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基于形封闭偏心凸轮与铰杆增力机构的手动压力机

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摘要　介绍了两种基于形封闭偏心凸轮与铰杆增力机构的手动压力机,分析了它们的工作原理,给出了相应的力学计算公式。这两种创新的手动压力机,均采用了形封闭偏心凸轮机构,不需要外加复位装置,具有结构简单,增力效果显著等优点。

关键词　偏心凸轮　形封闭机构　铰杆　增力机构　手动压力机

引言

目前常见的手动压力机,一般采用 3 种机构:1. 杠杠机构;2. 杠杆 —齿轮齿条机构;3. 力封闭偏心凸轮机构。相对来说,杠杆机构的增力系数取决于主被动臂的长度比,要获得较大的增力系数,必然使得杠杆的尺寸过长,导致结构不够紧凑;而采用力封闭偏心凸轮机构的手动压力机,则需要外加复位弹簧,导致在结构上显得赘余。

基于上述原因,我们选用形封闭偏心凸轮机构,与铰杆增力机构进行串联组合,设计了两种新型的手动压力机。

1　工作原理

1.1　采用过渡铰杆的手动压力机

压力机工作原理如图 1 所示,它主要由驱动杆、偏心凸轮、驱动滑块、过渡铰杆、双边铰杆增力机构和压头组成。为了解决装置自由度不足的问题,在偏心凸轮机构与双边铰杆增力机构之间,增加了一支过渡铰杆。

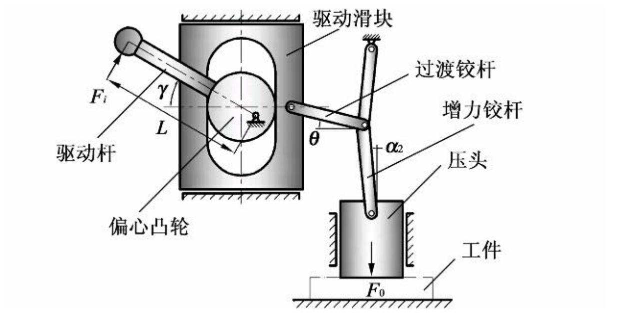

图 1

通过人手向驱动杆的球头部施加一个力,使驱动杆带动偏心凸轮顺时针转动,水平方向上便产生一个推动驱动滑块向右运动的力;该力通过铰接在驱动滑块上的过渡铰杆,传递到双边铰杆增力机构上,进而由该机构产生向下的推力,来推动压头向下运动对工件施加作用力,工作行程即告完成。工件加工完毕后,扳动驱动杆使偏心凸轮逆时针方向运动,使驱动滑块等向左运动,并最终带动压头向上运动脱离工件。

1.2　采用过渡滑块的手动压力机

该压力机的工作原理如图 2 所示,它主要由驱动杆、偏心凸轮、驱动滑块、过渡滑块、双边铰杆增力机构和压头组成。与第一种所不同的是,在驱动滑块的中间开有一个径向孔,孔中设置了一个过渡滑块,偏心凸轮机构与双边铰杆增力机构通过过渡滑块相连接,巧妙地解决了自由度不足的问题。

工作时,通过人手向驱动杆的球头部施加一个力,使驱动杆带动偏心凸轮顺时针转动,水平方向上便产生一个推动驱动滑块向右运动的力;该力传递到过渡滑块,由于双边铰杆增力机构的作用,使得驱动滑块径向孔中的过渡滑块向下运动,进而驱动压头作向下的滑移运动,从而作用于工件。工件加工完毕后,扳动驱动杆使偏心凸轮逆时针方向运动,使驱动滑块等向左运动,并最终带动压头向上运动脱离工件。

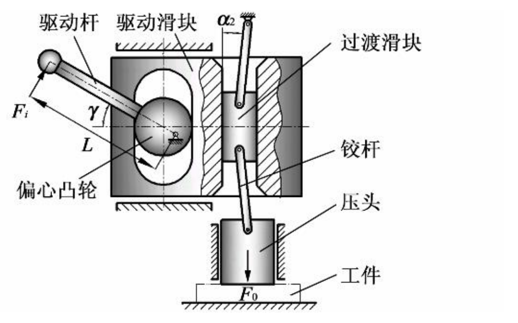

图 2

由图 1、图 2 不难看出,两种压力机都采用了形封闭偏心轮机构,在压头回程时可以方便地脱离工件,并且都采用了偏心凸轮机构和双边铰杆增力机构相组合的三级增力机构,增力效果显著。同样,它们的差异也显而易见,第一种采用了过渡铰杆的结构,尽管结构较为简单,但是刚性较差;第二种则采用了径向孔中放置

过渡滑块的方法,不仅灵活地解决了自由度不足的问题,还使得整个装置的刚性大大增强。

2 力学计算

以上压力机由两部分串联组合组成,一部分是偏心凸轮机构,另一部分是双边铰杆增力机构。其中,带驱动杆的偏心凸轮机构,实质上是一种串连式的两级增力机构,其中驱动杆的杠杆作用为第一级,偏心凸轮形成的斜楔作用为第二级。输入力经过该机构的二次放大后,又通过双边铰杆增力机构进行第三次力放大,所以此压力机为三级增力机构。

系统的增力系数是输出力 F_0 与输入力 F_i 的比值。不考虑摩擦损失的增力系数为理论增力系数,常用 i_t 表示;考虑摩擦损失后的增力系数为实际增力系数,常用 i_p 表示。

通过建立力学模型并分析计算可知,图 1 所示压力机的理论增力系数的计算公式为

$$i_{t1}=\frac{L\cos\gamma}{\rho\tan\alpha_1}\cdot\frac{1+\tan\alpha_2\tan\theta}{2\sin\alpha_2} \tag{1}$$

式中 L ——手柄端部到偏心凸轮偏心的距离

γ ——手柄相对于偏心凸轮中心的摆角

α_1 ——偏心轮力作用点处偏心凸轮的升角,

$\alpha_1=\dfrac{e\sin\gamma}{R-e\cos\gamma}$($e$ 为偏心距,R 为偏心凸轮的半径)

ρ ——偏心凸轮转动中心与其力作用点处的距离,$\rho=\sqrt{R^2+e^2-2Re\cos\gamma}$

θ ——过渡铰杆的理论压力角

α_2 ——增力链的理论压力角

其实际增力系数 i_{p1} 的计算公式为

$$i_{p1}=\frac{L\cos\gamma}{\rho[\tan(\alpha_1+\beta_1)+\tan\beta_3]}\times\{[1-\tan(\alpha_2+\beta_2)\tan\phi_2][1+\tan\theta\tan(\alpha_2+\beta_2)](1-\tan\beta_1\tan\phi_2)\}/[2\tan(\alpha_2+\beta_2)(1-\tan\theta\tan\phi_1)] \tag{2}$$

式中 β_1 ——偏心凸轮与驱动滑块作用处的摩擦角

β_2 ——铰链副的当量摩擦角,$\beta_2=\arcsin\dfrac{2r}{l}f$($r$ 为铰链轴半径,l 为铰杆上两铰链孔的中心距,f 为铰链副的摩擦因数)[3]

β_3 ——偏心凸轮转轴处的摩擦角

ϕ_1 ——驱动滑块与其导向槽之间的摩擦角,$\phi_1=\arctan\mu$(μ 为驱动滑块与导向槽之间的摩擦系数)

ϕ_2 ——压头与其导向槽间的当量摩擦角,其值由力输出件的受力及约束方式决定[4]。

在式(1)、式(2)中,前一部分为偏心凸轮机构的计算公式,后一部分为双边铰杆增力机构的计算公式。

图 2 所示压力机的理论增力系数 i_{t2} 和实际增力系数 i_{p2} 的计算公式分别为

$$i_{t2}=\frac{L\cos\gamma}{\rho\tan\alpha_1}\cdot\frac{1}{2\tan\alpha_2} \tag{3}$$

$$i_{p2}=\frac{L\cos\gamma}{\rho[\tan(\alpha_1+\beta_1)+\tan\beta_3]}\times\frac{[1-\tan(\alpha_2+\beta_2)\tan\phi_1][1-\tan(\alpha_2+\beta_2)\tan\phi_2]}{2\tan(\alpha_2+\beta_2)} \tag{4}$$

由此,两种压力机的实际输出力 F_{1p} 和 F_{2p} 的计算公式分别为

$$F_{p1}=\{L\cos\gamma[1-\tan(\alpha_2+\beta_2)\tan\phi_2][1+\tan\theta\tan(\alpha_2+\beta_2)](1-\tan\beta_1\tan\phi_1)\}F_i/[2\rho\tan\alpha_1\tan(\alpha_2+\beta_2)(1-\tan\theta\tan\phi_1)] \tag{5}$$

$$F_{p2}=\{L\cos\gamma[1-\tan(\alpha_2+\beta_2)\tan\phi_1][1-\tan(\alpha_2+\beta_2)\tan\phi_2]\}F_i/\{2\rho[\tan(\alpha_1+\beta_1)+\tan\beta_3]\tan(\alpha_2+\beta_2)\} \tag{6}$$

3 计算举例

对于铰杆机构,从理论上讲,其理论压力角 α_2 越小越好,但由于制造精度等原因,α_2 角的取值不可能很小,在工程实际中,一般取 $\alpha_{min}=3°\sim5°$。这里,我们取 $\alpha_2=4°$,$r=5$mm,$l=100$mm,$f=0.1$,计算可得 $\beta_2=0.57°$,同时取 $\mu=0.1$,$\phi_2=10°$,$\theta=6°$。对于偏心凸轮机构,取参数如下,$L=500$mm,$\gamma=30°$,$e=5$mm,$R=30$mm,$\beta_1=\beta_3=6°$。根据式(7)、式(8),代入计算得到两种手动压力机的实际增力系数分别为 $i_1=350.5$,$i_2=344.8$,二者相差甚微。

若取某手动压力机人手作用力为 500N,根据求得的增力系数 i_1,i_2,可计算得两种手动压力机的实际输出力分别为 $F_{1p}=175250$N,$F_{2p}=172400$N。而如采用单一的偏心凸轮机构,在其它条件一定的情况下,要取得如此大的输出力,经计算,驱动杆的力输入点到铰链轴的长度 L,需达 3m 以上,这在生产实践中可以说是没有可行性的。 (下转第 100 页)

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4 结论

本文提出的两种手动压力机具有以下特点:

(1) 采用了偏心凸轮和铰杆相组合的三级增力机构 ,由计算结果可知 ,输入力可放大 345 倍。因而在输入力一定的情况下 ,可以获得很大的输出力。

(2) 相对于力封闭机构而言 ,形封闭机构在压头回程时不需要复位弹簧 ,结构相对简单。

(3) 第一种压力机采用了过渡铰杆 ,压力机的刚性较差;第二种采用了过渡滑块 ,压力机的刚性大大加强;又因为两者增力系数相差不大 ,因而建议优先选用第二种压力机。

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nism synthesized have been obtained including mean point - to - point error , maximal point - to - point error , mean normal error and maximal normal error. The results provide reliable reference data for the designer. It is proved that the method is available by an example.

Key words Double straight - line guiding mechanism Path synthesis Error analysis Numerical atlas method

Abstract Based on the fundamental formulas , the diagrams are derivated and drawn reflecting the relationship between the structure and power flow of closed planetary gear train. They can be applied to analyze existed trains for the power flow direction. The process is not only clear and visual , but also convenient and fast. The design and power flow analysis are no longer difficult for closed planetary gear train.

Key words Closed planetary gear train Speed ratio Power flow Circulatory power flow

Abstract The virtual prototype of the flexible driver system is established by using ADAMS (Automatic dynamic analysis mechanical systems) , the effect of elastic coupling on natural vibration frequency of system is analyzed. The results also provide theoretical data to avoid system sympathetic vibration effectively.

Key words ADAMS Torsional vibration Natural frequency Flexible body

Abstract Some research is done on the intermittent - motion linkage with slight difference in length between the links. To analyze and evaluate the usability of this intermittent - motion linkage in practical engineering , some analyses of the special linkage , such as the law of motion , the mechanical advantage , the transmission and dynamic characteristic , are done. At the same time , the virtual prototype of this special mechanism is also taken into analysis on the Adams system by making use of its powerful functions on kinematics as well as dynamic simulations. The results of these studies show that this special intermitted - motion linkage has a good usage property. This research shows some theoretic foundations for the practical usage of the intermittent - motion linkage.

Key words Bar - linkage Intermittent - motion Analysis of linkage Adams

Abstract Traditional tolerance analysis methods require an explicit function to describe the relationship between the resultant assembly dimensions and manufactured component dimensions. Such an explicit assembly function may be difficult or impossible to obtain for complex 2 - D assemblies. A new method is presented for tolerance analysis of 2 - D mechanical assemblies based on vector loop - based models. It has a significant advantage over traditional tolerance analysis methods in that it does not require this explicit assembly function. Furthermore , the tolerance analysis problem of 2 - D assemblies can be solved by vector operating to obtain the tolerance sensitivity coefficients. An example is presented to demonstrate the effectiveness of the proposed method.

Key words 2 - D Tolerance analysis Vector loop assembly model Sensitivity coefficient

Abstract A systematic approach to the design and optimization of the ideal universal joint with a given input torque for a given joint angle is developed. Universal joints which use the approach presented here will be ensured not to have interference between the various parts of the mechanism when in operation.

Key words Universal joint Interference Optimum design

Abstract Two new kinds of hand press based on force amplifier composed of form - closed eccentric - cam and toggle are introduced. Their working principles are analyzed , and their mechanical calculating formulae are also given. Both these two new presses consist of form - closed eccentric - cam , so they have advantage of good function of reposition , concise structure and distinct effect in force - amplifying.

Key words Eccentric - cam Form - closed Toggle Force amplifier Hand press

Abstract The method of quick speed calculation of differential gears by designing AuotLISP programs is presented. The method is convenient for selecting differential gears in machining of gears.

Key words Differential Gears AutoLISP Programs Machining Gears

Abstract Metal pushing V - Belt continuously Variable transmission (CVT) is the ideal transmission for automobile , it is also one of the most important research items of researches and automobile companies all over the world. The development history of metal pushing V - Belt CVT is introduced. The researches importance of metal pushing V - Belt CVT is discussed. The current status of the researche is studied. The control strategy developing tendency of CVT electro - hydraulic servo system is analyzed.

Key words Metal pushing V - Belt Continuously variable transmission Current status Developing tendency