness shall be assumed. We assume linear-elasticity and ideal plasticity along the wall. In comparison of the calculation methods: normal stress hypothesis, shear stress hypothesis or shape change energy hypothesis the latter gives nearly exact values for the maximum pressure p_i at the inner radius r_i. In consequence the outer radius r_a of the cylinder will be calculated with formula (6) according to [19]. In comparison with the standard calculation methods for cylinders ($d_a/d_i < 2$) this formula is accurate in pressure ranges above 500 bar.

$$p_i = \frac{R_e}{sf} \cdot \frac{r_a^2 - r^2}{\sqrt{3} \cdot r_a^2} \tag{6}$$

With $\frac{R_e}{sf} \geq \sigma$ The cylinder cap height h follows the maximum tangential tension

$$\sigma_t \leq \sigma_{t,max} = f \cdot p \cdot \frac{r^2}{h^2} \tag{7}$$

with the factor $f = 0.8$ considering the cap as a panel with clamped edge (welded construction). The same height is applied to the piston and the lid with minor error. The attachments at both sides are considered with the weight of one lid.

With these formulae the lowest cylinder mass for equal-work cylinders under the given restrictions of material and security factor can be calculated depending on the system pressure according to (1) with $s = r_a - r$.

4. RESULTS AND DISCUSSION

Relevant changes of cylinder mass result from variation of force, stroke, maximum pressure level and material. Calculation results are visualized in **Figure 2** for the example of cylinder ratio of $\alpha = 2$.

On the left hand side results for cylinders made of standard steel ($\sigma = 235\ N/mm^2$) are shown, on the right hand side cylinders made of steel with high tensile strength ($\sigma = 500\ N/mm^2$). All diagrams show different cylinder configurations: each line represents possible configurations for a kinematic with given work scenario. Along each line the lightest equal-work cylinders for different pressure levels are shown. The lower diagram shows the corresponding lengths.

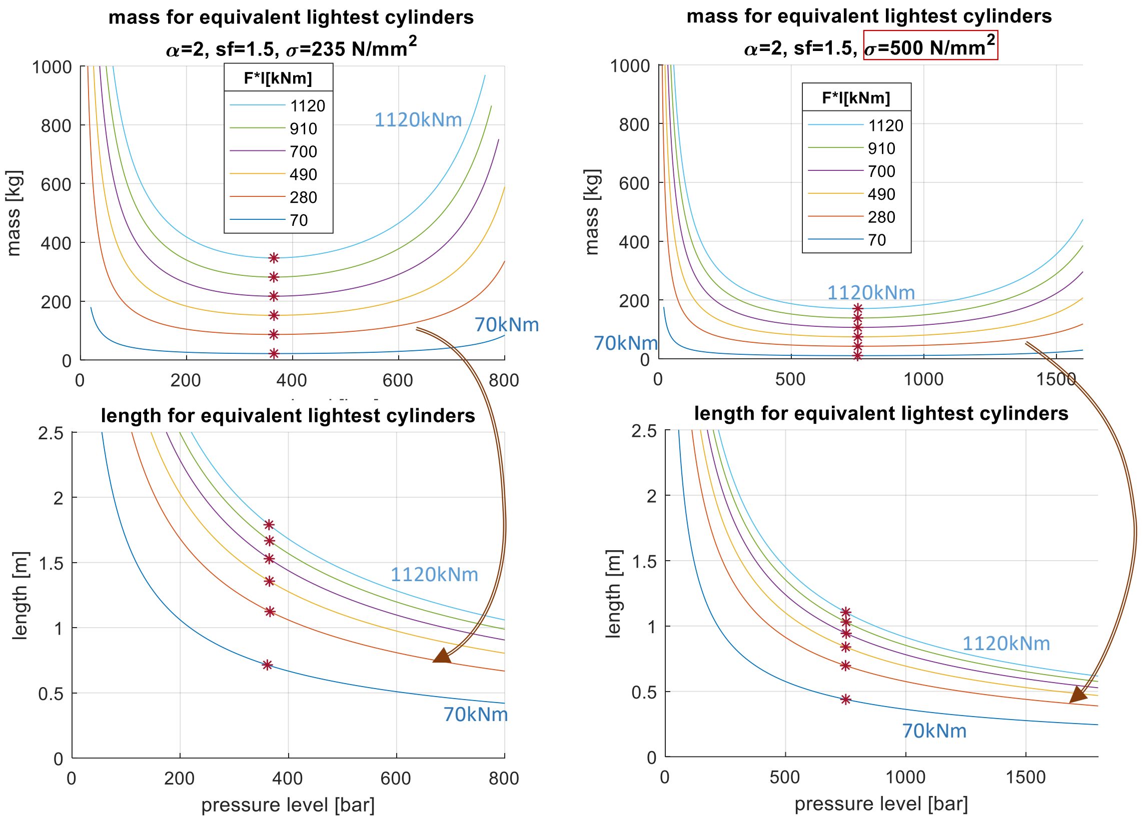

Figure 2: mass and length for lightest equal-work cylinders

The lightest standard industrial steel cylinders with an area ratio $\alpha = 2$ have an ideal pressure level of between 300 and 450 bar with a minimum around 370 bar. The ideal pressure for saving cylinder weight is thereby nearly independent of the machine size. The ideal cylinder length grows with the cylinder power. The calculations show that manufacturing cylinders from high-strength steel halves the weight of a comparable cylinder from standard steel. The main reason for that effect is the shorter stroke. The distribution of weight for equal-work cylinders with 700 kNm (compare Figure 2, purple lines) is shown in **Figure 3**. Weight of piston, caps, attachments and tube grow with rising pressure (even length decreases) whereas weight of rod and oil volume decreases resulting in a defined minimum of weight. As buckling depends on elasticity-module (which is nearly constant for all steel materials) the weight of the piston rod is independent from the strength of the chosen material.

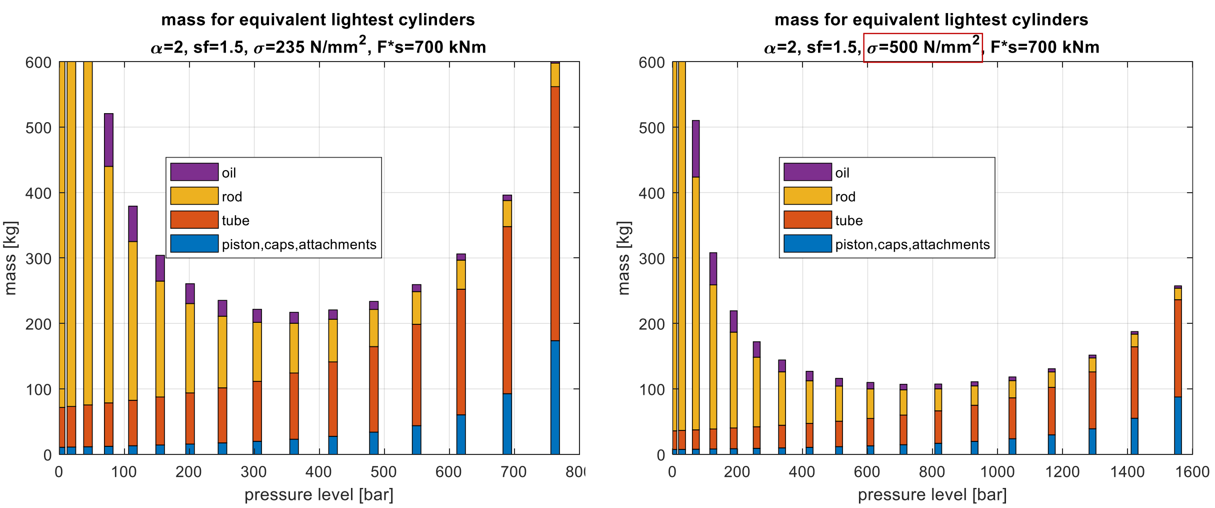

Figure 3: mass distribution for equal-work cylinders with 700 kNm and different material strength

Amongst material and area ratio the minima also depend on the chosen security factor. As a rule of thumb weight scales proportional with the security factor and optimal pressure halves.

Figure 4 shows the points of minimal mass of equal-work cylinders (red asterisks in Figure 2) for cylinders with varying area ratios α. For standard-steel on the left hand and for high performance steel on the right hand side.

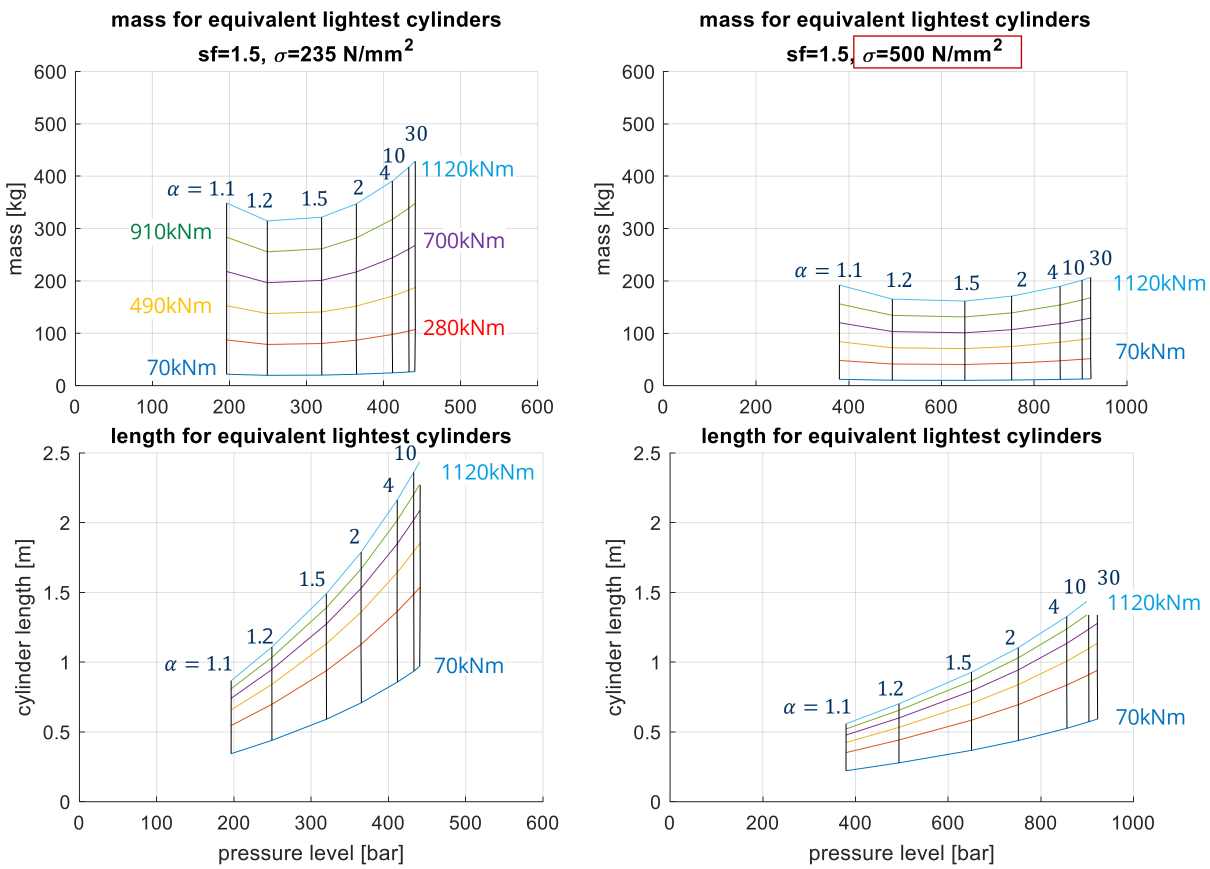

Figure 4: mass and length for lightest equal-work cylinders

The figure shows that weight heavily varies with the strength of the steel but the points of optimal weight stay constant: Cylinders with an area ratio of between $\alpha = 1.2$ and $\alpha = 1.5$ have the best power-to-weight ratio, independent of the material and the cylinder power needed. The smaller the area ratio α is, the lower is the ideal working pressure with respect to cylinder weight. With regard to this outcome the optimal pressure for the lightest cylinders can be plotted over the area ratio.

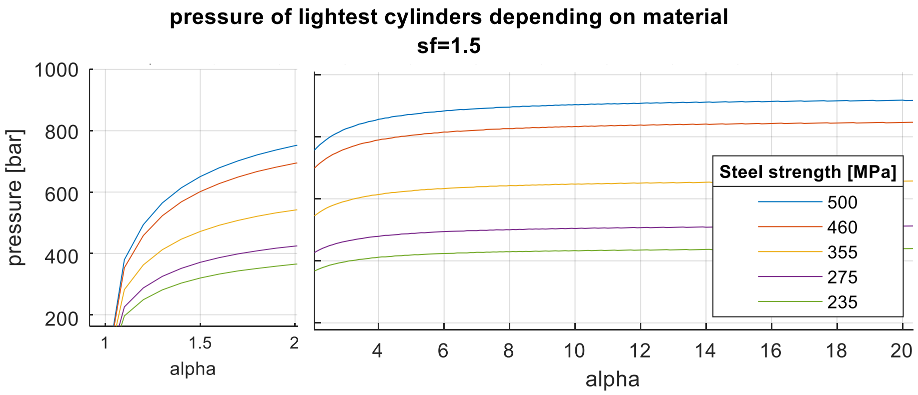

Figure 5: ideal system pressure for minimal-weight cylinders with regard to area ratio α

Figure 5: shows the ideal pressure for the lightest cylinders produced from different materials. As a rule of thumb it can be stated here that the optimal system pressure scales proportional to the strength of the used material.

5. SUMMARY AND CONCLUSION

With respect to hydraulic-driven kinematics the results give insights in how moving mass may be reduced if system pressure and cylinder size are matched to each other. Rising system pressure without using different materials may have the potential to save cylinder weight independent of the machine size in many applications. If high-strength steel is used, enormous weight savings may be realised with rising system pressures. The comparison of different system pressure levels in **Figure 6** exemplarily shows possible weight reductions of about 27% when rising system pressure from 160 bar to 410 bar and twice as large if a high-strength steel is used. The weight-saving potential increases with higher security-factors sf and slightly with bigger area ratios α.

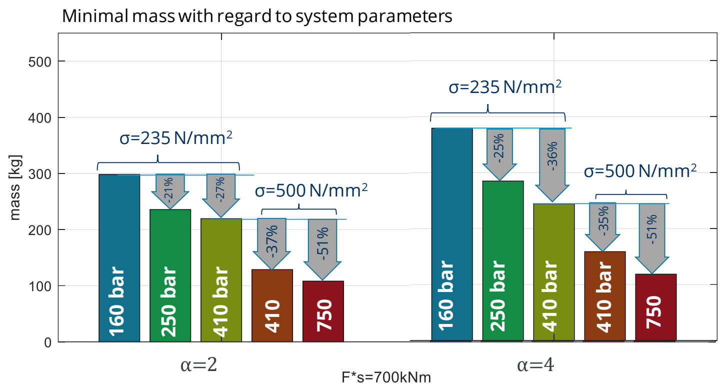

Figure 6: weight savings for cylinders with different materials and pressure levels

The availability of components for increasing the system pressure above 500 bar is currently not (yet) given. However, the investigation shows the potential for significant weight savings. Savings in the weight of the boom are directly reflected in additional handling capacity and or a reduced mass of the counterweight (approximately factor 3). As a result, the power density of the machines may be increased, alternatively machines of the same power get lighter. There might be drawbacks in application: Even if calculations for some practical cases show that simple strengthening of the cylinder mounting points is sufficient, there may be cases where forces become too high. Precision may also be lost if cylinders become very short. Changes in stiffness and natural frequency of high pressure systems should also be further investigated.

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NOMENCLATURE

σ	Tensile strength	N/m^2
α	cylinder ratio	$-$
F	Force	N
l	length	m
$\cdot_c$	cylinder	$-$
ρ	Density	kg/m^3
m	mass	kg

$\cdot_{oil}$	Oil	$-$
h	height	m
s	wall thickness	m
r_i	Inner radius	m
r	(outer) radius	m
x	Cylinder stroke	m
r_s	Rod radius	N
sf	Security factor	$-$
l_k/l	critical (buckling) length	$-$
E	Elasticity modulus	N/m^2
p	Pressure	N/m^2
R_e	Yield strength	N/m^2
F	Force	N
f	Form factor	$-$

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PROGRAMMING EXAMPLE

```
%MATLAB® example code for calculation of cylinder mass
% Author Tobias Radermacher, TU Dresden, Chair of Fluid-Mechatronic Systems
% License: GNU GPL
% download: github.com/boing0815
clc,clear, close all
%calculation of mass for F*l=const
sig=235E6; %N/m^2,max tension,reversibel
Re=sig;
sf=1.5;%security factor
alpha=2; %ratio
leg="";
l_light=[];
force=700; %niveau of force for f*l

f=figure(); hold on; grid on;
% calculation of lines with F*l=const
l=[0.1:0.3:1.6];
skal=0.02:0.02:4; %scaling ofo F*l
for h=1:length(l)
    ls=l(h)./skal;
    fs=force.*skal;
    for k=1:length(skal) %for every variant with f*l constant
        sp(k)=pfl(fs(k)*1000,ls(k),alpha,sf); %neccesary pressure level in bar
        m(k)=zylindermasse_von_r(sig,Re,sp(k)*1E5,fs(k)*1000,ls(k),alpha,sf); %calc cyl. mass; change force to N
    end
    [m_min_f_mal_l(h),ind(h)]=min(m);
    p_min_f_mal_l(h)=sp(ind(h));
    l_light(h)=ls(ind(h));
    f_light(h)=fs(ind(h));

    pl=plot(sp,m);
    uistack(pl,"bottom");
    test=convertCharsToStrings(sprintf('%.0f', force*l(h)));
    leg=[leg,test];
end
title(["mass for equivalent lightest cylinders","\alpha=2, sf=1.5, \sigma=235 N/mm^2"]);
```

```
xlabel('pressure level [bar]');
ylabel('mass [kg]');
xlim([0,1800])
ylim([0,600])
%marking min.
plot(p_min_f_mal_l,m_min_f_mal_l,'*')
%scale position for paper size
f.Position(3:4) = [400 300];
%make legend
le=legend(leg(1,2:end));
le.String=flip(le.String); %flip legend
le.Position = [0.4 0.5 0.1 0.2];
saveas(f,"lightest cylinders_mass.fig");
saveas(f,"lightest cylinders_mass.emf");

function p=pfl(F,l,alpha, sf)
    %calc of p for defined area ratio
    E=2.1E10; %E-module in N/m^2;
    %go for it
    % min. radius der of rod against buckling
    % buckling length 90%
    r_s=(F*sf*2*(0.9*l)^2/(E*pi^3))^(1/4);
    r_i=sqrt(alpha/(alpha-1))*r_s; %area ratio
    p=F/(r_i^2*pi);
    p=p/1E5; %in bar
end

function [m] = zylindermasse_von_r(sig,Re,p,F,l,alpha,sf)
%change mass to 1E-12, if useless geometry
% sig, Re in N/m^2, p in N/m^2, F in N, l in m

%input:
    %mat. properties
    rho=7850;%kg/m^3, dens. steel
    rho_oel=880; %kg/m^3
    E=2.1E10; %E-module in N/m^2
%calculation
    %rod radius
    r_s=(F*sf*2*(0.9*l)^2/(E*pi^3))^(1/4);
    %piston radius
    r_i=sqrt(F/(p*pi));
    %wall thickness (fat, plasticity=yes)
    r_a=r_i/sqrt(1-p*sf*sqrt(3)/sig);
    s=r_a-r_i;
    %wall thickness (thin, elastic):
    %s=r_i*p/(sig-p)
%cap heigth
    h=(r_i+s)*sqrt(0.8*p/(sig/sf));
%finally: the result:
    %mass:
    m_platten=rho*pi*(4*h*r_a^2);
    m_rohr=rho*pi*(2*r_a*s-s^2)*l;
    m_stange=rho*pi*l*r_s^2;
    mz=m_platten+m_rohr+m_stange;
    m_oel=(l-h)*r_i^2*pi*rho_oel;
    m=mz+m_oel;
    if s>0.1 || s<=0 %delete useless wall values
        m = NaN;
    end
end
```