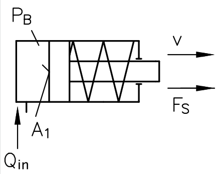
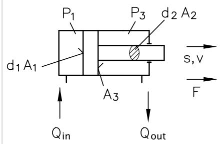
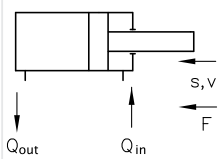
Formulas and units

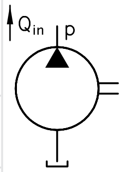
Hydraulic system and circuit design is limited only by the creativity of the application engineer. All basic circuit design begins with the ultimate actuator functions in mind however.
The most important condition for this is the definition or specification of relevant consumer variables, such as the loads (load forces, load torques or turning torques), motion functions (travel, speeds, rotational speeds, timing) etc.
The following formulas and tables are intended to serve as guideline only and should help with the planning of your hydraulic system.
Other factors that have an influence on the choice of hydraulic systems and components include noise emission values and thermal budget considerations.
The following formulae and tables are non-binding and are intended to make producing the rough design for a hydraulic system easier.

Equipment	**Formulas and description**		
General information	Basic equations (static, without any loss)		
	$Q=\frac{V}{t}$ $V=A \cdot s$ $F=p \cdot A$ $p=\frac{F}{A}$ $Q=A \cdot v$ $M=\frac{V \cdot p}{2\pi}$ $v=\frac{s}{t}$	force volume A: Q: v: V: torque s: M:	Force Pressure Area Flow Speed Volume Time Travel (stroke) Torque

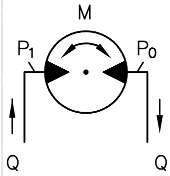
Equipment	**Formulas and description**		**Symbol**
Hydraulic cylinders ■ Single acting	$A[mm^2]=\frac{\pi}{4}\,d^2[mm]$ $v\left[\frac{m}{s}\right]=\frac{s[mm]}{1000t[s]}$ $F_S[N]=-0,1 \cdot p_B[bar] \cdot A[mm^2]$ $p_B[bar]=\frac{-10F_S[N]}{A_1[mm^2]}$ $Q_{in}[lpm]=0.06 \cdot A[mm^2] \cdot v\left[\frac{m}{s}\right]$	d: piston diameter [mm] A: piston area [mm²] F_S: force [N] p_B: operating pressure [bar] v: Piston speed $\left[\frac{m}{s}\right]$ Q_{in}: inflow [lpm] s: stroke [mm] t: time [S]	
■ Double acting	**Extending** Basic equations (balance of forces): $A_1=\frac{\pi}{4}\,d_1^2\approx 0,78d_1^2$ $A_3=\frac{\pi}{4}(d_1^2-d_1^2)$ $p_1 \cdot A_1=p_3 \cdot A_3-F$ $p_1=\frac{1}{A_1}(p_3 \cdot A_3-F)$ $Q_{zu}=A_1 \cdot v$ $Q_{ab}=A_3 \cdot v$	Simplified: $p_1[bar]=\frac{p_3[bar] \cdot A_3[mm^2]-10F[N]}{A_1[mm^2]}$ $F[N]=\frac{-p_1[bar] \cdot A_1[mm^2]+p_3[bar] \cdot A_3[mm^2]}{10}$ p_3 is the result of back pressure from pipes and valves for Q_{out} Attention: note possible pressure intensification!	
	Retracting Basic equations (balance of forces): $p_1 \cdot A_1=p_3 \cdot A_3+F$ $p_3=\frac{1}{A_3}(p_1 \cdot A_1-F)$ $Q_{zu}=A_3 \cdot v$ $Q_{ab}=A_1 \cdot v$	Simplified: $p_3[bar]=\frac{p_1[bar] \cdot A_1[mm^2]-10F[N]}{A_3[mm^2]}$ $F[N]=\frac{p_1[bar] \cdot A_1[mm^2]-p_3[bar] \cdot A_3[mm^2]}{10}$ p_1 result of back pressure from pipes and valves for Q_{out}	
	A_1: piston area [mm²] A_3: rod side area [mm²] d_1: piston ∅ [mm] d_2: rod ∅ [mm] F: force [N]	Q_{in}: inflow [lpm] Q_{out}: outflow [lpm] p_1: pressure, piston side [bar] p_3: pressure, rod side [bar] s: stroke, travel [mm]	

Equipment	Formulas and description			Symbol
Hydraulic pumps / hydraulic motors	Basic equations:	$\Delta p = p_1 - p_o$ [1]	Simplified:	Hydraulic pump, Hydraulic motor (see symbols below)
	Geometric volume per revolution (piston pumps):	$V = A \cdot h$	$V[cm^3] \approx \frac{A[mm^2] \cdot h[mm]}{1000}$	
	Flow:	$Q = V \cdot n$	$Q[lpm] \approx \frac{V[cm^3] \cdot n[min^{-1}]}{1000}$	
	Middle torque:	$M = \frac{V \cdot \Delta p}{2\pi}$	$M[Nm] \approx \frac{V[cm^3] \cdot \Delta p[bar]}{62}$	
	Power:	$P_{hydr} = \Delta p \cdot Q$	$P_{hyd}[kW] \approx \frac{\Delta p[bar] \cdot Q[lpm]}{612}$	
	Power consumption (pump):	$P_{mech} = \frac{\Delta p \cdot Q}{\eta_T} = \frac{M \cdot 2\pi n}{\eta_T}$ [2]	$P_{drive}[kW] \approx \frac{\Delta p[bar] \cdot Q[lpm]}{500}$	
	Power rating (motor):	$P_{max} = \Delta p \cdot Q \cdot \eta_T = M \cdot 2\pi n \cdot \eta_T$ [2]	$P_{out}[kW] \approx \frac{\Delta p[bar] \cdot Q[lpm]}{740} \approx \frac{M[Nm] \cdot n[min^{-1}]}{12000}$	
	V: displacement [cm³] A: effective piston area [mm²] h: double stroke [mm] n: rev. rating [rpm] M: middle torque [Nm] p: pressure [bar] Δp: effective pressure [bar] Q: flow [lpm] P_{hydr}: hydraulic performance [kW] P_{mech}: mechanical performance [kW] η_T : total efficiency (including volumetric and mechanical losses)		Guideline: A power rating of 1 kW for the drive is necessary to achieve a delivery flow of Q = 1 lpm with operating pressure p = 500 bar!	

Hydraulic pump



Hydraulic motor

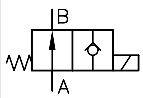


[1] p_o result of back pressure from pipes and valves
[2] incl. degree of efficiency $\eta_T \approx 0.82$

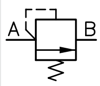
Equipment	Formulas and description	Symbol
Valves Directional valves Pressure valves Metering valves Check valves	Losses of pressure by streaming fluid The pressure loss in hydraulic systems consists of: ■ Back pressure of valves ■ Back pressure of pipes ■ Back pressure due to geometric shape (elbows etc.) Pressure losses Δp in the valves that are caused by the flow of fluid can be found in the Δp-Q characteristics of the relevant documentation. For the purposes of an initial rough design, a performance loss of approx. 20... 30% in the overall control system can generally be expected.	Examples: (see symbols below)

Examples:

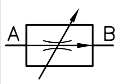
Directional valve



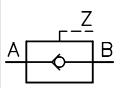
Pressure limiting valve



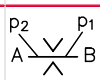
Flow control valve



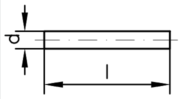
Releasable check valve



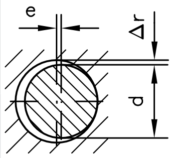
Equipment	Formulas and description		Symbol
Orifices (ideally, sharp edged) e.g. orifice inserts type EB; by-pass check valves type BC, BE	Basic equation: $Q \approx \alpha \cdot \frac{\pi}{4} d^2 \cdot \sqrt{\frac{2\Delta p}{\rho}}$ Q: flow [lpm] Δp: back pressure between A and B [bar] d: orifice diameter [mm]	Simplified: $Q \approx 0.55 d^2[mm] \cdot \sqrt{\Delta p[bar]}$ $d \approx 1.35 \cdot \sqrt{\frac{Q[lpmin]}{\sqrt{\Delta p[bar]}}}$	(see symbol below)

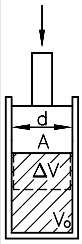


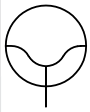
Equipment	Formulas and description		Symbol
	ρ: density (approx. 0.9 g/cm³) α: flow coefficient (approx. 0.78)	$\Delta p \approx \left(\frac{1.82 \cdot Q[lpm]}{d^2[mm]}\right)^2$	

Equipment	Formulas and description		Symbol
Pipes / hoses	The diameter of pipes and/or hoses should be selected in such a way that back pressure is minimized. Basic equations: $Re = \frac{v \cdot d}{\nu}$ $\lambda_R = \frac{64}{Re}$ $\Delta p = \lambda_R \cdot \frac{l}{d} \cdot \frac{\rho}{2} v^2$		
	λR: pipe back pressure coefficient Δp: back pressure [bar] l: pipe length [m] d: pipe diameter [mm] ν : cinematic viscosity [mm²/s] Q: flow [lpm] Re: Reynolds No. (< 2300) ρ: density (approx. 0.9 g/cm³) v: flow velocity $\left[\frac{m}{s}\right]$	Simplified: $Q[l/min] \leq 0,108 \cdot d[mm] \cdot \nu \left[\frac{mm2}{s}\right]$ $d[mm] \geqslant \frac{9.2 \cdot Q[lpm]}{\nu \left[\frac{mm2}{s}\right]}$ $\frac{\Delta p}{l}\left[\frac{bar}{m}\right] \approx \frac{6,1 \cdot \nu \left[\frac{mm2}{s}\right] \cdot Q\left[\frac{l}{min}\right]}{d^4[mm]}$	

Equipment	Formulas and description		Symbol
Back pressure due togeometric shape (elbows etc.)	Basic equations: $\Delta p = \varsigma \cdot \frac{\rho}{2} v^2$ $v = \frac{Q}{A} = \frac{4Q}{\pi d^2}$		
	90° elbow	ξ = 0,15	
	straight pipe fitting	ξ = 0,5	
	elbow fitting	ξ = 1,0	
	Simplified: $\Delta p[bar] = 2,2 \cdot \xi \cdot \frac{Q^2[lpm]}{d^4[mm]}$		
	Δp: back pressure [bar] ξ : back pressure coefficient ν : cinematic viscosity [mm²/s] d: pipe diameter [mm] ρ: density (approx. 0.9 g/cm³)		

Equipment	Formulas and description		Symbol
Leakage losses (by concentric (e = 0) and eccentric gaps)	Basic equation: $Q_L = \frac{\pi \cdot d \cdot \Delta r^3}{12 \cdot \nu \cdot \rho} \cdot \frac{\Delta p}{l} (1\ 1,5 \cdot \varepsilon^2)$	Simplified: $Q_L = 1848 \cdot \frac{d \cdot \Delta r^3}{\nu} \cdot \frac{\Delta p}{l} (1\ 1,5 \cdot \varepsilon^2)$ $\varepsilon = \frac{e}{\Delta r}$	
	e: eccentricity [mm] Δr: gap [mm] Δp: Pressure difference [bar] d: diameter [mm] ν : cinematic viscosity [mm²/s] l: gap length [mm] ρ: density (approx. 0.9 g/cm³)		

Equipment	Formulas and description		Symbol
Volumetric losses (due to pressure increase)	Basic equation: $\Delta V = \beta_p \cdot V_o \cdot \Delta p$ with $\Delta p = p_2 - p_1$		$F = \Delta p \cdot A$ 
	p_1: pressure, start [bar] p_2: pressure, end [bar] V_o: initial volume [l] Δ: ϑ volume alternation [l] β_P: compressibility	Simplified: $\Delta V = 0{,}7 \cdot 10^{-4} \cdot V_o \cdot \Delta p$ $\left(\text{with}\, \beta_p \approx 0.7 \cdot 10^{-4} \frac{1}{\text{bar}}\right)$	
Volumetric losses (due to temperature rise)	Basic equation: $\Delta V = \beta_T \cdot V_o \cdot \Delta \vartheta$ mit $\Delta \vartheta = \vartheta_2 - \vartheta_1$		
	ϑ_1: temperature, start [°C] ϑ_2: temperature, end [°C] $\Delta \vartheta$: temperature, difference [K] V_o: initial volume [l] ΔV: volume alternation [l] β_T: expansion coefficient	Simplified: $\Delta V = 0{,}7 \cdot 10^{-3} \cdot V_o \cdot \Delta \vartheta$ $\left(\text{mit}\, \beta_T \approx 0{,}7 \cdot 10^{-3} \frac{1}{K}\right)$	
Pressure increase caused by temperature rise (without volumetric compensation)		$\Delta V = 0{,}7 \cdot 10^{-4} \cdot \Delta p = 0{,}7 \cdot 10^{-3} \cdot \Delta \vartheta$ i.e. $\Delta \vartheta \approx 1\text{K} \Leftrightarrow \Delta p \approx 10\text{bar}$	
	Note: A temperature rise of trapped oil volume will cause a pressure increase! (i.e. a pressure limiting valve will be required sometimes) **Guideline:** The pressure will rise by approx. 10 bar for 1 K of temperature increase.		

Equipment	Formulas and description		Symbol
Hydraulic accumulators Pressure alternations, isotherm (slow) adiabatic (quick)	Hydraulic accumulators are intended for the supply of pressurized fluid during sudden demands (quick, adiabatic pressure alternations), compensation of leakage losses or to dampen oscillations (slow, isotherm pressure alternations).		
	Basic equations:	$p_1 = 1{,}1 \cdot p_o$	
	isotherm (slow)	$\Delta V = V_1 \cdot \left(1 - \frac{p_1}{p_2}\right)$	
	adiabatic (quick)	$\Delta V = V_1 \cdot \left(1 - \left(\frac{p_1}{p_2}\right)^{0{,}71}\right)$	
	p_o: filling pressure for the gas [bar] p_1: lower operating pressure [bar] p_2: upper operating pressure [bar] V_1: initial volume [l] ΔV: volume alternation [l]		

Equipment	Formulas and description
Cavitation	Approx. 9 % (volumetric) air are solved in oil at atmospheric pressure. There is the danger of bubble cavitation during atmospheric pressure below 0,2 bar. These situations can occur, accompanied by sudden noise, during suction process of pumps and cylinders as well as at extreme throttle sections. The hydraulic components where this occurs will show increased wear.

Equipment	Formulas and description	
Thermal level Dissipation power and oil temperature	The hydraulic power losses in a hydraulic system result in a temperature rise of the fluid and the equipment which is partly radiated to the surroundings via the surface of the system. They roughly amount 20 - 30% of the induced performance. The induced and the radiated heat will balance at some point after the warm-up of the system.	
	Basic equations: $P_V = 0{,}3 \cdot P_{hydr}$ $\vartheta_{Ölmax} \approx \vartheta_{Umg} + C \cdot \frac{P_V}{A}$	
	Surface with unhindered circulation c ≈75 Surface with bad circulation c ≈ 120 with fan (v ≈ 2 m/s) c ≈ 40 Oil/water radiator c ≈ 5	Simplified: $\vartheta_{Ölmax} \approx \vartheta_{Umg}\ C \cdot \frac{0{,}3 \cdot P_{hydr}[kW]}{A[m^2]}$
	P_v: performance loss, transformed in heat [kW] P_{hydr}: hydraulic performance [kW] ϑ_{oilmax}: max. fluid temperature [°C] ϑ_{amb}: ambient temperature [°C] A: surface of the system (tank, pipes etc.) [m^2]	

Conversion table

Nomenclature	Codings	Unit	≈	Factor X	Unit
Pressure	p	$1\frac{N}{mm^2}$	≈	10	bar
		1 MPa	≈	10	bar
		$1\frac{kgf}{cm^2}$	≈	1	bar
		1 psi	≈	0.07	bar
Force	F	$1\frac{kg \cdot m}{s^2}$	=	1	N
		1 lbf	≈	4.45	N
Length, travel, stroke	l, s, h	1 in	≈	25.4	mm
		1 ft	≈	304.8	mm
Torque	M	$1\frac{kg \cdot m^2}{s^2}$	=	1	Nm
Performance	P	1 PS, 1 hp	≈	0.74	kW
Area	A	1 ft^2	≈	92903	mm^2
		1 in^2	≈	645.16	mm^2
Volume	V	1 ft^3	≈	28.92	l
		1 in^3	≈	$1.64 \cdot 10^{-2}$	l
		1 UK gal	≈	4.55	l
		1 US gal	≈	3.79	l
Temperature	T, ϑ	5 (°F-32)/9	≈	1	°C
Mass	m	1 lb	≈	0.45	kg
Cinematic viscosity	ν	1 cST	=	1	$\frac{mm^2}{s}$