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Investigating the Design and Performance Optimization of Spindle Boxes in Modular Machine Tools

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Abstract: This paper presents the left headstock design of a three-side modular drilling machine that machines gearbox front shells. Modular machines rely on the headstock to power all axes. The part's hole number, position, cutting settings, and spindle type determine its design. The motor on the back shell of the headstock powers the spindle primary motion, and the power slipway powers feed movement. The three-side modular drilling machine decreases drilling time, enhancing gearbox front shell production line productivity. Universal pieces make up the headstock, which is developed for specific needs. This article details the left headstock design of this three-sided modular drilling machine. Analysis of gearbox front shell process characteristics. According to the process picture sheet, machining schematic sketch, and machine tool general chart, the original basis chart is drawn along the spindle structure, the transmission system is designed, the coordinates of all the arises are computed, and the coordinate checking drawing is drawn. Finally, after building the entity model, general drawing and pa Finite element analysis of key headstock components are done and after the study, design flaws are suggested.

Keywords: Modular Machine; Gearbox and Headstock; Drilling and CAD Technology; Reduction Box; Spindle Extension and Steering; Shaft Coordinates; Drilling Modular Machine; Gear Factory.

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1. Introduction

1.1. The source, research purpose and significance of the topic

This subject originated from the FAW Car Gear Factory. It is a piece of equipment required at the production line station. Its function is to complete three-sided drilling on the front housing of the gearbox [1]. This equipment is a modular machine tool [3]. The modular machine tool is a semi-automatic or automatic special machine tool based on general components and equipped with special components and fixtures designed according to the workpiece's workpiece's specific shape and process. Modular machine tools generally adopt multi-axis, multi-tool, multi-process, multi-face, or multi-station simultaneous

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processing, and the production efficiency is several to several tens of times higher than that of general-purpose machine tools. Since the common components have been standardized and serialized, they can be flexibly configured as required and shorten the design and manufacturing cycle [4]. Therefore, modular machine tools have the advantages of low cost and high efficiency, are widely used in mass and mass production, and can be used to form automatic production lines [17]. Modular machine tools are generally used for processing box-like or special-shaped parts [5]. The general parts of modular machine tools can be divided into five categories according to their functions: power parts, supporting parts, conveying parts, control parts, and auxiliary parts [6]. The power component is the component that provides the main motion and feed motion for the modular machine tool [7]. This mainly includes a power box, cutting head, and power sliding table.

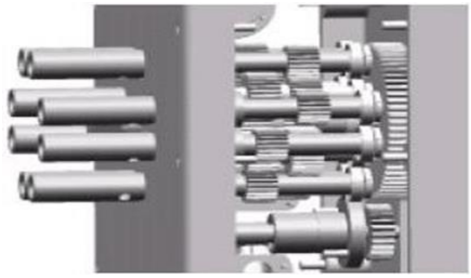

Figure 1: Combined machine tool headstock

The power components are related to the processed parts through the spindle box. Specifically, the spindle box enables each spindle to obtain a certain position and speed [8]. The specific position of the machined part's hole determines the main shaft's position [9]. The motion is transmitted from the drive shaft to the main shaft by a transmission gear arranged in the main shaft box according to a certain speed ratio so that the main shaft can obtain a predetermined speed (Figure 1). The spindle box is usually installed on the power head (or reduction box) and can be fastened to the sliding table or the bed [10]. The specific structure depends on the number, shape, and distribution of the holes in the processed parts. Due to the difference in the relative position, size, and quantity of the holes on the processed parts, it is impossible to make the headstock all universal. Therefore, as a component, the headstock is not a general-purpose component; it is only classified according to the parts to realize the universalization of the headstock parts. For example, the box body front and rear covers of the spindle box are classified according to outline size and shape to ensure the universalization of castings [18]. The main shaft is classified according to use and is divided into several types for drilling, boring, and tapping. Classified according to modulus m, tooth number z, and aperture d. Through these methods, almost all the headstock parts are universalized so that these universal parts can be configured into headstocks with different structures [11].

This time, the task This three-sided drilling combination machine tool is used to process three sets of holes in the front housing of the automobile transmission. Therefore, the combined machine tool is a processing line for the front housing of the transmission. One of the special machine tools is a drilling and boring machine tool, so its headstock is a general-purpose headstock [12]. This design Through the design of the headstock, we can further understand the modular machine tool's modular machine tool's composition structure. Nowadays, modular machine tool industry enterprises mainly provide special equipment for automobiles, motorcycles, internal combustion engines, agricultural machinery, engineering machinery, chemical machinery, military industry, energy, light industry, and home appliances [14]. With my country’s entry into the WTO, it is further integrated with the world’s machine tool industry [13]. The products of industry enterprises have begun to transform into numerical control and flexibility [15]. Judging from the production data of enterprises in the past two years, the market demand for CNC machine tools and machining centers is on the rise, while the traditional drilling, boring, and milling combined machine tools have a downward trend [16]. Although the demand for combined drilling machines is declining gradually, thorough research on basic combined machine tools and their components and the basic principles and technologies.

1.2. Research status and development trend of the subject

1.2.1. Development path

Special machine tools developed with the rise of the automotive industry. In the special machine tool, some parts are gradually developed into general parts due to repeated use, thus resulting in the combined machine tool [19]. The earliest modular machine tool was made in the United States in 1911 for processing automobile parts. Each machine tool manufacturer had a common component in the initial stage. To improve the interchangeability of common components of different manufacturers and facilitate the use and maintenance of users, Ford Motor Company and General Motors of the United States negotiated with the American machine tool manufacturers in 1953 to determine the principle of standardization of common components of modular machine tools, that is, strictly stipulate the size of the connection between the components, but the component structure is not specified. With the development of the automobile industry, the demand for modular machine tools has grown rapidly due to automated production and equipment renewal in some production departments [20]. The American Cross Company produced 700 modular machine tools in a single year in 1967, and the Soviet Union's annual output reached 5,400 in 1969. Later, with the continuous development of modular machine tools in various countries worldwide, some standards were gradually established for common components of modular machine tools. The general headstock is, of course, ignored. The general headstock's design, manufacturing, and assembly use standard and general parts and components, such as gears, spindles, drive shafts, spacers, bearings, box bodies, and front covers.

1.2.2. Research status

1.2.2.1. Research status of abroad

The headstock is an important component of the combined machine tool, which is used to arrange the work spindle of the machine tool and its transmission parts and corresponding additional mechanisms. The design of the spindle box is a very important part of the modular machine tool design process. The quality of its design will directly affect the design quality of the modular machine tool. At present, CAD technology has been widely used in the field of headstock design. Still, in the design process of the headstock, when CAD technology involves program decision-making, evaluation, analysis, important parameter selection, etc., it often requires more manual work. The level of the designer's intervention largely determines the spindle box's design level to a large extent. The key to the design of the headstock CAD system lies in the flexible application of CAD technology to the design and interference inspection of the transmission scheme. Determining the transmission scheme is the key to the whole design, mainly including the distribution of transmission ratio, the selection of gear modules, the grouping of transmission chains, the selection of transmission models, and the determination of gear parameters. The second is the interference inspection process, which mainly inspects the interference between gears, shafts, or sleeves and between gears and casing. Improving the intelligence of these two parts is the key to the headstock CAD system.

Research on CAD technology for modular machine tools abroad has been carried out earlier. In the early 1960s, some industrial developed countries first researched the CAD of the spindle box. Especially since the 1980s, with the development of computer technology, the development and application of interactive graphics and database management systems, the use of CAD technology for modular machine tools has continued to expand, such as the United States INGERSOLL, KINGSBURY, BURGMAST, CINCINNATI, MILACRON, LAMB, CROSS and other machine tool companies, German Huller-Hille (Huller-Hille) company, British Klaus (Cross) company, modular machine tools have developed to CAD/CAM integrated systems and the modular design of the headstock has become mature.

1.2.2.2. Domestic research status

Domestically, in the early 1970s, the Dalian Modular Machine Tool Research Institute began research work in this area. In 1978, modular machine tool CAD technology was listed as a key project of the national machinery industry. Since then, research on CAD technology for modular machine tools in my country has begun. At present, Dalian Machine Tool Plant, Changzhou Machine Tool Plant, Wuhan Machine Tool Group Corporation, Anyang Second Machine Tool Plant, Baoding Second Machine Tool Plant, Jining Machine Tool Plant, and Dalian Modular Machine Tool Research Institute, etc., generally apply CAD technology in modular machine tool design. In recent years, with the increasing maturity of CAD technology, various CAD technologies such as Pro/E, UG, Solid Edge, and AutoCAD have been widely used in modular machine tool design. Among them, the research work on the computer-aided design of the headstock has also made great achievements, and more effective software systems have been developed.

- The headstock CAD system developed by Dalian Modular Machine Tool Works simulates the operation of the headstock through mathematical models, checks all geometric interference, speed, and steering errors of the headstock, and can automatically generate the assembly table, parts list, and assembly drawing of the headstock and parts processing drawings, etc., software design is scientific and rigorous.
- Based on the design characteristics of modular machine tools combined with advanced design technology, Henan University of Science and Technology carried out secondary development on the three-dimensional design software UG platform. It established a modular machine tool CAD system. Among them, the design of the spindle box includes the design and the transmission system has two parts. The headstock design module is on the design platform of the overall assembly model of the headstock and accepts the data input by the user through a friendly human-computer interaction interface. This software adopts a three-dimensional form to express the transmission system of the headstock, which is convenient for interference inspection and parameter modification.
- Guangxi University of Technology uses the 3D CAD software Solid Edge9.0 to develop a 3D variable CAD system for the headstock. The so-called variable design is based on the user's design level. It expresses the size and geometric

relationships of the graphics in the form of variables. By assigning values to the graphics variables, the graphics can be generated or modified to obtain the desired Design graphics.

- Huangshi Technical College uses AutoCAD to develop CAI courseware for the overall scheme design of modular machine tool headstock transmission. The software uses AutoCAD's embedded Auto LISP language programming to realize the secondary development of AutoCAD. Establish a design menu in AutoCAD, which includes design software description, use method selection, box selection, shaft diameter calculation, transmission scheme design, interference check, and output results. Among them, the key modules are transmission scheme design and interference check. This CAI courseware obtains the design results of the spindle box through a friendly human-computer interaction interface, which is conducive to mastering the design method and avoids complicated calculations, greatly improving learning efficiency, and is suitable for teaching.

In addition, Yanshan University, Yangzhou University, Yangzhou Diesel Engine Factory, North China Electric Power University, Yuzhou University, and other departments have also developed CAD design software for the headstock, assisting in completing the whole process from modular machine tool headstock design to drawing.

1.2.3. Development trend

Because of the functions and characteristics of modular machine tools, they are important equipment for the high efficiency, high quality, and economical production of large quantities of mechanical products. With the development of the market economy, as the main application objects of modular machine tools, industries such as automobiles, tractors, engines, motorcycles, and their engines have undergone more profound changes, and the product life cycle continues to shorten. The variety is increasing day by day. Different processing products, different combined machine tools, and different headstocks. Engineering and technical personnel must continuously improve product development capabilities to meet market requirements. Therefore, how to design the spindle box efficiently and with high quality has become the goal of modular machine tool manufacturers. Regarding the current research status, CAD technology has been widely used in headstock design, providing designers with convenient, fast, and efficient design tools. Still, its degree of intelligence is not high. Developing hardware, software, intelligence, and other technologies provides the necessary technical foundation for developing more advanced headstock design software. Regarding the current research status at home and abroad, although there is much design software for the headstock, the design software related to the expert system has not been seen. It is still in the modular design stage, with no intelligent CAD design system. If the expert system technology is introduced into the headstock CAD system, the experience of experts can be used in the whole design process to imitate human behavior for thinking and reasoning. This will be a headstock design system that conforms to the main development trend of CAD.

1.3. Main research and design content

This design refers to the structure of the general headstock of existing machine tools and modular machine tools, and it designs a left headstock of a three-sided drilling modular machine tool. The content focuses on structural design, including overall design and component design. It includes the selection of electric motors, the determination and calculation of mechanical transmission system schemes, and the selection of other auxiliary components. The design of the headstock mainly includes the following aspects:

- Draw the original basis diagram of the headstock design
- Determination of main shaft and gear and power calculation
- Design and calculation of transmission system scheme
- Calculation of the coordinates of the spindle box and drawing a coordinate inspection chart
- Draw a general drawing of the headstock and parts drawing
- Establish a three-dimensional solid model of the headstock and perform finite element analysis on typical components.

Detailed design calculations and checks are made for the above 6 parts, and finally, the mechanical structure of the entire headstock is designed and expressed.

1.4. Structure of this article

This article can be divided into four parts: The first part is the introduction, which mainly introduces the research purpose and research significance of the headstock design, the development process, the domestic and foreign development status, and the development trend; the second part is about the parts to be processed (the front housing of the transmission) analysis of the drilling process; the third part carries out detailed design calculations and check calculations for the specific structure of the headstock; the fourth part carries on the modeling of the three- dimensional entity of the headstock and carries on the finite element to the analysis of the important part.

2. Drafting of Process Plan

This three-sided drilling machine tool is used to process several sets of holes in the front housing of the transmission. This briefly analyses the processing technology of this part.

2.1. Characteristic analysis of parts

The processed part is the front housing of the S1001 transmission, which belongs to the box body part, and its outline structure is shown in Figure 2.

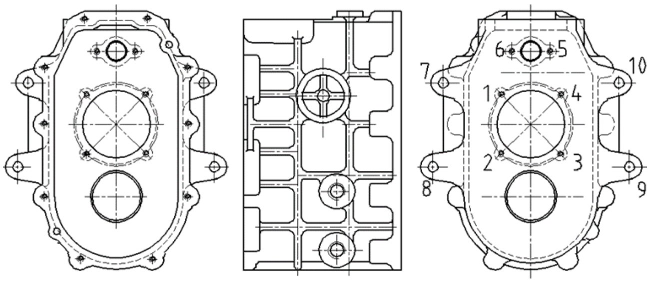

Figure 2: The box body part and its outline structure

The left headstock of the three-sided drilling combination machine tool designed in this paper is used to process the 1~10 holes in the left view of Figure 2, of which 6 holes 1~6 are M8 threaded holes, and only drilling is performed in this process. Hole, then tapped in the subsequent process, the 4 holes from 7 to 10 are holes with a diameter of. The three-sided drilling combination machine tool's right headstock and rear headstock process a group of holes on the right side of the front shell (a group of holes in the right view in Figure 2) and a group of holes on the upper part.

2.2. Tool selection and cutting amount determination

2.2.1. Tool selection

The holes on the left side of the front housing of the transmission can be divided into three groups: No. 1~4 are a group, 5 and 6 holes are a group, and No. 7~10 holes can be regarded as a group. No. 1~6 holes are all M8 threaded holes. This process is drilled first, and the next process is tapped. The diameter of the drilled hole is as follows. Therefore, the tool selects a stepped twist drill for drilling before tapping (Figure 3). For the D2 selection, the national standard is GB/T 6139-2007, the tool material is high-speed steel, and the rotation direction is right-handed.

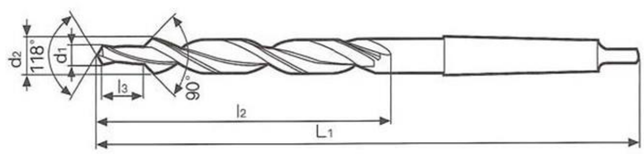

Figure 3: Housing of the transmission

The taper shank twist drills selected for holes 7~10, according to the latest national standard GB/T 1438.1-2008, check the size parameters of the tool. The tool material is high-speed steel, and the direction of rotation is right-handed.

2.2.2. determination of cutting parameters

First, select a more reasonable speed according to the selected tool (the unit is r / min). The feed per revolution of the tool is f （the unit is mm / r ）. Determine and adjust the feed and speed per revolution according to the so-called “restrictive tools” that have the longest working time and the largest load and are difficult to sharpen. Modular machine tools usually use a power slide to drive the tool feed. Therefore, the feed per minute of all tools on the spindle box driven by the same sliding table equals the sliding table's working speed. Which is, $n1f1= n2f2=\ldots= nifi=vf$ Check the cutting consumption of high-speed steel twist drills when drilling holes from the "Machining Process Manual" as follows：

The same is equal to the working speed of the sliding table. Which is, $n_1f_1= n_2f_2=\ldots= n_if_i=v_f$ Check the cutting consumption of high-speed steel twist drills when drilling holes from the "Machining Process Manual" as follows：

for Ø6.8hole, Feed rate f=0.1～0.18mm/r, v =10～18m/min, here take f_1=0.13mm/r, n_1=710r/min；

for Ø13hole, Feed ratef=0.18～0.25mm/r, v =10～18m/min, Take here n_2=400r/min, by $n_1f_1 = n_2f_2$ Available $f_2 = n_1f_1/n_2$=0.13 mm/r.

Ø6.8 The cutting speed of the drill is

$$V_1 \frac{\pi n1d1}{1000} = \frac{\pi\times710\times6.8}{1000} = 15.17\text{m/min}$$

Ø13 The cutting speed of the drill is

$$V_2 \frac{\pi n2d2}{1000} = \frac{\pi\times400\times13}{1000} = 16.34\text{m/min}$$

It can be seen from the wall thickness of the transmission front housing，1~6 The drilling depth of hole No. is 12mm，7~10. The drilling depth of hole No. is 18mm.

2.3. Positioning analysis and benchmark selection

Before drilling, the right side of the transmission front housing (the right side in the front view of Figure 4) has been finished and used as a precision datum. Three degrees of freedom are restricted on this surface: rotation around the axis, rotation around the axis, and movement along the axis; the flying process is also processed before this process. Two positioning holes are in the right view's lower left and upper right corners. The positioning hole in the upper right corner restricts the movement along the axis and the movement along the axis. The positioning hole in the lower left corner restricts the freedom of rotation around the axis (figure) 4). In this way, the six degrees of freedom of the front housing are restricted, which meets the positioning requirements.

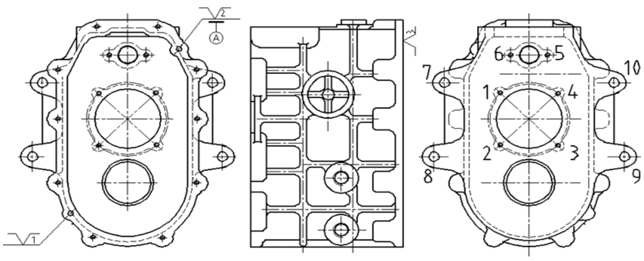

Figure 4: Location scheme diagram

The processed 1~4 holes, 5~6 holes, and 7~10 holes have certain position requirements relative to the axis A of the reference hole in the upper right corner of the right view in Figure 4. This analyses the process characteristics of the parts to be processed and clarifies the processing tasks of this process. Then, according to the material and other conditions of the part, the tool to be used for processing was selected. Then, the cutting amount was determined according to the selected tool and productivity. Finally, the benchmark selection and positioning plan of this process are determined.

3. Selection of Spindle Box Spindle Structure Type and Power

3.1. Draw the original basis diagram of the headstock design

The design of the spindle box is based on three pictures and one card, namely the general drawing of the machine tool or the contact size drawing of the machine tool Process drawing, the processing schematic diagram, and the productivity calculation card of processed parts. For the design of the headstock, the original basis diagram of the headstock design is usually drawn according to three pictures and one card (Figure 5). The drawing of the original basis diagram of the headstock design, including the following：

Determination of the positional relationship size of all spindles: The parts to be processed by this three-sided drilling combination machine tool are a group of holes on the left side of the front housing of the transmission. The parts drawing of the front housing can determine the positional relationship between the various spindles.

Determination of spindle speed and steering: The spindle speed was determined when the cutting parameters were determined. For ∅6.8 hole，$n1$ =710r/min；for ∅13 hole, $n2$ =400r/min。Since both taper shank step drills and taper shank twist drills are right-handed, the rotation direction of the spindle is all clockwise (looking from the back of the spindle box). Determination of the size of the spindle extension Schematic diagram of the size of the spindle extension part 3.1

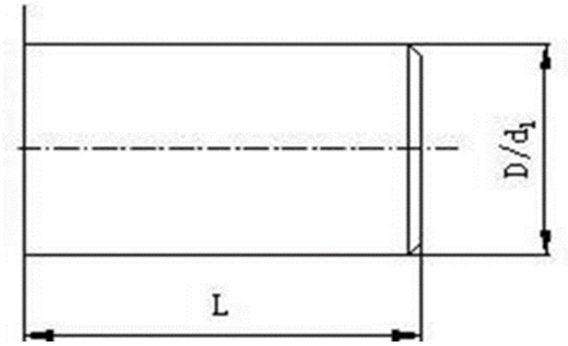

Figure 5: Schematic diagram of the size of the spindle extension part

The dimensions of the extension part of the main shaft can be initially obtained according to the series parameter table 3-6 of the general main shaft in the “Concise Manual for Modular Machine Tool Design”:

∅6.8 Outer size of hole $D/d1$for 30/20mm，L=115mm;
∅13Outer size of hole $D/d1$for 38/26mm，L=115mm.

Based on the above content, combined with the outline dimensions of the headstock and the contact dimensions of other related parts, the model of the power component draws the original basis diagram of the headstock design Table 1.

1. Parts to be processed

Number and name: front housing-transmission material and hardness： HT200, HBS=170~220

2. Spindle extension size and cutting amount

Table 1: Spindle extension size and cutting amount

Shaft number	Spindle extension size			Cutting consumption		
	D/d1/mm	L/mm	Work order	n /r/min	v /m/min	f /mm/r
1~6	30/20	115	Drill 6.8 hole	710	15.17	0.13
7~10	38/26	115	Drill 13 hole	400	16.34	0.23

3. Power components：Y132S-4B5 Motor Power P=5.5kW, Rotating speed n=1440r/min

3.2. Determination of spindle type and diameter and checking calculation of power required by spindle box

The headstock spindles of the three-sided drilling modular machine tool designed in this paper are all used in the drilling process, so the spindle must withstand the axial force in one direction. Therefore, ball-bearing spindles are selected for the main shaft. The front-end supports are thrust ball bearings and radial ball bearings, and the rear supports can be radial ball bearings or tapered roller bearings. This spindle box selects radial ball bearings as the rear support. Each spindle's diameter and the spindle box's required power are calculated as follows.

3.2.1. Calculation of Spindle Diameter

The torque calculation formula for drilling processing obtained from the "Machining Process Manual" is:

$$M=C_M d_0^{Z_M} f^{y_M} k_M \tag{3.1}$$

Where：C_M—Cutting torque coefficient depends on the material being processed and cutting conditions

d_0— Diameter of processed hole

Z_M— Processed aperture index

f— Processing feed

y_M— Feed Index

k_M— The machining condition correction coefficient is related to the processed material, tool sharpening shape, and tool bluntness

Axial force calculation formula;

$$F=C_F\ d_0^{Z_F}\ f^{y_F} k_F \tag{3.2}$$

Where：C_F— The cutting axial force coefficient depends on the processed material and cutting conditions.

d_0— Diameter of processed hole

Z_F— Processed aperture index

f — Processing feed

y_F— Feed Index.

k_F— The machining condition correction coefficient is related to the processed material, tool sharpening shape, and tool bluntness.

The cutting power calculation formula i

$$P_m=\frac{M_V}{30d_0} \tag{3.3}$$

When the high-speed steel drill bit is used to process cast iron, the coefficients and indexes are as follows：

CM=0.206, ZM=2.0，yM=0.8， kM=1.0；
CF=420, ZF=1.0， yF=0.8， kF=1.3

Drill 1～6 number 6 individual □6.8 hole Know the cutting speed from the previous calculation $V1$=15.17m/min， Feed Rate f_1=0.13mm/r， Cutting speed $v1$Feed rate f_1 and the above coefficients and index bands (3-1) 、 (3-2) 、 (3-3) In the formula ：

$F_1=CFd^{z}_{0\ 1}\ fyFkF$=420× $6.8^{1.0}$ × $0.13^{0.8}_{01}$ ×1.3=725.9N

$M_1=C_M d^{Z_M} f^{y_M} k_M$=0.206× $6.8^{2.0}$ × $0.13^{0.8}$ × 1.0 = 1.862Nm

$$P_{m_1}=\frac{M_1v_1}{30_{d01}} = \frac{1.862\times15.17}{30\times6.8} = 0.1384\text{kW}$$

Select the shaft diameter according to the calculated torque look-up table and take the spindle diameter=15mm. Yes, but due to the spindle's rigidity, the spindle box's original requirement, according to the figure, is 30/20, so take d=20mm. Drill 7~10 Number 4 Holes ∅13 Cutting speed $v2$=16.34m/min, Feed rate f=0.23 mm/r, Cutting speed $v2$ Feed Rate f $_2$ And the above coefficients and indices are substituted into the above formula (3-1) 、 (3-2) 、 (3-3)

$F_2 = C_F\ d^{Z_F}_{02}\ f^{y_F}_{2}\ k_F$=420× $13^{1.0}$ × $0.23^{0.8}$ × 1.3 = 2190.4N

$M_2=C_M d^{Z_M}_{02}\ f^{y_M}_{2} k_M$=0.206× $13^{2.0}$ × $0.23^{0.8}$ × 1.0 = 10.74Nm

$$P_m = \frac{M2v2}{} = \frac{10.74\times16.34}{} = 0.45\text{kw}$$

30_{d02} 30×13

Look up the spindle diameter=25mm, Spindle extension part sized/d_1 take 38/26mm

3.2.2. Checking calculation of power required by headstock

The power calculation of the headstock includes the power and feed force required by the headstock. The power and feed force are calculated separately below.

3.2.2.1. calculation of power required by headstock

Check the relevant manual to know the required power of the headstock P_{main}

Calculated according to the following formula:

$$P\ main = P_{cutting} + P_{Idling} + P_{lose} = \sum_{i=1}^{n} P_{cutting\ i} + \sum_{i=1}^{n} P_{Idling\ i} + \sum_{i=1}^{n} P_{lose\ i} \quad (3.4)$$

Where：
$P_{cutting}$ — — Cutting power, the unit is kW;
P_{Idling}— —Idling power, the unit is kW；
P_{lose}— —The power loss is proportional to the load. The unit is kW. The cutting power of the spindle box is;

$P_{cutting}$= 6P_{ml} + 4P_{m2} = 0.1384 × 6 + 0.45 × 4 = 2.63kW Check the "Concise Manual of Modular Machine Tool Design" for the idling power meter of the spindle box shaft：

Spindle 1~6 idle power of each axis P_{Idling} 1=0.074 kW;
Spindle 7~10 Idling power of each axis $P_{Idling\ 2}$=0.046 kW；
Therefore, the total idling power of all spindles is：
$P_{main\ Idling}$=6× $P_{Idling1}$ + $P_{Idling2}$ =6× 0.0074 + 4 × 0.046 = 0.628kw

According to the idling power of the main shaft, it is good to estimate the idling power of all drive shafts,00.8kW. Then, the total idling power of the headstock is：
$P_{Idling} = P_{main\ Idling} + P_{pass\ Idling}$=0.628+0.8=1.428kW

To calculate the power loss of the headstock, it is better to take the total transmission efficiency of the headstock η for0.85. Then, the power loss of the headstock is：

P_{loss}=$P_{cutting}$×(1-η) = 2.63 × (1-0.85) = 0.3945 kW Bring the power into the formula (3.4) in, the total power of the headstock is：
P_{main}=$P_{cutting}$ + P_{Idling} + P_{loss}=2.63+1.428 + 0.3945 = 4.47kW

According to this calculation, the motor selection of 5.5 kW is considered suitable.

3.2.2.2. calculation of feed force required by headstock

The required feed force of the headstock $Fmulti{-}axis\ box$（The unit is N) can be calculated according to the following formula);

$$F_{multi-axis\ box} = \sum_{i=1}^{n} F_i \quad (3.5)$$

Taking the calculated feed force of each spindle into equation (3-5), we get:

$Fmulti{-}axis\ box = \sum_{i=1}^{n} F_i$= 6×725.9+4×2190.4= 13117 N

In practice, to overcome the frictional resistance caused by the movement of the sliding table, the feed force of the power sliding table should be greater than $F_{multi-axis\ box}$. This first draws the original basic diagram of the spindle box design based on three pictures and one card; then, the spindle type is selected according to the processing conditions, and the torque on the spindle is calculated to determine the spindle diameter. Finally, the power check of the headstock is carried out, which includes two contents: power calculation and axial force calculation.

4. Design and Calculation of Spindle Box Transmission System

The design of the spindle box transmission system is based on the position and speed of the drive shaft of the power box or motor, the position of each spindle and its speed requirements, the design of the transmission chain, the drive shaft and each spindle are connected, so that each spindle can obtain a predetermined speed and steering.

4.1. Calculate the coordinate size of the drive shaft and main shaft

Calculate the coordinate size of the drive shaft and the spindle according to the original basis diagram of the spindle box design. A coordinate origin must be selected for the spindle box to calculate the coordinates of each axis of the spindle box. Since the headstock of the three-sided drilling combination machine tool is installed on the power sliding table, the intersection of the bottom surface of the headstock and the vertical line passing through the positioning pinhole is generally selected as the coordinate origin. In this way, the coordinate values of drive axis O and each spindle can be obtained by calculation as shown in Table 2.

Table 2: Coordinate values of drive shaft and spindle

Coordinate	**Drive**	**Pin O1**	**Spindle 1**	**Spindle 2**	**Spindle 3**	**Spindle 4**
X	285	0.000	250.352	319.648	319.648	250.352
Y	315	30.000	180.352	185.352	249.648	249.648
coordinate	Spindle	Spindle	spindle7	Spindle 8	Spindle 9	Spindle
X	199	199.97	237.000	335.000	335.000	237.000
Y	193	237.00	113.000	101.000	329.000	317.000

4.2. Design of the spindle box drive system

Generally speaking, several schemes can be designed for the transmission system of the same headstock. Therefore, various transmission schemes must be compared during design, and the best scheme must be selected. Whether the design of the transmission system is good or not, it will directly affect the quality of the spindle box, the degree of generalization, the size of the design and manufacturing workload, and the cost level. Therefore, the spindle box transmission system design has some general requirements.

4.2.1. General requirements of spindle box drive system [2]

- Under the condition of ensuring the strength, rigidity, speed, and steering of the main shaft, strive to minimize the specifications and number of the drive shaft and teeth. For this reason, one intermediate drive shaft should be used to drive multiple main shafts, and the gears should be arranged in the same row. When the centre distance does not meet the standard, the method of shifting gear can solve the problem.
- Try not to use the spindle to drive it so as not to increase the spindle load and affect the processing quality. Make the structure compact, the transmission ratio of the gear pair in the main shaft box is generally greater than 1/2 (the best transmission ratio is 1～1/1.5), and the transmission ratio of the gear pair in the back cover is allowed to be 1/3～1/3.5; try to avoid speed up drive).
- The gears used for the rough machining spindle should be arranged in row I as much as possible to reduce the torsional deformation of the spindle; the gears on the finishing spindle should be arranged in row III to reduce the bending deformation of the spindle end.
- When there is a rough and finishing spindle in the spindle box, it is best to start with the power box drive shaft gear transmission and divide it into two transmission routes to avoid affecting the machining accuracy. The number of rotating shafts directly driven cannot exceed two so as not to cause difficulties in assembly.

4.2.2. Drafting the spindle box transmission system

The basic method of drawing up the spindle box transmission system is: first, distribute all the spindle centres as much as possible on several concentric circles, and set the central transmission shaft on the centre of each concentric circle; some axes distributed in non-concentric circles should also be set in the middle Driveshaft; then take concentric circles according to the selected central drive shafts, and use the fewest drive shafts to drive these central drive shafts; finally, connect the drive shafts with the power box drive shafts by closing the drive shafts [2]. First, determine the position of the hole to be machined. From the original basis diagram of the spindle box design, the position distribution of the spindle on the box body can be obtained, as shown in Figure 6.

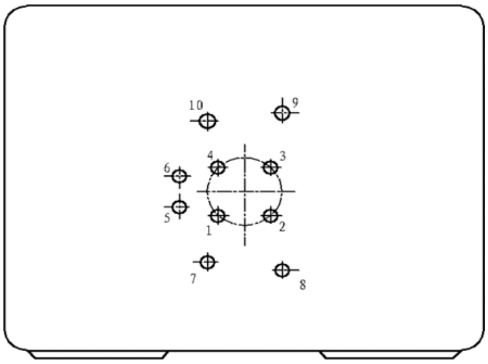

Figure 6: Distribution diagram of the spindle of the headstock

To determine the position of the drive shaft, observe the position distribution of these 10 main shafts. They are all concentrated. Therefore, the position of the drive shaft should be set between these main shafts. This can shorten the transmission chain and facilitate the motor installation on the box cover. So you might as well put the drive shaft O Set on the same vertical line as the centre of the concentric circle where the main shafts 1, 2, 3, and 4 are located. Considering the motor's installation space and ease of installation, the drive shaft is set above the circle's centre at a distance of 9 and 10. The relatively close position of the spindle. Although the main shafts 9 and 10 are relatively close to the drive shaft O, the main shaft cannot directly drive the 9 and 10 shafts. In addition, the main shaft has to drive other drive shafts, resulting in too many shafts driven by the drive shaft. On the one hand, the load is too large.

On the other hand, it brings difficulties to assembly. Therefore, a transmission shaft 11 can be set directly above the drive shaft. The drive shaft first drives the transmission shaft 11 to rotate, and then the transmission shaft 11 transmits the motion to other shafts. If the transmission shaft 11 directly drives the main shaft 9 and the main shaft 10, for the same reason, this will bring a relatively large load to the transmission shaft 11 and cause installation difficulty. It can be seen from this that two more intermediate transmission shafts must be provided, one to drive the main shaft 10 and the other to drive the main shaft 9. The transmission shaft that drives the main shaft 10 is denoted as 12, and its approximate position is set on the left side of the transmission shaft 11. The other transmission shaft is distributed on the right side of the transmission shaft 11 to divide it into two transmission chains. The main shafts 5 and 6 are on the same vertical line, so a central drive shaft 14 is set on the vertical bisecting line connecting the centres of 5 and 6 because the centre distance between the 12 shafts and the 14 shafts is relatively large. It may be reasonable In the direction of rotation, an intermediate transmission shaft 13 must be set between the two, and the transmission shaft 13 is driven by the transmission shaft 12 to move.

Then, the transmission shaft 13 drives the transmission shaft 14 to move. In addition, the movement of the main shaft 7 must be driven by this transmission chain. Since the main shaft box will be equipped with an oil pump in the future, transmission shaft 15 can be installed at the lower left of transmission shaft 14. The transmission shaft 15 drives the oil pump shaft and the transmission shaft 16, respectively, and then the transmission shaft 16 drives the main shaft 7 to rotate. The main shafts 1, 2, 3, and 4 are distributed on a concentric circle, so a central drive shaft 19 is set at their centre. A transmission shaft 18 is arranged on the right side of transmission shaft 19, and the transmission shaft 18 drives the transmission shaft 19 to rotate. A transmission shaft 17 is arranged between the transmission shaft 18 and the main shaft 8, and the transmission shaft 18 drives the main shaft 8 to rotate through the transmission shaft 17. The intermediate drive shaft on the right side of the drive shaft 11 is taken as the drive shaft 22. While the shaft 22 drives the main shaft 9 to rotate, the intermediate drive shaft drives the drive shaft 18 to rotate. To obtain the rotation direction of the main shaft required by the design and avoid the excessive radial size of the gear on the transmission shaft, the transmission from shaft 22 to shaft 18 is realized by a two-stage transmission. Therefore, the transmission shaft 20 and the shaft 22 are arranged between the shaft 18 and the shaft 22. Transmission shaft 21. So far, the design of the entire headstock transmission system is completed, and the tree diagram of the headstock transmission is drawn (Figure 7). In the figure, main shafts 1-10 are "tree tops," drive shaft O is "tree roots," and the bifurcation points are drive shafts 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21, 22, the transmission pair between each axis is "tree branch," and the arrow indicates the direction of movement.

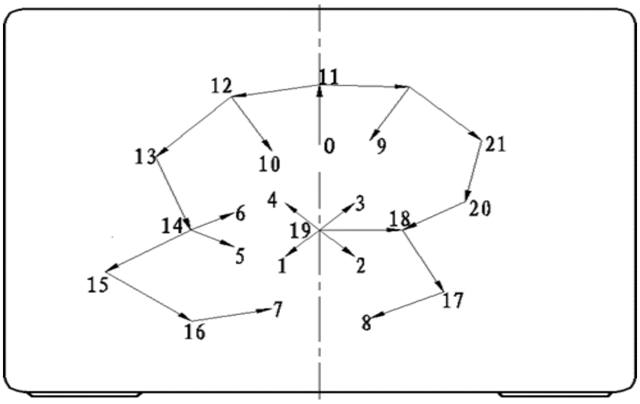

Figure 7: Tree diagram of spindle box transmission

4.3. Determine the position of the drive shaft and the number of gear teeth

After the route design of the main shaft box transmission system is completed, the specific positions of other transmission shafts and the number of gear teeth and modules on all shafts are determined according to the known positions of the main shaft and drive shaft. Determine the position of the drive shaft 19 and the number of teeth of each gear. It is easy to determine that the position of the drive shaft 19 is the centre of the concentric circles of spindles 1, 2, 3, and 4. Gear modulus (unit: mm) can be estimated according to the following formula：

$$m \geq (30\sim32)\sqrt{\ ^{P\,3}}\quad — \tag{4.1}$$

Where：

P —The power transmitted by the gear, the unit is kW；

Z —Number of pinion teeth in a pair of meshing gears；

n —The rotation speed of the pinion, the unit is r/min

The number of gear teeth on the spindle is 1 = 24 (set in row I). From the previous calculation, we can see: spindle 1 The power transmitted by the upper gear = 0.1384kW, and the various values are brought into the above gear modulus checking formula, and then:

$M \geq 32 \times \sqrt{\ }$

To ensure the tooth root strength of the gear, the thickness from the tooth root to the wall of the hole or keyway should be made *2m.* Therefore, taking the modulus of this wheel, it is easy to get the centre distance between the transmission shaft 19 and the main shaft 1 = 49mm, then the number of gear teeth on the transmission shaft 19 and the rotation speed of the transmission shaft 19 are:

$z^I_{19} = \frac{2A_{1-19}}{m} - z^I_1 = \frac{2\times49}{2}$ -24=25 Located in row I

$n_{19} =$

Since the main shafts 1, 2, 3, and 4 are on a concentric circle, and the speed of each main shaft is the same, the position relative to the drive shaft 19 is also similar. Therefore, the gear pair between the main shafts 2, 3, and 4 and the drive shaft 19 is completely the same as the main shaft. The gear pair between 1 and the drive shaft 19 is the same. You can get the following ：

$z^I_3 = z^I_1$=24, m =2 Located in row I

$z^{II}_2 = z^{II}_4 = z^I_1$=24, m =2 Located in row I

z^{II}_{1}=24, m=2 Located in row III

Determine the position of the drive shaft 14 and the number of gear pairs between the main shaft 5 and 6. The centre of the transmission shaft 14 is taken on the vertical bisector of the line connecting the centres of the main shafts 5 and 6, and the horizontal position can be determined after the number of gear pairs is determined. The gear ratio between the transmission shaft 14 and the main shaft 5 and 6:

u_{14-5}= u_{14-6}= u_{19-1}, The gear pair parameters are as follows

z^{III}_{5}=24, m=2 Located in row II

z^{II}_{6}=24, m=2 Located in row I

$z^{II}_{14}=z^{III}_{14}$=25, m=2 Located in the second and third rows, respectively

$n14=n5$ 5 = 710×25

The centre distance between two axes A_{5-14}=A_{6-14}= 49 mm; from this, the position of shaft 14 can be determined. Determine the position of the drive shaft 18 and the number of gear pairs with the drive shafts 17 and 19. The centre of transmission shaft 18 is taken on the same horizontal line as the centre of the transmission shaft 19, and the gear parameters on transmission shaft 19 are taken as follows:

z^{IV}=31, m =3 Located in row IV

Take the transmission ratio i_{19-18}= 1.3. Then, the gear parameters on shaft 18 and the shaft speed are as follows：

$$z^{IV}_{18}=\frac{z^{IV}_{19}}{i_{19-18}}=23.8, \text{Take } z^{IV}_{18}=24, m=3 \text{ Located in rowIV}$$

$$n_{18}=n_{19}\frac{z^{IV}_{19}}{z^{IV}_{18}}=682\times\frac{3}{24}$$

Center distance of drive shaft 18 and 19, $A_{18-19}=\frac{m}{2}(z^{IV}_{18}+z^{IV}_{19})$ = 82.5

In this way, the position of the drive shaft 18 is also determined. When the position of shaft 18 is determined, the large gear on shaft 17 meshes with the gear on shaft 18, and the small gear on shaft 17 meshes with the gear on the main shaft 8. The gear parameters on the main shaft 8 are taken as follows：

z^{I}_{8}=32， m =3 Located in row I

Take the transmission ratio i_{8-17}=1.45, The axis 17 Number of pinion teeth on:

$$z^{I}=\frac{z}{i}=\frac{32}{}=22.07,$$

take z^{I} =22， m =3 Located in a row I Center distance between axis 8 and axis 17 A

$$=\frac{m}{}(z^{I}+z^{I})=81mm$$

Speed of shaft 17 $n_{17}=n8\frac{8}{}$ = 582 r/min

The number of teeth of large gear on shaft 17 $z^{IV}_{17}=\frac{n18z18}{n_{17}}$=36.3, so the gear parameters on axis 17

z^{IV}_{1}=36, m=3 Located in row IV

Correct the speed of the lower shaft 17， $n_{17}= n_{18} \frac{z^I_{18}}{z^I_{17}} = 587$ r/min

The centre distance between shaft 18 and shaft 17 $A_{18-17} = \frac{m}{2}(z^{IV}_{18}+ z^{IV}_{17}) = 90$mm, that is, the distance between shaft 17 and shafts 8 and 18 has been determined, so the position of the shaft 17 is also determined. Determine the position of the drive shaft 11 and the number of gear pairs between the drive shaft O and the drive shaft 22. First, take the number of gear teeth on the drive shaft O z_o=22, m=3 Located in row I. Drive shaft speed no=1440r/min, From the drive shaft O to the main shaft 9 after three levels of speed reduction to reach n_9= n_{10}= 400r/min, Total gear ratio I = 1/ 3.6， Might as well take the first reduction gear ratio i_{O-11}= 22/36 Then the gear parameters on drive shaft 11 and drive shaft O are as follows：

z^{IV}_1=36, m=3 Located in row IV

z^{IV}_0=22, m=3 Located in row IV

Take the second-stage reduction gear ratio as i_{11-22}=24/36. The parameters of a pair of gears meshing on the two shafts are as follows:

z^{II}_1 =24, m=3 Located in row II

z^{II}_2 =36， m=3 Located in row II

Speed of shaft 11, $n_{11}= n_0 . i_{0-11}$=880r/min

Then, the third-stage reduction gear ratio $i_{22\text{-}9} = \frac{1}{i_{11-22} . i_{0-11}} = \frac{1}{1.46}$, Take the parameters of the pinion on the drive shaft 22 as follows:

z^I_2=22， m=3 Located in row I

The number of gear teeth on the main shaft 9 $z^I = z^I$ 1 = 32.12, take.

z^I_9=32, m=3 Located in row I

In the same way, the number and modulus of gear teeth on shaft 10 and shaft 11 are as follows:

z^{III}_1=32, m=3 Located in row I

z^{III}_1=22, m=3 Located in row III

Centre distance between Drive Shaft 11 and Drive Shaft O

$A_{11-0} = \frac{m}{2}(z^I_{11} + z^I_0)$ =87mm in addition to the condition that the shaft 11 and

The drive shaft O is on the same vertical line, and the position of shaft 11 is completely determined. The following centre distance can be determined:

$A_{11-22} = A_{11-12} = \frac{m}{2}(z^{II}_{11} + z^{II}_{22})$=90mm

$A_{9-22} = m/2\ (z^I_9 + z^I_{22})$ = 81mm

$A = m\ (z^{III}_{10} + z^{III}_{12})$ =81mm

Therefore, the positions of the drive shafts 22 and 12 are determined. Determine the number of gear pairs between the drive shaft 21 and shaft 22 and the drive shaft 20 and 18.

$$n_{22}=n_{11} . i_{11-22}=587 r/min$$

$$i_{22-18}= \underline{n_{18}} = 1.5$$

$n_2 2$

due to $i_{22-18}=i_{22-21}.i_{21-20}.i_{20-18}$, take i21-20=1.25, i20-18= 1

then $i_{22-21}=\frac{i_{22-18}}{i_{21-20}.i_{20-18}}=1.2$

From this, the parameters of each gear can be determined as follows：

$z^{IV}{}_2$=24， m=3 Located in row IV

$z^{IV}{}_2$=30, m=3 Located in row IV

$z^{II}{}_2$=36, m=3 Located in row IV

Determine the position of the drive shaft 13 and the number of teeth of the gear pair on the 14th shaft:

$n_{12}=n_{22}=587r/min$

Then12 Axis to 14 shaft transmission ratio $i_{12-14}=\frac{n14}{n12}=1.162$, Take $_{12-13}$=1, Available$_{12-13}$= 1.162

That is, the gear on shaft 13 is a fallen wheel, which only serves to change the direction of rotation. So, the gear on axis 13

$z^{II}{}_1$ =36，m=3 Located in row IV

$z^{IV}{}_{14}=\frac{n_{12}}{i13-14}=30.9$ Take $Z^{IV}{}_{14}$=31，m=3 Located in row IV

$A_{13\text{-}12}$ = m/2($z^{II}{}_{13}+z^{IV}{}_{12}$) =108mm

$A_{13\text{-}14}=m/2\ (z^{IV}{}_{13}+Z^{IV}{}_{14})=100.5mm$

Therefore, the position of axis 13 is also uniquely determined.

4.3.1. Determine the position of the drive shafts 15, 16 and the number of teeth of each gear

Transmission ratio from drive shaft 14 to main shaft 7 $i_{14-7}=\frac{n_{14}}{n_{17}}=1.705$, Take $i_{15\text{-}14}$ = 36/31, $i_{16\text{-}15}$ =1

Then, the number of teeth and modulus of the large gear on the drive shaft 16 and the gear on the main shaft 7 are as follows:

$z^{III}{}_1$=22， m=3 Located in row III

$$i_{7-16}=\frac{i_{14-7}}{i_{16-15}\ i}=1.468$$

$z^{III}{}_7$= 32， m=3 Located in row III

The number of pinion teeth on shaft 16 $z^{III}{}_{16}=z^{III}{}_7.i_{16-7}\ z^{III}$ =21.8. Take $z^{III}{}_1$=22，m=3 located in row III. To further calculate the centre distance of the gear on the relevant shaft and the shaft speed as follows:

$A_{7\text{-}16}=m/2\ (z^{III}{}_7+z^{III}{}_{16})$ =81mm
$A_{15\text{-}14}=m/2(z^{IV}{}_{15}+z^{IV}{}_{14})$ =100.5mm
$n_{15}=n_{14}\times i_{15-14}$ = 587r/min
$n_{16}=n_{15}$= 587r/min

4.3.2. Determination of the position of the lubrication pump shaft 23

The main shaft box of the three-sided drilling combination machine tool adopts a vane lubricating oil pump for lubrication, and the oil pumped out by the oil pump is divided into various lubrication parts through an oil separator. For the horizontal standard headstock, the gears between the front and rear walls of the headstock and the bearings on the walls are lubricated with oil pans, and the gears between the box body and the rear cover and the front cover are lubricated with oil pipes. Since the headstock of the combined machine tool is a medium-sized headstock, only one vane lubricating oil pump is enough. The vane oil pump uses an R12-1 miniature lubricating oil pump with a compact structure and reliable performance. The theoretical speed of the selected vane oil pump is n=710r/min, and its placement position should be as close as possible to the oil sump to make it easy to pump oil. The pump is directly driven by a gear mounted on the pump shaft. According to the transmission system of the spindle box and the box space, choose to place the vane lubricating oil pump directly above shaft 15, namely the vane lubricating oil pump shaft centre and the drive shaft 15. The centre is located on the same vertical line and the same horizontal line as the centre of the transmission shaft 14 so that the position of the oil pump shaft is determined. Install the lubricating oil pump on the front wall of the box, and place the gears of the oil pump in row I, which is convenient for maintenance. The gear on the pump shaft is driven by the gear on the transmission shaft 15 to drive the vane oil pump. The required speed of the pump shaft 23 is n=710r/min, the Rotation speed of the drive shaft is 15n15=587r/min, and the transmission ratio $i_{15\text{-}23}$ = $n_{15/}$ $n_{23=}$ 1/1.21. Take the number of gear teeth on the pump shaft $z^{I}{}_{23}$ =24, Modulus m=2, pinion on drive shaft 15 $z^{I}{}_{15} = z^{I}{}_{23}$. $i23-15$=29.04, So take $z^{I}{}_{1}$ =29. The actual speed of the pump shaft is：

$$n_{23}=n_o \times \frac{z^{IV}{}_{o}}{Z^{IV}{}_{11}} \times \frac{z^{II}{}_{11}}{z^{II}{}_{12}} \times \frac{z^{II}{}_{12}}{z^{II}{}_{13}} \times \frac{z^{IV}{}_{13}}{z^{IV}{}_{14}} \times \frac{z^{IV}{}_{14}}{z^{IV}{}_{15}} \times \frac{z^{I}{}_{15}}{z^{I}{}_{23}} = 709\text{r}$$

4.3.3. Setting of handle shaft position

The three-sided drilling combination machine tool has 10 tools on the left spindle box. To facilitate the replacement and adjustment of tools or to check the accuracy of the spindle during assembly and maintenance, a handle shaft is generally required to rotate the spindle manually. To be easy to pull, the rotation speed of the handle shaft should be as high as possible. In addition, it must be noted that there should be a large space around the handle shaft so that the rotation angle of the handle shaft is not less than 60 degrees when it is pulled once. Therefore, we chose transmission shaft 11 as the handle shaft. After the gear parameters of the spindle box drive system are determined, recalculate the actual speed of all spindles as follows: The actual speed of spindles 1, 2, 3, and 4 is:

$$n_1 = n_2 = n_3 = n_4 = n_0 \times \frac{z^{IV}{}_{0}}{z^{IV}{}_{11}} \times \frac{z^{II}{}_{11}}{z^{II}{}_{22}} \times \frac{z^{II}{}_{22}}{z^{II}{}_{21}} \times \frac{z^{II}{}_{21}}{z^{II}{}_{18}} \times \frac{z^{IV}{}_{20}}{z^{IV}{}_{18}} \times \frac{z^{IV}{}_{18}}{z^{IV}{}_{19}} \times \frac{z^{I}{}_{19}}{z^{I}{}_{1}} = 710\ r/min$$

The actual speed of spindles 5 and 6 is:

$$n_5 = n_6 = n_0 \times \frac{z^{IV}{}_{0}}{z^{IV}{}_{11}} \times \frac{z^{II}{}_{11}}{z^{II}{}_{12}} \times \frac{z^{II}{}_{12}}{z^{II}{}_{13}} \times \frac{z^{IV}{}_{13}}{z^{II}{}_{14}} \times \frac{z^{II}{}_{14}}{z^{IV}{}_{5}} \times = 710\ r/min$$

The actual speed of spindle 7 is:

$$n_7 = n_0 \times \frac{z^{IV}{}_{0}}{z^{IV}{}_{11}} \times \frac{z^{II}{}_{11}}{z^{II}{}_{12}} \times \frac{z^{II}{}_{12}}{z^{II}{}_{13}} \times \frac{z^{IV}{}_{13}}{z^{II}{}_{14}} \times \frac{z^{IV}{}_{14}}{z^{IV}{}_{15}} \times \frac{z^{IV}{}_{15}}{z^{IV}{}_{16}} \times \frac{z^{III}{}_{16}}{z^{III}{}_{7}} = 403\ r/min$$

The actual speed of spindle 8 is：

$$n_7 = n_0 \times \frac{z^{IV}{}_{0}}{z^{IV}{}_{11}} \times \frac{z^{II}{}_{11}}{z^{II}{}_{22}} \times \frac{z^{II}{}_{22}}{z^{II}{}_{21}} \times \frac{z^{II}{}_{21}}{z^{II}{}_{20}} \times \frac{z^{IV}{}_{20}}{z^{IV}{}_{18}} \times \frac{z^{IV}{}_{18}}{z^{IV}{}_{17}} \times \frac{z^{I}{}_{17}}{z^{I}{}_{8}} = 403\ r/min$$

The actual speed of spindles 9 and 10 is:

$$n_9 = n_{10} = n_0 \times \frac{z^{IV}{}_{0}}{z^{IV}{}_{11}} \times \frac{z^{II}{}_{11}}{z^{II}{}_{22}} \times \frac{z^{I}{}_{22}}{z^{I}{}_{9}} = 403\ r/min$$

The calculated actual speed of each main shaft is shown in Table 3 (basically consistent with the requirements of the original basis diagram, the relative speed loss is within 5%, which meets the design requirements); the actual speed of the lubricating oil pump n_{23}709r/min, also meets the requirements. Generally, the cutting parameters in the machining diagram should be corrected according to the actual speed of the spindle. Since the actual speed is very close to the required speed (the error is less than 1%), there is no need to make corrections.

Table 3: The actual speed of each spindle

Spindle number	Spindle 1	Spindle 2	Spindle 3	Spindle 4	Spindle 5
Actual speed n(r/min)	710	710	710	710	710
Spindle number	Spindle 6	Spindle 7	Spindle 8	Spindle 9	Spindle 10
Actual speed n(r/min)	710	403	403	403	403

Finally, mark the number of teeth, modulus, and row number of all gears designed by the transmission system in the transmission system diagram according to the specified format (Figure 8).

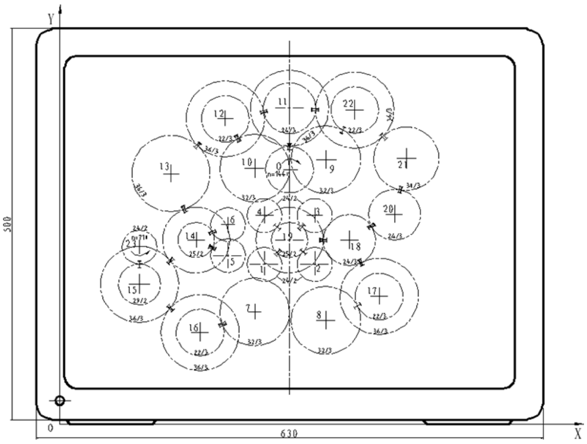

Figure 8: The left headstock transmission system

4.3.4. Calculate drive shaft diameter

When determining the shaft diameter of each drive shaft in the headstock transmission system, the load on the shaft must first be calculated. The drive shaft is regarded as being subjected to a pure torque, regardless of the influence of the bending moment, and lower materials are used. A strength calculation method for allowable stress. Therefore, to determine the shaft diameter of a certain drive shaft, we only need to calculate the torque transmitted by it and then check the "torque that the shaft can bear" table ("Combined Machine Tool Design Concise Manual") based on this torque to determine the shaft diameter. The torque of the drive shaft is calculated as follows：

$$M=M_1 i_1 + M_2 i_2 + \cdots + M_n i_n \tag{4.2}$$

Where： M— Total torque acting on the shaft, the unit is N m

M_n— Torque on the nth axis， N m

i_n— Gear ratio of the nth pair of shafts。 axis 19

$$M_{19} = 4M_1 i = 7.75N.m$$

Press "Torque that the shaft can bear" table Available shaft diameter d=20mm Yes, but to increase the rigidity of the shaft and reduce the types of transmission parts, Take shaft diameter d=25mm（Same below).

axis 17

$$M_{17} = M_8 i = 7.38N.m$$

press "Torque that the shaft can bear" table Available shaft diameter d=25mm axis 18

$$M_{18} = M_{17} I_{17} + M_{19} i_{19} = 10.92N.m$$

press "Torque that the shaft can bear" table Available shaft diameter d=25mm axis 20

$$M20 = M18i\text{=10.92 N. m}$$

press "Torque that the shaft can bear" table，Available shaft diameter d=25mm axis 21

$$M_{21} = M_{20} i = 13.65N.m$$

press "Torque that the shaft can bear" table， Available shaft diameter d=25mm axis 22

$$M_{22} = M_9 i_9 + M_{21} + i_{21} = 23.76N.m$$

press "Torque that the shaft can bear" table， Available shaft diameter d=25mm axis 16

$$M_{16} = M_7 i = 7.38N.m$$

press "Torque that the shaft can bear" table， Available shaft diameter d=25mm axis 23 Is the shaft on the vane oil pump and its rated power P =0.1kW, Rotating speed n=710r/min, Then the torque on it can be calculated as follows:

$$M_{23} = \frac{30p}{\pi n} = 1.35\text{N.m}$$

axis 15

$$M_{15} = M_{16}i_{16} + M_{23}i_{23} = 9.18N.m$$

press “Torque that the shaft can bear” table, Available shaft diameter d=25mm axis 14

$$M_{14} = 2M_5i_5 + M_{15}i_{15} = 11.78N.m$$

press “Torque that the shaft can bear” table, Available shaft diameter d=25mm axis 13

$$M_{13} = M_{14}i = 13.68N.m$$

press “Torque that the shaft can bear” table, Available shaft diameter d=25mm axis 12

$$M_{12} = M_{13}i_{13} + M_{10} + i_{10} = 21.06N.m$$

press “Torque that the shaft can bear” table, Available shaft diameter d=25mm axis 11

$$M_{11} = M_{12}i_{12} + M_{22}i_{22} = 29.5N.m$$

Press the “Torque that the shaft can bear” table, Available shaft diameter d=30mm.

4.4. Check the strength of parts

After the transmission system is drawn up, the selected transmission shaft journal and gear modulus should be checked in the transmission design to check whether it meets the working requirements.

4.4.1. Check the diameter of main shaft and drive shaft

All the main shafts and transmission shafts in the headstock are subject to very small bending moments, so the effect of bending moments can be disregarded and checked according to pure torsion theory. Since the shaft diameter is calculated and selected according to the torque received by the transmission bearing, the shaft diameter will not be checked here.

4.4.2. Check of gear strength

Gear strength checks: we only need to check some gears' contact fatigue strength and bending fatigue strength with the largest and weakest loads in the headstock.

4.4.2.1. Selection of gear material, accuracy grade and heat treatment

The main shaft gear of the three-sided drilling combination machine tool includes transmission gears, motor gears, and vane lubricating oil pump shaft gears. The materials are all made of No. 45 steel. The gear accuracy grades are selected from 8-7-7. The teeth are subjected to high-frequency quenching G54. Gear strength checks only need to check the contact fatigue strength and bending fatigue strength of some gears with the largest and weakest load in the headstock.

4.4.2.2. Check method of tooth surface bending strength Tooth surface bending fatigue strength is checked by pressing

$$\sigma_F = \frac{2KT1}{bm^2z1} \times Y_{Fa}\ Y_{Sa}\ Y_6 \leq [\sigma] \tag{4.3}$$

Where： K—Load factor, Can be calculated as follows，KA For use factor，$K\beta$ Is the tooth load distribution factor， Is the load distribution coefficient between teeth $K\alpha$，Kv Is the dynamic load factor；

$$K=K_AK_VK_\beta K_\alpha \tag{4.4}$$

$T1$— Driving gear transmits torque;
b—— Gear width；
m—— Gear modulus
$Z1$—— Number of teeth of pinion；

YFa—— Tooth form factor;
YSa—— Tooth root stress correction factor;
Y $_\delta$ = 0.25+$\frac{0.75}{S\alpha}$, among them; ε_α = 1.88 – 3.2 ($1/z_1 + 1/z_2$)
$[\sigma]F$—— Allowable bending stress, calculated as follows.

$$[\sigma]_F = \frac{\sigma_{Flim} \cdot Y_N}{S_{Fmin}} \tag{4.5}$$

Where: YN— Bending fatigue strength life factor. Check the value in the figure. The abscissa in the figure is the number of cycles of working stress of the gear wheel N， N. Calculate as follows;

$$N=60njL_h \tag{4.6}$$

Where：n—— Gear shaft speed, r/min;
j—— The number of meshing times of the same tooth surface per revolution of the gear;
Lh—— Gear working life,h.
$\sigma F\ lim$— Test gear bending fatigue strength limit;
$SF\ min$— The minimum safety factor for bending fatigue strength calculation for general gears and most industrial gears, according to general reliability requirements, take $SF\ min$ = 1.2.

4.4.2.3. Check method of tooth surface contact strength

Table 4: Method of tooth surface contact strength

Gear pair number	K_A	K_V	K	K	T_1 / N m	b /mm	F lim1 /MPa
19—I—1	1.25	1.1	1.13	1.2	1938	24	530
11—II—22	1.25	1.17	1.08	1.2	15480	24	530
22—II—21	1.25	1.20	1.08	1.2	10920	24	530
Gear pair number	**m**	**z_1**	**Y_{Fa1}**	**Y_{Sa1}**	**Y_{Fa2}**	**Y_{Sa2}**	**Y**
19—I—1	2	24	2.63	1.59	2.66	1.57	0.71
11—II—22	3	24	2.66	1.57	2.44	1.62	0.70
22—II—21	3	30	2.51	1.63	2.44	1.62	0.70
Gear pair number	**Y_{N1}**	**Y_{N2}**	**$S_{F\ min}$**	**Z_E**	**Z**	**Z_H**	**F lim 2 /MPa**
19—I—1	0.94	0.93	1.25	189.8	0.90	2.5	510
11—II—22	0.92	0.94	1.25	189.8	0.88	2.5	510
22—II—21	0.94	0.94	1.25	189.8	0.88	2.5	510
Gear pair number	**u**	**Z_{N1}**	**Z_{N2}**	**H lim1**	**H lim 2**	**$S_{H\ lim}$**	
19—I—1	25/24	0.96	0.97	1150	1150	1.0	
11—II—22	36/24	0.95	0.97	1150	1150	1.0	
22—II—21	36/24	0.97	0.97	1150	1150	1.0	

$$\sigma_H = Z_E Z_\delta Z_H \sqrt{\frac{2KT_1}{bd_1^2} \cdot \frac{u+1}{u}} \le [\sigma]_H \tag{4-8}$$

Where: the formula: ZE— Material elasticity coefficient，$\sqrt{MPa}$;
Z_δ— Coefficient of contact strength coincidence, according to equivalent gear coincidence $\varepsilon\alpha$ Check by chart;
Z_H— Node area coefficient;
K— Load factor， By Type（4.5）Calculation;
T_1— Driving gear transmits torque;
d_1— Index circle diameter of pinion;
u— Gear ratio,
$[\sigma]_H$— Allowable contact stress，$[\sigma]_H$ Calculated by the following formula

$$[\sigma_H] = \frac{\sigma_{H\,lim}}{S_{H\,lim}} Z_N \quad (4.8)$$

Where: $\sigma_{H\ lim}$— Test gear contact fatigue strength limit, the unit is MPa ; $S_{H\ lim}$— Minimum safety factor for calculation of contact fatigue strength, for general gears and most industrial gears, according to general reliability requirements, take $S_{H\ lim} = 1$; Z_N— Life factor of contact fatigue strength, checked by chart, the abscissa in the figure is the number of cycles of gear working stress N, N Calculated by (4.7) (Table 4).

4.4.2.4. Checking the bending strength and contact strength of the gear tooth surface

Select the relatively loaded headstock, Check the weaker pairs of gears, and Check the "Machine Design Manual" and related manuals to determine that the parameter values of the gears used in each pair of calibrations are listed in Table 5.

Table 5: Various parameter values for gear strength check

Gear pair number	**KA**	**Kv**	**K**	**K**	**T1/ N m**	**b /mm**	**F lim1 /MPa**
19—I—1	1.25	1.1	1.13	1.2	1938	24	530
11—II—22	1.25	1.17	1.08	1.2	15480	24	530
22—II—21	1.25	1.20	1.08	1.2	10920	24	530
Gear pair number	**m**	**z1**	**YFa1**	**YSa1**	**YFa 2**	**YSa 2**	**Y**
19—I—1	2	24	2.63	1.59	2.66	1.57	**0.71**
11—II—22	3	24	2.66	1.57	2.44	1.62	**0.70**
22—II—21	3	30	2.51	1.63	2.44	1.62	**0.70**
Gear pair number	**YN1**	**YN 2**	**SF min**	**ZE**	**Z**	**ZH**	**F lim 2/MPa**
19—I—1	0.94	0.93	1.25	189.8	0.90	2.5	510
11—II—22	0.92	0.94	1.25	189.8	0.88	2.5	510
22—II—21	0.94	0.94	1.25	189.8	0.88	2.5	510
Gear pair number	**u**	**ZN1**	**ZN 2**	**H lim1**	**H lim 2**	**SH Lim**	
19—I—1	25/24	0.96	0.97	1150	1150	1.0	
11—II—22	36/24	0.95	0.97	1150	1150	1.0	
22—II—21	36/24	0.97	0.97	1150	1150	1.0	

Note: The numbers on both sides of the gear pair number in the table indicate the axis where the gear is located, and the number in the middle indicates the row number of the gear.

Substitute the values in the above table into formulas (4-4), (4-5), (4-6), (4-7), (4-8), and (4-9) to calculate and get each pair Bending stress of gear pair, contact stress and its allowable value are shown in Table 6.

Table 6: Gear strength verification results

Gear pair	**gear**	$_F$ **/MPa**	***F* /MPa**	$_H$ **/MPa**	***H* /MPa**
19—I—1	z^{I}_{19}	8.8	383.5	207.5	1104
	z^{I}_{1}	8.9	379.4	207.5	1115.5
11—II—22	z^{II}_{11}	50.0	262.8	453.4	1092.5
	z^{II}_{22}	47.3	383.5	453.4	1115.5
22—II—21	z^{II}_{22}	18.3	383.5	258.6	1115.5

	z^{II}_{21}	18.9	383.5	258.6	1115.5

It can be seen from the data in Table 6 that of all gear pairs $\sigma F < [\sigma]F$， $\sigma H < [\sigma]H$, this proves that the bending fatigue strength and contact fatigue strength of all gears meet the requirements of use.

This can be divided into four parts： First, determine the position coordinates of all spindles and drive shafts. Draw up the scheme of the spindle box transmission system scheme can be drawn up choose the best transmission scheme; second, Determine the specific position of the drive shaft and the number of teeth of the gear, modulus, and number of rows; again，Determine the shaft diameter of all drive shafts according to the torque transmitted by the drive shaft； Finally, check the strength of important parts, ensure to meet safety requirements.

5. Headstock coordinate calculation and drawing of interference check diagram

The coordinate calculation of the spindle box accurately calculates the coordinates of each intermediate drive shaft according to the known position and transmission relationship of the drive shaft and the main shaft. Its purpose is to provide the coordinate size of the hole for the supplementary processing drawing of the headstock box parts and to draw the coordinate inspection diagram to check whether the gear arrangement and structural arrangement are correct and reasonable.

5.1. Select the machining reference coordinate system

To facilitate the processing of the headstock box the reference coordinate system must be selected during design. Usually use the Cartesian coordinate system XOY。According to the placement and processing conditions of the spindle box, it can be determined as follows: The headstock of the three-sided drilling modular machine tool is based on the bottom surface, installed directly on the power slide, movement through the sliding table, Drive the tool feed. Therefore, the coordinate system's horizontal axis (X-axis) is selected at the bottom, and the longitudinal axis (Y-axis) passes through the positioning pinhole so that the process reference and the design reference are consistent and it is easy to ensure the machining accuracy of the spindle. The selected machining reference coordinate system is shown in Figure 9.

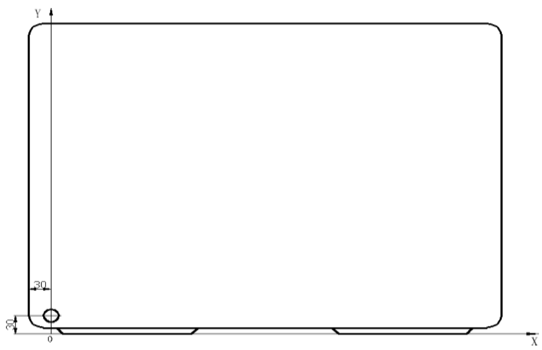

Figure 9: Calculate the coordinates of the spindle and drive shaft

According to the original design of the headstock, according to the selected reference coordinate system XOY, calculate the coordinates of each spindle and drive shaft. (The calculation accuracy must be three digits after the decimal point.) For the coordinate values of the spindle and drive shaft, see Table 2.

5.2. Calculate the coordinates of the drive shaft

When calculating the drive shaft coordinates, first calculate the drive shaft coordinates with direct transmission relationship and then calculate the other drive shaft coordinates. It drives the axis coordinates. There are many types of transmission shafts. This headstock has two major types: a transmission shaft with a fixed distance from one shaft and a transmission shaft with a fixed distance from two shafts. The specific calculation is as follows：

Axis 19

Because drive shaft 19 is located in the centre of the concentric circles where the main shafts 1, 2, 3, and 4 are located:

$$X_{19} = \frac{X_1+X_2}{2} = 285.000$$

$$Y_{19} = \frac{y_1+y_2}{2} = 215.000$$

Axis 18

The centre of shaft 18 is on the same horizontal line as the centre of shaft 19 on its right side, so

$$X_{18} = X_{19} + A_{18-19} = 367.500$$

$$y_{18} = y_{19} = 215.000$$

Axis 11

$$x_{11} = x_0 = 285.000$$
$$y_{11} = y_{24} + A_{0-11} = 402.000$$

Axis 20

The positions of axis 20 and axis 21 are not uniquely determined, so you might as well set:

$$y_{20} = y_{18} + \frac{A_{18-20}}{2} = 251.000$$

$$\text{Then, } x_{20} = x_{18} + \sqrt{(A^2_{18\text{-}20} - (\frac{A_{18-20}}{2})2)} = 429.854$$

Lubricating oil pump shaft 23

$$x_{23} = x_{14} - \sqrt{(A_{14-152} - A_{23-152})} = 70.500$$
$$y_{23} = y_{14} = 215.000$$

The transmission shafts 12, 13, 14, 15, 16, 17, 21, and 22 all have a fixed distance from the two shafts. The transmission shaft and the two shafts are fixed distances. Two pairs of gears are used to drive two Knowledge axes. The coordinates can be calculated based on the known two-axis coordinates and two pairs of gear centre distances. The calculation method [2] is shown in Figure 10, where and are the coordinates of the two known axes, R1 and R2. Is the gear centre distance between the two known shafts and the transmission shaft, that is, is the coordinate of the transmission shaft to be calculated:

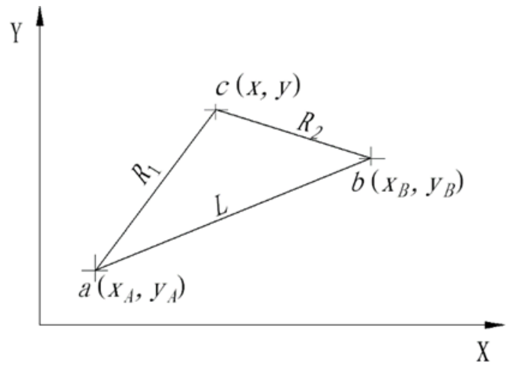

Figure 10: Two-axis fixed distance diagram

In Figure 11, the point is its origin. Make the coordinates (I, J) of the point in the small coordinate system positive, set it in counterclockwise order, make auxiliary lines, and label them as shown in the figure from which the point coordinate calculation formula can be derived. Which is:

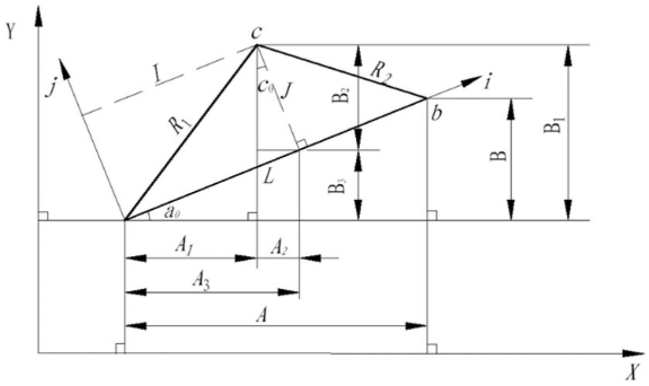

Figure 11: Coordinate calculation diagram with two-axle fixed-distance drive shaft

Assume, A=$x_B - x_A$, B=$y_B - y_A$

$$L=\sqrt{A^2+B^2}$$

$$I=\frac{1}{2L}(R_1^2+L^2-R_2^2)$$

$$J=\sqrt{R_1^2-I^2}$$

Because $\sin c_0 = \sin a_0 = \frac{B}{L}$, $\cos c_0 = \cos a_0 = \frac{A}{L}$

So $A1 = A3 - A2 = I\cos a_0 - J\sin c_0 = \frac{AI-BJ}{L}$

$B1 = B3 + B2 = I\sin a_0 + J\cos c_0 = \frac{BI+AJ}{L}$

Restore to the XOY coordinate system. The required coordinate of point c is:

$$X = x_A + A_1 = x_A + \frac{AI-BJ}{L} \quad (5.1)$$

$$y = y_A + B_1 = Y_A + \frac{BI+AJ}{L} \quad (5.2)$$

Due to the many transmission shafts, computer programming calculates the transmission shaft coordinates with a fixed distance from the two shafts. The calculation program flowchart is shown in Figure 12.

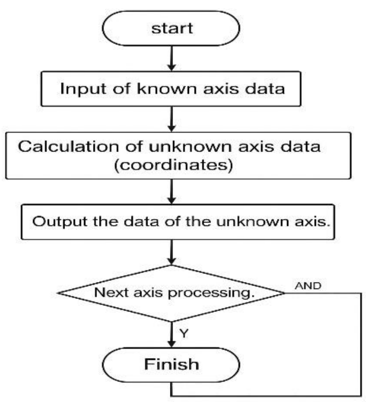

Figure 12: Flowchart of calculation of transmission shaft coordinates

The program compiled by the above algorithm and flowchart is as follows：

Function f=coordinate computation (X, Y)

XA=input ('please input the X coordinate of the first axis\nXA='); %Enter the first axis X coordinate value

YA=input ('please input the Y coordinate of the first axis\nYA='); %Enter the Y coordinate value of the second axis

```
XB=input ('please input the X coordinate of the other axis\nXB='); %Enter the X coordinate value of the second axis
YB=input ('please input the Y coordinate of the other axis\nYB=) %Enter the Y coordinate value of the second axis;
A=XB-XA:
B=YB-YA:

R1=input ('please input the distance between the unknown axis and the first axis\n'); %Enter the centre distance between the unknown axis and the first axis R2=input ('please input the distance between the unknown axis and the other axis\n'); %Enter the centre distance between the unknown axis and the second:

axis L=sqrt(A^2+B^2);
I=(R1^2+L^2-R2^2)/(2*L);
J=sqrt(R1^2-I^2); a0=asin(B/L); c0=a0;
A1=I*cos(a0)-J*sin(c0); B1=I*sin(a0) +J*cos(c0);
K=input ('please input K\n'); % Set a parameter to determine the position of the unknown axis relative to the two known axes
if K>0 % The first case, such as axis 14，12，22，21，13
    X=XA+A1;
    Y=YA+B1;
Else
if K<0 %The second case, such as 17 axis b=acos((L^2+R2^2-R1^2)/(2*L*R2)); b1=atan(A/B);
    b2=b-b1;
    A4=L*sin(b1);
    A5=R2*sin(b2);
    B4=B;
    B5=R2*cos(b2); A1=A4+A5; B1=B4-B5; X=XA+A1;
    Y=YA+B1;
Else %The third case, such as 16 axis a2=a tan(-B/A);
      a1=a cos((R1^2+L^2-R2^2)/(2*R1*L)); a=a1+a2;
      A1=R1*cos(a);
      B1=R1*sin(a);
       X=XA+A1;
```

At the end of the program, the coordinates of all drive shafts are obtained, and the coordinate values of all main shafts and drive shafts are listed in a table, as shown in Table 7.

Table 7: Coordinates of the spindle and drive shaft of the headstock

Coordinate	**Drive shaft O**	**Pump shaft 23**	**Spindle 1**	**Spindle 2**	**Spindle 3**	**Spindle 4**
X	285.000	70.800	250.352	319.648	319.648	250.352
Y	315.000	215.000	180.352	185.352	249.648	249.648
Coordinate	**Spindle 5**	**Spindle 6**	**Spindle 7**	**Spindle 8**	**Spindle 9**	**Spindle10**
X	199.973	199.973	237.000	335.000	335.000	237.000
Y	193.000	237.000	113.000	101.000	329.000	317.000
Coordinate	**Transmission Shaft 11**	**Transmission Shaft 12**	**Transmission Shaft 13**	**Transmission Shaft 14**	**Transmission Shaft 15**	**Transmission Shaft 16**
X	285.000	196.258	121.305	156.189	70.800	157.482
Y	402.000	387.008	309.252	215.000	162.000	97.577
Coordinate	**Transmission**	**Transmission**	**Transmission Shaft**	**Transmission Shaft**	**Transmission Shaft**	**Transmission Shaft**
X	408.559	367.500	285.000	429.854	445.899	374.964
Y	134.912	215.000	215.000	251.000	330.395	399.455

Note: 1. The X axis of the coordinate system XOY is selected on the bottom of the box, and the Y axis passes through the positioning pin and the coordinates are:

x_0 = 0.000, y_0 = 30.000, Positioning pin O_2 the coordinates are：

x_0 = 570.000, y_0 = 30.000

2. The calculation precision of the coordinate () value is accurate to three decimal places (the third digit is rounded off).

5.3. Check center distance error

The whole system on the spindle box is processed according to the calculated coordinates. The assembly requires that the gears between the two shafts mesh normally. Therefore, it is necessary to check whether the actual centre distance A, determined according to the coordinate calculation, meets the standard centre distance R required for gear meshing between the two shafts. The difference between R and A is:

Checking standard: allowable error of center distance(δ) ≤ (0.001~0.009) mm. The calculation formula for the drive shaft is as follows：

$$\delta_{i-j} = R_{i-j} - A_{i-j} = \frac{m}{2}(z_i + z_j) - \sqrt{(x_1 - x_j)^2 + (y_i - y_j)^2} \quad (5\text{-}3)$$

Where： $\delta((i-j)$ — The center distance error between the root axis and the root axis.

$Ri-j$— The standard center distance between the root axis and the root axis.

$Ai-j$— the actual centre between the root axis and the root axis distance.

m—Modulus of a pair of meshing gears on the i-th and j-th shaft；

z_I— Number of gear teeth on the i-th shaft；

z_j— Number of gear teeth on the j-th shaft；

x_i, y_i— The actual coordinate value of the i-th axis；

$y_i y_j$— The actual coordinate value of the j-th axis;

Due to many axes, the program calculates the center distance error to simplify the calculation and reduce the workload. The procedure is as follows: function f=error analysis(W):

XA=input ('please input the X coordinate of the first axis\nXA=')
YA=input ('please input the Y coordinate of the first axis\nYA=') XB=input ('please input the X coordinate of the other axis\nXB=) YB=input ('please input the Y coordinate of the other axis\nYB=')

A=XB-XA;
B=YB-YA;
R=input ('please input the distance between two axis\n');
L=sqrt(A^2+B^2); %Calculate the actual center distance of the two axes W=R-L; %Calculate the center distance error of the two axes fprintf ('the error of the distance between the two axis W=%f\n,' W) (% Output two-axis center distance error.

5.4. Draw coordinate check map

After the coordinate calculation is completed, the coordinate and transmission relationship inspection diagram should be drawn to check the correctness of the transmission system comprehensively.

5.4.1. The main content of the coordinate check map

- Check whether the coordinate position is correct through gear meshing; check the spindle speed and steering.
- Further, check whether each part has interference.
- Check whether the position of additional mechanisms, such as hydraulic pumps and oil separators, is appropriate.

5.4.2. drawing of coordinate check map

After the coordinates are determined, draw the coordinate check chart in the following order;

- Draw the outline size and coordinate system of the spindle box XOY.
- Draw the position of each spindle, drive shaft axis, and the diameter of each spindle extension according to the calculated coordinate values, and indicate the axis number and the rotation speed and steering of the spindle, drive shaft, hydraulic pump shaft, etc.

- Draw the index circle of each gear with a dotted line, and indicate the number of teeth, modulus, and row number of each gear. The final coordinate check drawing is shown in Figure 13.

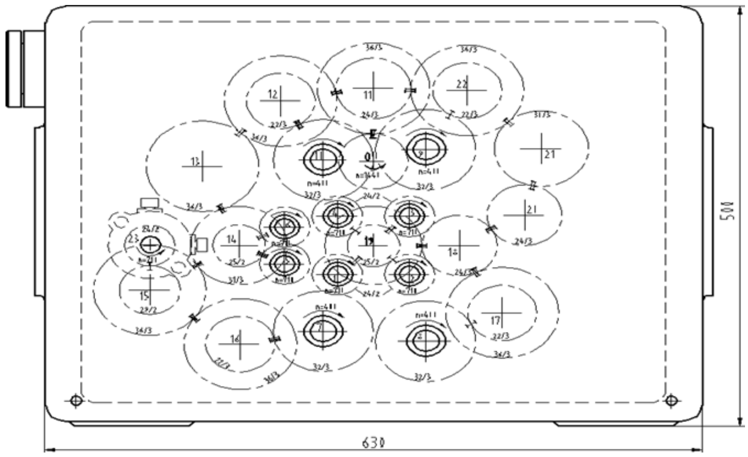

Figure 13: Checking the coordinates of the spindle box

5.5. Draw headstock general drawing and parts drawing

5.5.1. Headstock general drawing design

The general headstock drawing design includes four parts: drawing the main view, unfolding the drawing, preparing the assembly table, and formulating technical conditions.

- The main view mainly shows the position of the main shaft of the headstock and the gear transmission system, the number of gear teeth, the module number and the number of rows, the lubrication system, etc. Therefore, drawing the main view is to mark each axis number on the designed transmission system diagram, draw the lubrication system, mark the speed of the main shaft, oil pump shaft, drive shaft, oil pump shaft steering, drive shaft steering, and coordinate dimensions, minimum main shaft height dimension, etc. and mark the part number.
- Expanded diagram Its characteristic is that the shaft has many structural graphics. Most of the main shafts and transmission shafts and the parts on the shafts are universal and arranged regularly. Generally, a simplified expansion drawing is used and matched with an assembly table to indicate the assembly structure of each axis assembly of the multi-axis box.
- The expanded view mainly shows the assembly relationship of each axis and the parts on the axis, including the main shaft, transmission shaft, drive shaft, handle shaft, oil pump shaft, and the corresponding gears, spacer sleeves, oil sleeves, bearings or oil pumps, and other mechanical parts, as well as the relative positions of installation. The axial and radial dimensions of each part in the figure are drawn to scale. The axial distance and the expansion sequence can be drawn not according to the transmission relationship, but the number of gear rows, shaft number, and diameter specifications must be indicated. For short-distance shafts, a complete set of related shafts must be drawn according to the actual spacing to check whether there is visual interference.
- Only one can be drawn for the same type of main shaft and drive shaft with the same structure, and the same shaft's axis number can be indicated on the shaft end. The axial assembly structure is the same, but two or two sets of shafts with different gear sizes and positions can be drawn together. Each side of the axis line represents one or a set of shafts.
- The three main box thickness dimensions of the main spindle box should be fully marked on the expanded drawing: the related dimensions of the box wall and the inner cavity, the extension of the spindle, etc.
- The general drawing should also have a partial cross-sectional view showing the positioning structure between the power sliding table and the box, as well as the front cover and the box.
- The main shaft and drive shaft assembly table sets the model specifications, size parameters, and quantities of basic parts such as gears and sleeves on each shaft (main shaft, drive shaft, oil pump shaft) in the main shaft box, as well as standard parts, purchased parts, etc., as required, Expressed by the assembly table. This makes the chart comparison clear and easy to see, saves design time, and facilitates assembly.
- Technical conditions of the spindle box and assembly requirements should be indicated on the general drawing of the spindle box. Technical conditions for manufacturing and acceptance of the spindle box: ZBJ58011-89《Technical

conditions for manufacturing multi-axis box of modular machine tools 》To manufacture, press ZBJ58012-89《Technical conditions for acceptance of multi-axis box of modular machine tools》Carry out acceptance.

- Spindle accuracy: press JB3043-82《Multi-axis box accuracy of modular machine tools 》 standard for acceptance.

5.5.2. Headstock parts design

In the design of the headstock general drawing, most parts use general-purpose parts, standard parts, and purchased parts; for parts such as displacement gears (displacement gears are not involved in the design of the headstock), special gears and other parts, parts drawings should be designed; For general-purpose parts such as spindle box, supplementary processing drawings must be drawn. Working drawings of special parts, such as pump shaft gears, etc., can be specified according to the working drawings of general parts. Refer to the drawings of similar common parts and designs in combination with special requirements. Supplementary machining diagrams of general parts, such as the main shaft box body, front cover, rear cover, etc., should be drawn according to the requirements of the main head box general drawing. For the parts that need to be supplemented (such as multi-axis box main shaft bearing hole, drive shaft bearing hole, oil pump and its shaft Holes, positioning pin holes, etc.), it is usually customary to draw the structure of the supplementary processing part with a thick solid line, and its size, form, and position tolerance, surface roughness, etc. are marked following the international standard format of mechanical drawing; the original outline of general castings, etc. A thin, solid line represents it. This calculates the shaft position in the spindle box drive system in detail. First, a coordinate system is established, and the coordinates of all drive shafts are calculated. Then, check the centre distance error of the gear pair to determine whether the gear needs to be displaced, and draw the coordinate inspection diagram of the headstock to check whether all parts inside the headstock interfere. Finally, design and draw the general drawing of the headstock and the parts drawing to complete the two-dimensional design of the entire headstock.

6. Conclusion

This article presents a comprehensive design calculation for the left headstock of the three-sided drilling modular machine tool used for drilling operations at a station on the S1001 transmission front housing production line. The paper includes creating two-dimensional engineering drawings and a three-dimensional headstock model, successfully completing the design task, and achieving the intended objectives. The implementation of the three-sided drilling combination machine tool significantly reduces drilling time and enhances the overall production efficiency of the transmission front housing production line. The design process spanned over three months, culminating in a successful completion. Unlike previous curriculum-based papers, this design task demanded higher technical knowledge and encompassed a broader range of subjects. It critically assessed accumulated academic knowledge and facilitated a deeper understanding of mechanical engineering principles. The graduation paper focused on applying comprehensive knowledge and skills to address practical engineering problems, aiming to strengthen analytical and problem-solving capabilities. Throughout the project, foundational theories and specialized mechanical design knowledge were effectively applied independently. The design work enhanced technical proficiency and expanded familiarity with areas of study that had previously received limited attention. This paper demonstrates a significant step in integrating theoretical understanding with practical engineering design.

Acknowledgment: The graduation project has been done successfully under the tutor's guidance. Review the experience and design flaws. The design has many issues. The beginning of everything is hard. The structure is designed without precise information at the start. When visiting FAW Gear Factory, just stocks are visible. The internal structure of the headstock, including shaft distribution, gear arrangement, and lubrication, is difficult to observe. Constant information assessment and consultation fixed it. Time constraints make this graduation design feel weak. Kinematics and dynamics modeling and analysis of the assembled headstock three-dimensional entity are not possible, allowing some time for this design.

Data Availability Statement: The data for this study can be made available upon request to the corresponding author.

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Conflicts of Interest Statement: The authors have no conflicts of interest to declare.

Ethics and Consent Statement: Analyzed and summarized relevant materials using standards, tools, books, network resources, and computers for design computation and analysis. My professional level and capacity to apply knowledge have improved after three months of study and research. Determine the component positions and sizes based on the sketch, layout, and analysis calculations in the sketch design stage. During detailed computation and design, all parts' dimensions are considered for matching, interference, and strength. Thorough consideration, continual optimization and improvement, and headstock design completion.

References

1. X. Jiaying. “A Concise Manual for Modular Machine Tool Design,” Mechanical Industry Press, Beijing, China, 1999.
2. T. Qingchang and Z. Hongzhi, “Mechanical Design,” Beijing, China, 2004.
3. Q. Rongrong and C. Kewei, “Principles of Machinery.” Higher Education Press, Beijing, China, 2006.
4. L. Hou Hongsheng, “Mechanical Engineering Graphics,” Mechanical Engineering, Beijing, China, 2005.
5. C. Daxian, Mechanical Design Manual, Chemical Industry Press, Beijing, China, 2008.
6. W. Kou Zunquan, Curriculum Design of Mechanical Design. Mechanical Industry Press, Beijing, China, 2006.
7. G. Xiuhua and W. Zhiming, “Engineering analysis and electronic prototype simulation,” Mechanical Industry Press, Beijing, China, 2004.
8. M. Nie Yuqin, “Mechanics of Materials,” Mechanical Industry Press, Beijing, China, 2005.
9. H. Hongsheng. Practical Course of Computer Graphics, Science Press, Beijing, China, 2005.
10. G. Yongli and C. Xiaohua, The basis of mechanical precision design. Jilin People’s Publishing House, Changchun, Jilin Province, China, 2001.
11. W. Xiankui. Mechanical processing technology manual (separate edition) drilling, reaming, and reaming processing, Machinery Industry Press, Beijing, China, 2009.
12. Y. Junyi and Z. Qing, Fundamentals of Machinery Manufacturing Technology, Machinery Industry Press, Beijing, China, 2008.
13. B. M. Лобусев and Z. Huanyao, “YHE-3100 unified series headstock,” in Modular Machine Tool and Automatic Processing Technology, China, 1980.
14. X. Shijun, “Some experience in the design of standard headstock for drilling and attacking,” in Modular Machine Tool and Automatic Processing Technology, China, 1982.
15. X. Feng, “Reasonable use of spindle driven spindle in the design of modular machine tool headstock transmission,” Mechanical Industry Press, 1995.
16. L. Liang Taowen, “Application of CAD Technology in the Design of Headstock,” in Modular Machine Tool and Automatic Processing Technology, Mechanical Industry Press, 1992.
17. Z. Yahui, “Design of Drilling and Tapping Compound Headstock,” in Modular Machine Tool and Automatic Processing Technology, China, 2001.
18. X. Xizeng and F. Zijun, “Design and calculation of modular machine tool headstock,” Mechanical design, China, 1994.
19. D. Guoqin, “Computer Aided Design of Modular Machine Tool Headstock,” Machinist, China, 2002.
20. “www.mt35.com-official homepage,” Mt35.com. [Online]. Available: http://www.mt35.com. [Accessed: 04-Oct-2023].