Hydraulic actuation system design and computation

1 clear about the design request to carry on the operating mode analysis.

When design hydraulic system below, first should be clear about the question, and takes it as the design basis.

Main engine use, technological process, overall layout as well as to hydraulic gear position and spatial size request; The main engine to the hydraulic system performance requirement, like the automaticity, the velocity modulation scope, the movement stability, the commutation pointing accuracy as well as the request which to the system efficiency, warm promotes; Hydraulic system working conditions, like temperature, humidity, vibration impact as well as whether has situation and so on corrosiveness and heat-sensitive material existence.

In in the above work foundation, should carry on the operating mode analysis to the main engine, the operating mode analysis including the movement analysis and the mechanical analysis, also must establish the load and the operating cycle chart to the complex system, from this understood the hydraulic cylinder or the oil motor load and the speed change as necessary the rule, below makes the concrete introduction to the operating mode analysis content

1.1 movements analyses

The main engine functional element according to the technological requirement movement situation, may use the displacement circulation chart (L—t), the speed circulation chart (v—t), or the speed and the displacement circulation chart indicated, from this carries on the analysis to the movement rule.

1.1.1 displacements circulation attempts L—t

The chart 1.1 is the hydraulic press hydraulic cylinder moves the circulation chart, the y-coordinate L expression piston moves, the x-coordinate t expression starts from the piston to the reposition time, the rate of curve expression movement of plunger speed.

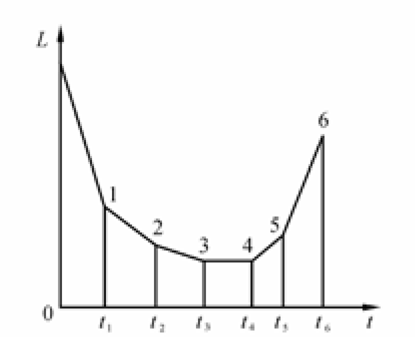

Chart 1.1 displacements circulation chart

1.1.2 speeds circulation chart v—t (or v—L)

In the project the hydraulic cylinder movement characteristic may induce is three kind of types. The chart 1.2 is three kind of types hydraulic cylinders v —t chart, the first kind of like chart 1.2 center solid lines show, the hydraulic cylinder starts to make the uniform accelerated motion, then uniform motion,

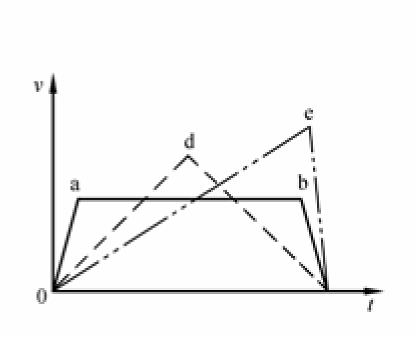

Chart 1.2 speeds circulation chart

Finally uniform retarded motion to end point; The second kind, the hydraulic cylinder preceding partly makes the uniform accelerated motion in the overall travelling schedule, in another one partly makes the uniform retarded motion, also the acceleration value is equal; The third kind, the hydraulic cylinder one most above makes the uniform accelerated motion in the overall travelling schedule by a smaller acceleration, then uniform decelerates to the travelling schedule end point. V—t chart three velocity curve, not only clearly has indicated three kind of types hydraulic cylinders movement rule, also indirectly has indicated three kind of operating modes dynamic performance.

1.2 mechanical analyses

1.2.1 hydraulic cylinders loads and duty cycle chart

1.2.1.1 hydraulic cylinders load strength computations

When the operating mechanism makes the straight reciprocating motion, the hydraulic cylinder must overcome the load is composed by six parts

$$F = F_c + F_f + F_i + F_g + F_m + F_b \tag{1.1}$$

In the formula: F_c In order to resistance to cutting; F_f In order to friction drag; F_i For inertia resistance; F_g For gravity; F_m In order to seal the resistance; F_b In order to drain the oil the resistance.

1.2.1.2 hydraulic cylinders cycle of motion various stages overall load strength

The hydraulic cylinder cycle of motion various stages overall load strength computation, generally includes the start acceleration, quickly enters, the labor enters, quickly draws back, decelerates applies the brake and so on several stages, each stage overall load strength has the difference.

(1) starts the acceleration period: By now the hydraulic cylinder or the piston were in from static enough to starts and accelerates to the certain speed, its overall load strength including guide rail friction force, packing assembly friction force (according to cylinder mechanical efficiency ηm=0.9 computation), gravity and so on item, namely:

$$F = F_f + F_i + F_g + F_m + F_b \tag{1.2}$$

(2) fast stage:

$$F = F_f + F_g + F_m + F_b \tag{1.3}$$

(3) the labor enters the stage:

$$F = F_c + F_f + F_g + F_m + F_b \tag{1.4}$$

(4) decelerates:

$$F = F_f + F_i + F_g + F_m + F_b \tag{1.5}$$

To the simple hydraulic system, the above computation process may simplify. For example uses the single proportioning pump to supply the oil, only must calculate the labor to enter the stage the overall load strength, if the simple system uses the limiting pressure type variable displacement pump or a pair of association pumps for the oil, then only must calculate the fast stage and the labor enters the stage the overall load strength.

1.2.2 oil motors load

When the operating mechanism makes the rotary motion, the oil motor must overcome the outside load is:

$$M = M_e + M_f + M_i \tag{1.6}$$

1.2.2.1 operating duties moment of force Me. The operating duty moment of force is possibly a definite value, also possibly as necessary changes, should carry on the concrete analysis according to the machine working condition.

1.2.2.2 friction moments. In order to revolve the part journal place friction moment, its formula is:

$$M_f = GFR(N \cdot M) \tag{1.7}$$

In the formula: G is revolves the part weight (N); F is the rubbing factor, when the start for the factor, after the start for moves the rubbing factor; R is the journal radius (m).

1.2.2.3 moment of inertiaM_i. The moment of inertia which in order to revolve the part acceleration or decelerates when produces, its formula is:

$$M_i = J\varepsilon\Delta t(N \cdot M) \tag{1.8}$$

In the formula: ε Is the angle acceleration (r/s^2); Δt is the acceleration or decelerates the time (s); J is revolves the part rotation inertia

$(Kg \cdot m^2)$, $J = 1GD^2/4G$

In the formula: GD^2 In order to rotate the part the flywheel effect ($N \cdot M^2$).

Each kind may look up <Machine design Handbook>

According to the type (1.6), separately figures out the oil motor in a operating cycle various stages load size, then may draw up the oil motor the duty cycle chart

2 determinations hydraulic system main parameter

2.1 hydraulic cylinders design calculations

2.1.1 initially decides the hydraulic cylinder working pressure

In the hydraulic cylinder working pressure main basis cycle of motion various stages biggest overall load strength determined, in addition below, but also needs to consider the factor:

2.1.1.1 each kind of equipment different characteristic and use situation.

2.1.1.2 considerations economies and the weight factor, the pressure elects lowly, then part size big, the weight is heavy; The pressure chooses high somewhat, then part size small, the weight is light, but to the part manufacture precision, the sealing property requests high.

Therefore, the hydraulic cylinder working pressure choice has two ways: One, elects according to the mechanical type; Two, according to cuts the load to elect.

If the table 2.1, the table 2.2 shows.

The table 2.1 presses the load to choose the execution file the working pressure

Load /N	＜5000	500 ～ 10000	10000 ～ 20000	20000 ～ 30000	30000 ～ 50000	＞50000
Working pressure /MPa	≤0.8 ～ 1	1.5 ～ 2	2.5 ～ 3	3 ～ 4	4 ～ 5	＞5

The table 2.2 presses the mechanical type to choose the execution file the working pressure

Mechanical type	Engine bed				Farm machinery	Project machinery
	Grinder	Aggregate machine-tool	Dragon Gate digs the bed	Broaching machine		
Working pressure /MPa	a≤2	3 ≀ 5	≤8	8 ≀ 10	10 ≀ 16	20 ≀ 32

2.2 oil motors design calculation

2.2.1 computations oil motor displacement

Under oil motor displacement according to the type decided that,

$$V_m = 6.28T / \Delta P_m \, \eta_{\min} \, (m^3/r) \tag{2.1}$$

In the formula: T is the oil motor load moment of force (N•m); ΔP_m For oil motor import and export pressure difference (n/m^3);is the oil motor mechanical efficiency, the common gear and the plunger motor takes 0.9 ~ 0.95, the leaf blade motor takes 0.8 ~ 0.9.

2.2.2 computations oil motor needs the current capacity oil motor the maximum current capacity

$$q_{\max} = V_m n_{\max} \, (m^3/s) \tag{2.2}$$

In the formula: V_m is the oil motor displacement (m^3/r); $n_{\max}$ is the oil motor highest rotational speed (r/s).

3 hydraulic pressure parts choice

3.1 hydraulic pumps determinations with need the power the computation

3.1.1.1 determines the hydraulic pump the biggest working pressure. The hydraulic pressure pumping station must the working pressure determination, mainly acts according to the hydraulic cylinder in the operating cycle various stages to have most tremendous pressure p1, in addition the oil pump loses Sigma Delta p the oil mouth to the cylinder place always pressure**ΣΔp**, namely

$$P_B = P_1 + \sum \Delta P \tag{3.1}$$

$\sum \Delta P$ loses, the pipeline including the oil after the flow valve and other parts local pressures along the regulation loss and so on, before system pipeline design, may act according to the similar system experience to estimate, common pipeline simple throttle valve velocity modulation system ΣΔp is (2 ~ 5) ×105Pa, with the velocity modulation valve and pipeline complex system $\sum \Delta P$ is (5 ~ 15) ×105Pa, $\sum \Delta P$ also may only consider flows after various control valves pressure loss, but ignores the circuitry along the regulation loss, various valves rated pressure loses may searches from the hydraulic pressure part handbook or the product sample, Also may refer to the table 1.3 selections

The table 3.1 is commonly used, the low pressure each kind of valve pressure loses (**Δp_n**)

Valve	**Δp_n** (×105Pa)	Valve	**Δp_n** (×105Pa)	Valve	**Δp_n** (×105Pa)	Valve	**Δp_n** (×105Pa)
Cone-way valve	0.3 ~ 0.5	Cone -way valve	3 ~ 8	Cone -way valve	1.5 ~ 2	Cone -way valve	1.5 ~ 2
Cross valve	1.5 ~ 3	Cross valve	2 ~ 3	Cross valve	1.5 ~ 3	Cross valve	3 ~ 5

3.1.2 determines the hydraulic pump current capacityqB

Pumps the current capacity q_B basis functional element operating cycle must the maximum current capacity q_{max} and the system divulges the determination

3.1.2.1 At the same time when more than hydraulic cylinders movement, the hydraulic pump current capacity must be bigger than the maximum current capacity which at the same time the movement several hydraulic cylinders (or motor) needs, and should consider the system divulging wears the volumetric efficiency drop after the hydraulic pump, namely

$$q_B = K(\sum q)_{\max} (m^3/s) \tag{3.2}$$

In the formula: K is the system leakage coefficient, generally takes 1.1 ~ 1.3, the great current capacity takes the small value, the small current capacity takes the great value $(\sum q)_{max}$; For at the same time movement hydraulic cylinder (or motor) is biggest (m^3/s).

3.1.2.2 chooses the hydraulic pump the specification

Table 3.2 hydraulic pumps overall effectiveness indices

Hydraulic pump type	Gear pump	The screw rod pumps	Vane pump	Ram pump
Overall effectiveness index	0.6 ～ 0.7	0.65 ～ 0.80	0.60 ～ 0.75	0.80 ～ 0.85

Rotational speed and pumps which according to the above power, may select the standard electric motor from the product sample, again carries on, causes when the electric motor sends out the maximum work rate, in permission scope.

3.2 valves class parts choice

3.2.1 choices bases

The choice basis is: Rated pressure, maximum current capacity, movement way, installment fixed way, pressure loss value, operating performance parameter and working life and so on.

3.2.2 selector valves class parts should pay attention question

3.2.2.1 should select the standard stereotypia product as far as possible, only if does not have already time only then independently designs special-purp

3.2.2.2 valves class parts specification main basis class after this valve fat liquor most tremendous pressure and maximum current capacity selection. When chooses the overflow valve, should according to the hydraulic pump maximum current capacity selection; When chooses the throttle valve and the velocity modulation valve, should consider its minimum stable current capacity satisfies the machine low-speed performance the request

3.3 accumulators choices

3.3.1 accumulators use in to supplement when the hydraulic pump supplies the oil insufficiency, its dischargeable capacity is

$$V = \sum A_i L_i K - q_B t (m^3) \tag{3.3}$$

In the formula: A is the hydraulic cylinder active surface (m^2); L is the hydraulic cylinder travelling schedule (m); K is the hydraulic cylinder loss coefficient, when the estimate may take K = 1.2; Supplies the oil current capacity for the hydraulic pump (m^3/s); T is the operating time (s).

3.3.2 accumulators make the emergency energy, its dischargeable capacity is:

$$V = \sum A_i L_i - q_B t (m^3) \tag{3.4}$$

When the accumulator uses in absorbs the pulsation to relax the hydraulic pressure impact, should take it as in the system a link if to be connected partially together synthesizes considers its dischargeable capaci According to the dischargeable capacity which extracts and considered other requests, then chooses the accumulator the form

3.4 pipelines choices

3.4.1 drill tubings types choice

In the hydraulic system uses the drill tubing divides the hard tube and the hose, the choice drill tubing should have enough passes flows the section and the bearing pressure ability, simultaneously, should reduce the pipeline as far as possible, avoids the extreme turn and the section sudden change.

3.4.1.1 steel pipes: Center the high tension system selects the seamless steel pipe, the low pressure system selects the welded steel pipe, the steel pipe price lowly, performance good, the use is widespread

3.4.1.2 copper pipes: The copper tube working pressure below 6.5 ~ 10MPa, the instable tune, is advantageous for the assembly; Yellow copper pipe withstanding pressure higher, reaches 25MPa, was inferior to the copper tube is easy to be curving. Copper pipe price high, earthquake resistance ability weak, is easy to cause the fat liquor oxidation, should as far as possible little use, only uses in the hydraulic unit to match meets not the convenient spot.

3.4.2 drill tubings sizes determination

3.4.2.1 drill tubings inside diameters d presses down the type computation

$$d=\sqrt{\frac{4q}{\pi v}} = 1.13\times10^{3}\sqrt{\frac{q}{v}} \qquad (3.5)$$

In the formula: Q is passes the drill tubing the maximum current capacity (m^3/s); V speed of flow which permits for the pipeline in (m/s). The common oil suction pipe takes 0.5 ~ 5 (m/s); The pressure oil pipe takes 2.5 ~ 5 (m/s); The oil return pipe takes 1.5 ~ 2 (m/s).

3.4.2 drill tubings δ sizes determination

$$\delta \geq p \bullet \frac{d}{2}(\sigma) \qquad (3.6)$$

In the formula: P is in the tube the biggest working pressure; When n is the safety coefficient, steel pipe p < 7MPa, takes n=8; When p < 17.5MPa, takes n=6; When p > 17.5MPa, takes n=4.

According to drill tubing inside diameter and wall thickness which calculates, looks up the handbook selection standard specification drill tubing

3.5 fuel tank design

The fuel tank function is the oil storage, disperses the oil discharge the quantity of heat, in the precipitation oil the impurity, is leisurely in the oil the gas

3.5.1 fuel tanks designs main point

3.5.1.1 fuel tanks should have the enough volume to satisfy the radiation, simultaneously its volume should guarantee in the system the fat liquor completely flows when the fuel tank does not seep out, the fat liquor liquid level should not surpass the fuel tank highly 80%.

3.5.1.2 suction boxes tubes and the oil return pipe spacing should be as far as possible big

3.5.1.3 fuel tanks bases should have the suitable ascent, releases the oil mouth to set to the most low spot, in order to drains the oil

3.6 oil filters choices

Chooses the oil filter the basis to have following several

3.6.1 bearing capacities

According to system pipeline working pressure determination.

3.6.2 filters the precision:

According to is protected the part the precision request determination

3.6.3 flow the ability:

According to through maximum current capacity determination.

3.6.4 resistance pressure drops:

Should the satisfied filter material intensity and the coefficient request.

4 hydraulic systems performance

In order to judge the hydraulic system the design quality, needs to lose to the system pressure, to give off heat , the efficiency and system dynamic characteristic and so on

4.1 circuitries pressure loses

After hydraulic pressure part specification model and pipeline size determination, may the more accurate computing system pressure loss, the pressure loss include: The oil loses , ΔP_L the local pressure after the pipeline ΔP_C along the regulation pressure damages flows after the valve class part pressure loss ΔP_V , namely:

$$\Delta P = \Delta P_L + \Delta P_C + \Delta P_V \tag{4.1}$$

System adjustment pressure:

$$P_0 \geq P_1 + \Delta P \tag{4.2}$$

In the formula: P_0For hydraulic pump working pressure or leg adjustment pressure; P_1In order to execution working pressure.

If calculates ΔP in the primary election system working pressure time the is sketchier than designation pressure to lose is much bigger than, should remove entire related part, auxiliary specification, again definite pipeline size.

4.2 systems give off heat

The system gives off heat originates from the system interior energy loss, like the hydraulic pump and the functional element power loss, the overflow valve overflow loses, the hydraulic valve and the pipeline pressure loss and so on.

The system gives off heat the power P computation

$$P = P_B (1 - \eta)(W) \tag{4.3}$$

In the formula: PB is the hydraulic pump power input (W); η Is the hydraulic pump overall effectiveness index

If in a operating cycle has several working procedures, then may act according to each working procedure the calorific capacity, extracts the system unit time the average calorific capacity:

$$P = \frac{1}{T}\sum_{i=1}^{n} P_{b_i}(1-\eta)t_i(W) \qquad (4.4)$$

In the formula: T is the operating cycle cycle (s); t_i For i working procedure operating time (s); p_i is in the circulation the i working procedure power input (W).

4.3 systems efficiency

The hydraulic system efficiency is by the hydraulic pump, the functional element and the hydraulic pressure return route efficiency determined

The hydraulic pressure η_c return route efficiency generally may use the type to calculate:

$$\eta_c = \frac{p_1 q_1 + p_2 q_2 + \dots\dots\dots}{p_{b_1} q_{b_2} + p_{b_2} q_{b_2}} \qquad (4.5)$$

In the formula: p_1, q_1; p_2, q_2 …… For each functional element working pressure and current capacity; p_{B1}, q_{B1}; p_{B2}, q_{B2} is each hydraulic pump supplies the oil pressure and the current capacity.

Hydraulic system overall effectiveness index:

$$\eta = \eta_B + \eta_m + \eta_c \qquad (4.6)$$

In the formula: η_B For hydraulic pump overall effectiveness index; η_m In order to functional element overall effectiveness index; η_c For return route efficiency

5 draws up the regular worker mapping and the compilation technology document

Passes through after the hydraulic system performance and the essential revision, then may draw up the regular worker mapping, it including plan hydraulic system schematic diagram, system pipeline assembly drawing and each kind of non- standard part design drawing.

In the official hydraulic system schematic diagram must mark various hydraulic pressure part the model specification. Regarding automaticity higher engine bed, but also should include the movement part the cycle of motion chart and the electro-magnet, the pressure switch active status.

5.1 determinations hydraulic system parameter

May know by the operating mode analysis in, the labor enters the stage the load strength to be biggest, therefore, the hydraulic cylinder working pressure according to this load strength computation, according to the hydraulic cylinder and the load relations, p_1=40×105Pa. This engine bed for the drill hole aggregate machine-tool, for prevented drills through before when occurs flushes the phenomenon, the hydraulic cylinder oil discharge cavity should have the back pressure, `p_2 =6×105Pa, for causes quickly to enter quickly draws back the speed to be equal, selects $A_1 = 2A_2$ the differential motion cylinder, the hypothesis quickly enters the oil discharge pressure which, quickly draws back to lose for △ p=7×105Pa.

5.2 choices hydraulic pressure part

5.2.1 chooses the hydraulic pump and the electric motor

5.2.1.1 determines the hydraulic pump the working pressure.

Front had determined the hydraulic cylinder the biggest working pressure for 40×105Pa, selects the intake pipe road pressure to lose △ p=8×105Pa, its adjustment pressure is generally bigger than the system biggest working pressure 5×105Pa, therefore pumps working pressure P_B = (40 + 8 + 5) ×105 = 53×105Pa

This is the working pressure which the high-pressured small current capacity pumps.

The hydraulic cylinder quickly draws back when the working pressure quickly enters when is bigger than, takes its pressure to lose Delta p ’= 4×105Pa, then quickly draws back time pumps the working pressure is:

P_B =(16.4＋4)×105 || 20.4×105Pa

This is the working pressure which the low pressure great current capacity pumps.

5.2.1.2 hydraulic pumps current capacities. Quickly enters when the current capacity is biggest, its value is 30L/min, the quantity enters when the labor, its value is 0.51L/min, takes K = 1.2, Then: q_B ‖ 1.2×0.5×10-3=36L/min

Because time the overflow valve steady work most is small is 3L/min, therefore slightly pumps the current capacity to take 3.6L/min

Calculates according to above, selects the YYB-AA36/6B double joint vane pump

5.2.1.3 definite pipelines sizes: According to the working pressure and the current capacity, according to the type (3.5), the type (3.6) determine the pipeline inside diameter and wall thickness. (Omits)

5.2.1.4 determinations fuel-tank capacity fuel-tank capacity may according to the empirical formula estimate, take V = (5 ~ 7) q. In this example: V = 6q = 6 (6 + 36) = 252L related system performance omits.

液压传动系统设计与计算

1 明确设计要求进行工况分析

在设计液压系统时，首先应明确以下问题，并将其作为设计依据。

主机的用途 工艺过程 总体布局以及对液压传动装置的位置和空间尺寸的要求；主机对液压系统的性能要求，如自动化程度、调速范围、运动平稳性、换向定位精度以及对系统的效率、温升等的要求；液压系统的工作环境，如温度、湿度、振动冲击以及是否有腐蚀性和易燃物质存在等情况。

在上述工作的基础上，应对主机进行工况分析，工况分析包括运动分析和动力分析，对复杂的系统还需编制负载和动作循环图，由此了解液压缸或液压马达的负载和速度随时间变化的规律，以下对工况分析的内容作具体介绍。

1.1 运动分析

主机的执行元件按工艺要求的运动情况，可以用位移循环图(L-t)，速度循环图(v-t)，或速度与位移循环图表示，由此对运动规律进行分析。

1.1.1 位移循环图 L-t

图 1.1 为液压机的液压缸位移循环图，纵坐标 L 表示活塞位移，横坐标 t 表示从活塞启动到返回原位的时间，曲线斜率表示活塞移动速度。

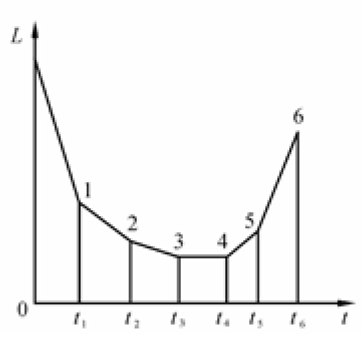

图 1.1 位移循环图

1.1.2 速度循环图 v-t(或 v-L)

工程中液压缸的运动特点可归纳为三种类型。图 1.2 为三种类型液压缸的 v-t 图，第一种如图 1.2 中实线所示，液压缸开始作匀加速运动，然后匀速运动，

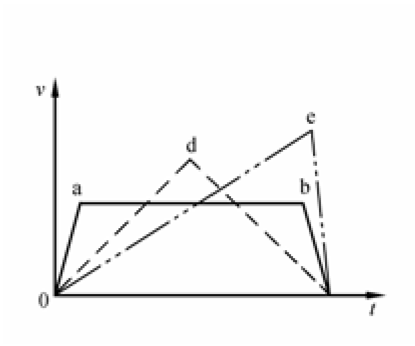

图 1.2 速度循环图

最后匀减速运动到终点；第二种，液压缸在总行程的前一半作匀加速运动，在另一半作匀减速运动，且加速度的数值相等；第三种，液压缸在总行程的一大半以上以较小的加速度作匀加速运动，然后匀减速至行程终点。v—t 图的三条速度曲线，不仅清楚地表明了三种类型液压缸的运动规律，也间接地表明了三种工况的动力特性。

1.2 动力分析

动力分析，是研究机器在工作过程中，其执行机构的受力情况，对液压系统而言，就是研究液压缸或液压马达的负载情况。

1.2.1 液压缸的负载及负载循环图

1.2.1.1 液压缸的负载力计算

工作机构作直线往复运动时，液压缸必须克服的负载由六部分组成：

$$F = F_c + F_f + F_i + F_g + F_m + F_b \tag{1.1}$$

式中：F_c 为切削阻力；F_f 为摩擦阻力；F_i 为惯性阻力；F_g 为重力；F_m 为密封阻力；F_b 为排油阻力。

1.2.1.2 液压缸运动循环各阶段的总负载力

液压缸运动循环各阶段的总负载力计算，一般包括启动加速、快进、工进 快退、减速制动等几个阶段，每个阶段的总负载力是有区别的。

(1)启动加速阶段：这时液压缸或活塞处于由静止到启动并加速到一定速度，其总负载力包括导轨的摩擦力、密封装置的摩擦力(按缸的机械效率 η_m=0.9 计算）、重力和惯性力等项，即：

$$F = F_f + F_i + F_g + F_m + F_b \tag{1.2}$$

(2)快速阶段：

$$F = F_f + F_g + F_m + F_b \tag{1.3}$$

(3)工进阶段：

$$F = F_c + F_f + F_g + F_m + F_b \tag{1.4}$$

(4)减速：

$$F = F_f + F_i + F_g + F_m + F_b \tag{1.5}$$

对简单液压系统，上述计算过程可简化。例如采用单定量泵供油，只需计算工进阶段的总负载力，若简单系统采用限压式变量泵或双联泵供油，则只需计算快速阶段和工进阶段的总负载力。

1.2.2 液压马达的负载

工作机构作旋转运动时，液压马达必须克服的外负载为：

$$M = M_e + M_f + M_i \tag{1.6}$$

1.2.2.1 工作负载力矩 M_e。工作负载力矩可能是定值，也可能随时间变化，应根据机器工作条件进行具体分析。

1.2.2.2 摩擦力矩 M_f。为旋转部件轴颈处的摩擦力矩，其计算公式为：

$$M_f = GFR(N \cdot M) \tag{1.7}$$

式中：G 为旋转部件的重量(N)；f 为摩擦因数，启动时为静摩擦因数，启动后为动摩擦因数；R 为轴颈半径(m)。

1.2.2.3 惯性力矩 M_i。为旋转部件加速或减速时产生的惯性力矩，其计算公式为：

$$M_i = J\varepsilon\Delta t(N \cdot M) \tag{1.8}$$

式中：ε为角加速度(r/s²)；Δω为角速度的变化(r/s)；Δt 为加速或减速时间(s)；J 为旋转部件的转动惯量($Kg \cdot m^2$)，。$J = 1GD^2/4G$

式中：GD^2 为回转部件的飞轮效应($N \cdot M^2$)。

各种回转体的 GD^2 可查《机械设计手册》。

根据式(1.6)，分别算出液压马达在一个工作循环内各阶段的负载大小，便可绘制液压马达的负载循环图

2　确定液压系统主要参数

2.1 液压缸的设计计算

2.1.1 初定液压缸工作压力

液压缸工作压力主要根据运动循环各阶段中的最大总负载力来确定，此外，还需要考虑以下因素：

2.1.1.1 各类设备的不同特点和使用场合。

2.1.1.2 考虑经济和重量因素，压力选得低，则元件尺寸大，重量重；压力选得高一些，则元件尺寸小，重量轻，但对元件的制造精度，密封性能要求高。

所以，液压缸的工作压力的选择有两种方式：一是根据机械类型选；二是根据切削负载选。

如表 2.1、表 2.2 所示。

表 **2.1** 按负载选执行文件的工作压力

负载/N	＜5000	500～10000	10000～20000	20000～30000	30000～50000	＞50000
工作压力/MPa	≤0.8～1	1.5～2	2.5～3	3～4	4～5	＞5

表 **2.2** 按机械类型选执行文件的工作压力

机械类型	机　　床				农业机械	工程机械
	磨床	组合机床	龙门刨床	拉床		
工作压力/MPa	a≤2	3～5	≤8	8～10	10～16	20～32

2.2 液压马达的设计计算

2.2.1 计算液压马达排量

液压马达排量根据下式决定：

$$V_m = 6.28T/\Delta P_m \eta_{\min} (m^3/r) \tag{2.1}$$

式中：T 为液压马达的负载力矩(N?m)；ΔP_m 为液压马达进出口压力差(N/m^3)；$\eta_{\min}$ 为液压马达的机械效率，一般齿轮和柱塞马达取 0.9～0.95，叶片马达取 0.8～0.9。

2.2.2 计算液压马达所需流量液压马达的最大流量

$$q_{\max} = V_m n_{\max} (m^3/s) \tag{2.2}$$

式中：V_m 为液压马达排量（m^3/r）；n_{max} 为液压马达的最高转速（r/s）。

3 液压元件的选择

3.1 液压泵的确定与所需功率的计算

3.1.1 液压泵的确定

3.1.1.1 确定液压泵的最大工作压力。液压泵所需工作压力的确定，主要根据液压缸在工作循环各阶段所需最大压力 p_1，再加上油泵的出油口到缸进油口处总的压力损失 $\Sigma\Delta p$，即

$$P_B = P_1 + \sum \Delta P \qquad (3.1)$$

$\sum \Delta P$ 包括油液流经流量阀和其他元件的局部压力损失、管路沿程损失等，在系统管路未设计之前，可根据同类系统经验估计，一般管路简单的节流阀调速系统 $\sum \Delta P$ 为(2～5)×105Pa，用调速阀及管路复杂的系统 $\sum \Delta P$ 为(5～15)×105Pa，$\sum \Delta P$ 也可只考虑流经各控制阀的压力损失，而将管路系统的沿程损失忽略不计，各阀的额定压力损失可从液压元件手册或产品样本中查找，也可参照表 1.3 选取。

表 3.1 常用中、低压各类阀的压力损失(Δp_n)

阀名	Δp_n(×10^5Pa)	阀名	Δp_n (×105Pa)	阀名	Δp_n (×105Pa)	阀名	Δp_n (×105Pa)
单向阀	0.3～0.5	背压阀	3～8	行程阀	1.5～2	转阀	1.5～2
换向阀	1.5～3	节流阀	2～3	顺序阀	1.5～3	调速阀	3～5

3.1.2 确定液压泵的流量 q_B

泵的流量 q_B 根据执行元件动作循环所需最大流量 q_{max} 和系统的泄漏确定。

3.1.2.1 多液压缸同时动作时，液压泵的流量要大于同时动作的几个液压缸(或马达)所需的最大流量，并应考虑系统的泄漏和液压泵磨损后容积效率的下降，即

$$q_B = K(\sum q)_{\max}(m^3/s) \qquad (3.2)$$

式中：K 为系统泄漏系数，一般取 1.1～1.3，大流量取小值，小流量取大值；$(\sum q)_{\max}$ 为同时动作的液压缸(或马达)的最大总流量(m^3/s)。

3.1.2.2 选择液压泵的规格：根据上面所计算的最大压力 p_B 和流量 q_B，查液压元件产品样本，选择与 p_B 和 q_B 相当的液压泵的规格型号。

表 3.2 液压泵的总效率

液压泵类型	齿轮泵	螺杆泵	叶片泵	柱塞泵
总效率	0.6～0.7	0.65～0.80	0.60～0.75	0.80～0.85

按上述功率和泵的转速’可以从产品样本中选取标准电动机’再进行验算’使电动机发出最大功率时’其超载量在允许范围内。

3.2 阀类元件的选择

3.2.1 选择依据

选择依据为∶额定压力’最大流量’动作方式’安装固定方式’压力损失数值’工作性能参数和工作寿命等。

3.2.2 选择阀类元件应注意的问题

3.2.2.1 应尽量选用标准定型产品’除非不得已时才自行设计专用件。

3.2.2.2 阀类元件的规格主要根据流经该阀油液的最大压力和最大流量选取。选择溢流阀时’应按液压泵的最大流量选取∶选择节流阀和调速阀时’应考虑其最小稳定流量满足机器低速性能的要求。

3.3 蓄能器的选择

3.3.1 蓄能器用于补充液压泵供油不足时’其有效容积为∶

$$V = \sum A_i L_i K - q_B t(m^3) \tag{3.3}$$

式中∶A 为液压缸有效面积(m^2)∶L 为液压缸行程(m)∶K 为液压缸损失系数’估算时可取 K ‖ 1.2∶q_B 为液压泵供油流量(m^3/s)∶t 为动作时间(s)。

3.3.2 蓄能器作应急能源时’其有效容积为∶

$$V = \sum A_i L_i - q_B t(m^3) \tag{3.4}$$

当蓄能器用于吸收脉动缓和液压冲击时 应将其作为系统中的一个环节与其关联部分一起综合考虑其有效容积。

根据求出的有效容积并考虑其他要求’即可选择蓄能器的形式。

3.4 管道的选择

3.4.1 油管类型的选择

液压系统中使用的油管分硬管和软管’选择的油管应有足够的通流截面和承压能力’同时’应尽量缩短管路’避免急转弯和截面突变。

3.4.1.1 钢管∶中高压系统选用无缝钢管’低压系统选用焊接钢管’钢管价格低’性能好’使用广泛。

3.4.1.2 铜管；紫铜管工作压力在 6.5～10MPa 以下，易变曲，便于装配；黄铜管承受压力较高，达 25MPa，不如紫铜管易弯曲。铜管价格高，抗震能力弱，易使油液氧化，应尽量少用，只用于液压装置配接不方便的部位。

3.4.2 油管尺寸的确定

3.4.2.1 油管内径 d 按下式计算：

$$d=\sqrt{\frac{4q}{\pi v}}=1.13\times10^{3}\sqrt{\frac{q}{v}} \qquad (3.5)$$

式中：q 为通过油管的最大流量(m^3/s)；v 为管道内允许的流速(m/s)。一般吸油管取 0.5～5(m/s)；压力油管取 2.5～5(m/s)；回油管取 1.5～2(m/s)。

3.4.2.2 油管壁厚δ按下式计算：

$$\delta \ge p \bullet \frac{d}{2}(\sigma) \qquad (3.6)$$

式中：p 为管内最大工作压力；n 为安全系数，钢管 p＜7MPa 时，取 n=8；p＜17.5MPa 时，取 n=6；p＞17.5MPa 时，取 n=4。

根据计算出的油管内径和壁厚，查手册选取标准规格油管。

3.5 油箱的设计

油箱的作用是储油，散发油的热量，沉淀油中杂质，逸出油中的气体。

3.5.1 油箱设计要点

3.5.1.1 油箱应有足够的容积以满足散热，同时其容积应保证系统中油液全部流回油箱时不渗出，油液液面不应超过油箱高度的 80%。

3.5.1.2 吸箱管和回油管的间距应尽量大。

3.5.1.3 油箱底部应有适当斜度，泄油口置于最低处，以便排油。

3.6 滤油器的选择

选择滤油器的依据有以下几点：

3.6.1 承载能力：

按系统管路工作压力确定。

3.6.2 过滤精度：

按被保护元件的精度要求确定。

3.6.3 通流能力 ：

按通过最大流量确定 。

3.6.4 阻力压降 ：

应满足过滤材料强度与系数要求 。

4 液压系统性能的验算

为了判断液压系统的设计质量，需要对系统的压力损失、发热温升、效率和系统的动态特性等进行验算。

4.1 管路系统压力损失的验算

当液压元件规格型号和管道尺寸确定之后，就可以较准确的计算系统的压力损失，压力损失包括：油液流经管道的沿程压力损失 ΔP_L、局部压力损失 ΔP_C 和流经阀类元件的压力损失 ΔP_V，即：

$$\Delta P = \Delta P_L + \Delta P_C + \Delta P_V \tag{4.1}$$

系统的调整压力：

$$P_0 \geq P_1 + \Delta P \tag{4.2}$$

式中：P_0 为液压泵的工作压力或支路的调整压力；P_1 为执行件的工作压力。

如果计算出来的 ΔP 比在初选系统工作压力时粗略选定的压力损失大得多，应该重新调

整有关元件、辅件的规格，重新确定管道尺寸。

4.2 系统发热温升的验算

系统发热来源于系统内部的能量损失，如液压泵和执行元件的功率损失、溢流阀的溢流损失、液压阀及管道的压力损失等。

系统发热功率 P 的计算:

$$P = P_B(1-\eta)(W) \tag{4.3}$$

式中：P_B 为液压泵的输入功率(W)；η为液压泵的总效率。

若一个工作循环中有几个工序，则可根据各个工序的发热量，求出系统单位时间的平均发热量：

$$P = \frac{1}{T}\sum_{i=1}^{n} P_{b_i}(1-\eta)t_i(W) \tag{4.4}$$

式中：T 为工作循环周期(s)；t_i 为第 i 个工序的工作时间(s)；p_i 为循环中第 i 个工序的输入功率(W)。

4.3 系统效率验算

液压系统的效率是由液压泵、执行元件和液压回路效率来确定的。

液压回路效率 n_c 一般可用下式计算：

$$\eta_c = \frac{p_1 q_1 + p_2 q_2 + \dots\dots\dots}{p_{b_1} q_{b_2} + p_{b_2} q_{b_2}} \tag{4.5}$$

式中：p_1，q_1；p_2，q_2；……：为每个执行元件的工作压力和流量；p_{B1}，q_{B1}；p_{B2}，q_{B2} 为每个液压泵的供油压力和流量。

液压系统总效率：

$$\eta = \eta_B + \eta_m + \eta_c \tag{4.6}$$

式中：η_B 为液压泵总效率；η_m 为执行元件总效率；η_c 为回路效率。

5　绘制正式工作图和编写技术文件

经过对液压系统性能的验算和必要的修改之后’便可绘制正式工作图”它包括绘制液压系统原理图`系统管路装配图和各种非标准元件设计图°

正式液压系统原理图上要标明各液压元件的型号规格 对于自动化程度较高的机床’还应包括运动部件的运动循环图和电磁铁`压力继电器的工作状态°

5.1 确定液压系统参数

由工况分析中可知’工进阶段的负载力最大’所以’液压缸的工作压力按此负载力计算’根据液压缸与负载的关系’选 p_1=40×105Pa°本机床为钻孔组合机床’为防止钻通时发生前冲现象’液压缸回油腔应有背压’设背压 p_2=6×105Pa’为使快进快退速度相等 选用 $A_1 = 2A_2$ 差动油缸 假定快进 快退的回油压力损失为Δ**p**=7×105Pa°

5.2 选择液压元件

5.2.1 选择液压泵和电动机

5.2.1.1 确定液压泵的工作压力°

前面已确定液压缸的最大工作压力为 40×105Pa 选取进油管路压力损失Δp=8×105Pa’其调整压力一般比系统最大工作压力大 5×105Pa’所以泵的工作压力

P_B▌(40＋8＋5)×105‖53×105Pa

这是高压小流量泵的工作压力°

液压缸快退时的工作压力比快进时大’取其压力损失Δp′ ‖4×105Pa’则快退时泵的工作压力为：

P_B=(16.4＋4)×105‖20.4×105Pa

这是低压大流量泵的工作压力°

5.2.1.2 液压泵的流量°快进时的流量最大’其值为 30L/min’最小流量在工进时’其值为 0.51L/min’取 K‖1.2’

则：　　　　　　　q_B▌1.2×0.5×10-3=36L/min

由于溢流阀稳定工作时的最小溢流量为 3L/min’故小泵流量取 3.6L/min°

根据以上计算’选用 YYB-AA36/6B 型双联叶片泵°

5.2.1.3 确定管道尺寸:根据工作压力和流量’按式(3.5)`式(3.6)确定管道内径和

壁厚 。(从略)

5.2.1.4 确定油箱容量油箱容量可按经验公式估算 ’取 V‖(5≀7)q 。

本例中 ׃V▮6q▮6(6＋36)▮252L 有关系统的性能验算从略 。