

Design and Development of Sleeve inserting Machine

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Abstract

Modern cars have brakes on all four wheels, operated by a hydraulic system. The brakes may be disc type or drum type. The front brakes play a greater part in stopping the car than the rear ones, because braking throws the car weight forward on to the front wheels. Braking system is an integral part of the four wheeler vehicles. Brake tubes are located in a high-corrosion area and although many other automotive components operate in the same hostile environment, few are less forgiving in the event of a failure. Thus, one of the major considerations in the design of an automotive hydraulic brake system is the integrity of the brake tubing which distributes the system pressure. The use of an inherently corrosion resistant material is the best protection against long-term brake tube corrosion. Therefore, sleeves are used to protect the brake tube from corrosion. Hence, we are designing typical sleeve inserting machine that will make the process of insertion of the sleeve upon the brake rod through which high pressure fluid flows.

Keywords: Sleeve Inserting Machine, Microcontroller, Roller, PLC, Sensors.

1. Introduction

A brake is a mechanical device which inhibits motion. Hydraulics is the use of a liquid under pressure to transfer force or motion, or to increase an applied force. The pressure on a liquid is called Hydraulic Pressure and the brakes which are operated by means of hydraulic pressure are called Hydraulic Brakes.

When we press the brake pedal, it pushes on primary piston through a linkage. Pressure is built in the cylinder and the brake tubes as the brake pedal is depressed further. The pressure between the primary and secondary piston forces the secondary piston to compress the fluid in its circuit. If the brakes are operating properly, the pressure will be same in both the circuits. So, brake tubes are mainly used for transmitting fluid force to the wheels of the vehicle in order to stop the vehicle according to desirable motion.

Hence, the use of an inherently corrosion resistant material is the best protection against long-term brake tube corrosion. Therefore, sleeves are used for protection of brake tubes against corrosion. The brake tubes that are covered with sleeves are shown below:

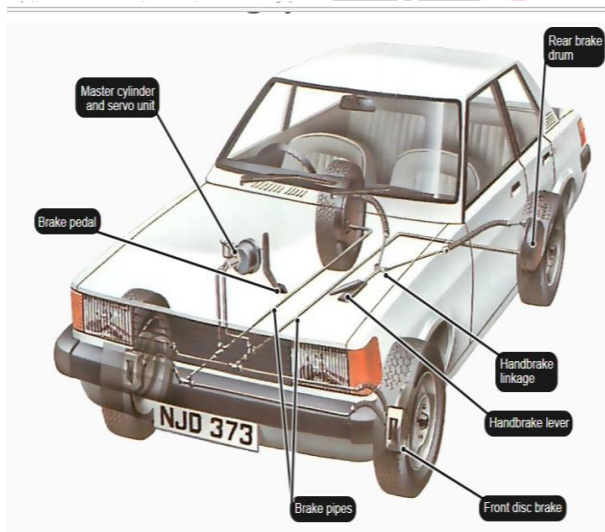


Fig.1

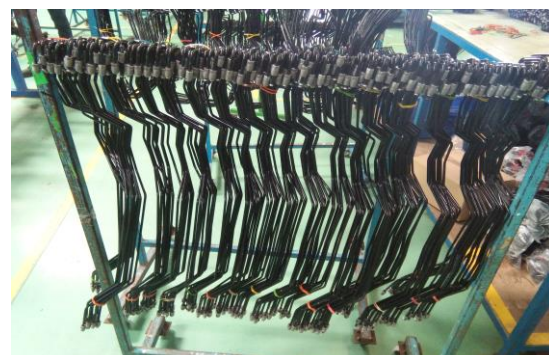
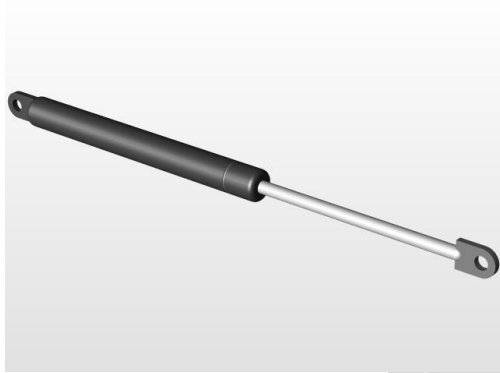


Fig.2

These brakes are based on the principle of Pascal's law.

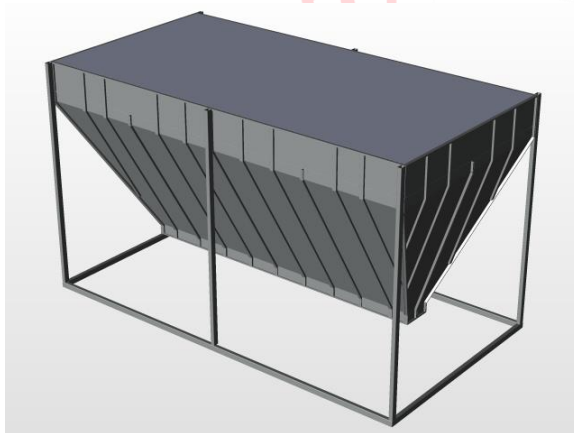
Currently these sleeves are fitted over the rods manually in various industries. So, we are designing automatic system for covering the brake tubes by sleeves. It will save time and cost ultimately.

Pneumatic Cylinder:



Function of the cylinder is to push the sleeve when sleeve falls inside the hopper. The stroke length and base diameter of pneumatic cylinder is 50 mm and 16mm.

Hopper:



Function of the cylinder is to push the sleeve into brake tube. Brake tube is placed in front of the sleeve to be inserted such that the axis of brake tube and sleeve is properly aligned. When piston from pneumatic cylinder gives the forward stroke to the sleeve, the sleeve will move forward and will cover the brake tube until the piston from pneumatic cylinder touches the brake tube. The stroke length and base diameter of pneumatic cylinder is supposed to be 50 mm and 16mm.

Cutter:

The side-and-face cutter is designed with cutting teeth on its side as well as its circumference. They are made in varying diameters and widths depending on the application. The teeth on the side allow the cutter to make unbalanced cuts (cutting on one side only) without deflecting the cutter as would happen with a slitting saw or slot cutter (no side teeth). Cutters of this form factor

were the earliest milling cutters developed. From the 1810s to at least the 1880s they were the most common form of milling cutter, whereas today that distinction probably goes to end mills.

4. Calculations

1) For Motor:

By Trial and Error Method:

$$\text{Power} = \text{Voltage} \times \text{Current}$$

$$= V \times A$$

$$= 12 \times 1$$

$$\text{Power} = 12 \text{ Watt}$$

Trial 1: Take $N = 100 \text{ rpm}$

We know that,

$$\text{Power} = \text{Voltage} \times \text{Current} = 12 \text{ Watt}$$

And also,

$$\text{Power} = 2\pi NT/60$$

Therefore,

$$12 = (2 \times \pi \times 100 \times T)/60$$

Hence,

$$T = 1.146 \text{ Nm}$$

But practically this calculated torque is not sufficient for the working of the system and hence we go for another trial as follows;

Trial 2: Take $N = 60 \text{ rpm}$

We know that,

$$\text{Power} = \text{Voltage} \times \text{Current} = 12 \text{ W}$$

And also,

$$\text{Power} = 2\pi NT/60$$

Therefore,

$$12 = (2 \times \pi \times 60 \times T)/60$$

Hence,

$$T = 1.91 \text{ Nm}$$

Practically this calculated torque is sufficient for the working of system but still calculating for another iteration and hence trial 3 is as follows;

Trial 3: Take $N = 30 \text{ rpm}$

We know that,

$$\text{Power} = \text{Voltage} \times \text{Current} = 12 \text{ W}$$

And also,

$$\text{Power} = 2\pi NT/60$$

Therefore,

$$12 = (2 \times \pi \times 30 \times T)/60$$

Hence,

$$T = 3.81 \text{ Nm}$$

But practically this calculated torque is more than sufficient for the working of the system and hence not

Sr. No.	Speed(rpm)	Torque(Nm.)
1.	100	1.146
2.	60	1.91
3.	30	3.81

selecting this motor.

The standard DC Motor Speed Calculations are as follows:

Stock Item	Molon Part No.	Torque (in. - lb.)	Torque (in. - oz.)	FIL Speed (RPM)	FIL Amps	Dimensions "X" "Y" "Z"
12 VOLT	C20-1202-1M	18	288	1.5	0.11	1.555 3.008 1.091
X	C20-1204-1M	18	288	3.5	0.16	1.401 3.059 1.091
X	C20-1207-1M	18	288	7	0.50	1.401 2.854 1.091
X	C20-1209-1M	18	288	9	0.38	1.401 2.854 1.091
X	C20-1217-1M	18	288	17	0.58	1.225 3.380 1.409
X	C20-1230-1M	18	288	30	1.00	1.225 3.380 1.409
	C20-1211-1M	7.8	125	11	0.22	1.225 2.883 1.091
	C20-1223-1M	7.8	125	23	0.68	1.225 2.678 1.091
	C20-1231-1M	7.8	125	31	0.52	1.225 2.678 1.091
X	C20-1256-1M	7.8	125	56	0.85	1.056 3.211 1.409
X	C20-12108-1M	7.8	125	108	1.40	1.056 3.211 1.409
	C20-1242-1M	2.2	36	42	0.22	1.056 2.714 1.091
	C20-1286-1M	2.2	36	86	0.68	1.056 2.509 1.091
	C20-12115-1M	2.2	36	115	0.52	1.056 2.509 1.091
	C20-12210-1M	2.2	36	210	0.85	0.895 3.050 1.409
	C20-12400-1M	2.2	36	400	1.40	0.895 3.050 1.409

After Comparing standard and obtained values with different standard torques and speeds we came to the conclusion that the best possible motor which is required for the sufficient working of the system is the Motor with 31 rpm and torque 3.85 Nm.

2) For Cylinder

We are using double acting cylinder and Force exerted by double acting cylinder on outstroke is given by:

$$F = P\pi ((d_1)^2 - (d_2)^2) / 4$$

Where,

d1 = Diameter of Piston = 48 mm

d2 = Piston rod diameter = 10 mm

P = 3 bar = 300 kPa

Therefore,

$$F = 300\pi ((48 \times 48) - (10 \times 10)) / 4$$

$$F = 519.305 \text{ N}$$

Design of Shaft:

For a main shaft which is a power generator, power is given as,

$$P = F \times V \text{----- (1)}$$

Our whole assembly will have weight approximately equal to 60kilograms. Thus total force acting will be on 5 wheels.

Out of those 4 wheels we have maximum load acting on rear wheels mounted on shaft. This shaft is subjected to approximately 50 kilograms of load. So force acting on shaft is given by,

$$F = mg \text{----- (2)}$$

Putting m=50kg

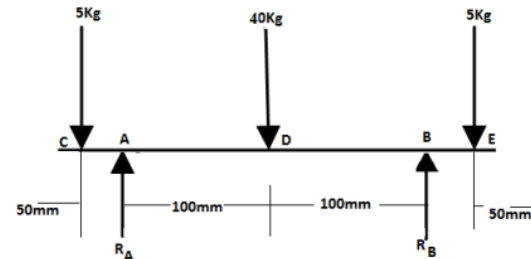


Fig. 8 Free body diagram

$$g = 9.81 \text{ m/s}^2$$

Thus

$$F = 50 \times 9.81 = 490.5 \text{ N}$$

Velocity is found out to be 10 cm/s i.e. V = 0.10 m/s

Thus

$$\text{Power, } P = 490.5 \times 0.10 = 49.05 \text{ watts}$$

We know that torque is given as, $T = P \times 60 / (2\pi n)$

Assuming No. of Revolution, n=50rpm

Thus, we have Torque, $T = 49.05 \times 60 / (2\pi \times 25)$

$$T = 9.36 \times 10^3 \text{ N-mm}$$

For a given shaft we have from diagram, Vertical reactions at wheels i.e. fixed supports,

$$R_A = R_B = (5 + 40 + 5) / 2 = 25 \text{ kg} = 25 \times 9.81$$

$$= 245.25 \text{ N}$$

From bending moment diagram, maximum bending moment is found to be $M = 1750 \text{ Kg-mm} = 17.167 \times 10^3 \text{ N-mm}$

The resultant moment on a given shaft is given as

$$MR = (M + T) / 2$$

$$= ((17.167 \times 10^3)^2 + (9.36 \times 10^3)^2)^{1/2}$$

$$= 19.552 \times 10^3 \text{ N-mm}$$

Also we know that shaft diameter is given as,

$$d = [(MR \times 16) / (\pi \times \tau)]^{1/3}$$

SHEAR STRESS VALUES

Service Condition	τ_i (MPa)
Heavily loaded short shafts carrying no axial load	48-106
Multiple bearing long shafts carrying no axial load	13-22
Axially loaded shafts (bevel gear drive or helical gear drive)	8-10
Shafts working under heavy overloads (stone crushers, etc.)	4.5-5.3

Consider shear stress, $\tau = 50 \text{ MPa}$

$$d = [((19.552 \times 10^3) \times 16) / (\pi \times 50)]^{1/3}$$

$$d = 12.581 \text{ mm}$$

This is ideal diameter of shaft which is needed. Since a shaft may be subjected to extra load as it has to work in rough conditions and from availability point of view, we chose a safe diameter from DDHB (Table 3.5a) of standard shaft diameter of 15 mm. Thus diameter of shaft, $d = 15 \text{ mm}$

CAD DESIGN MODEL:

From the above calculations of various parts used for

design we created a CAD design model on CATIA software. The various views of that model is shown below:

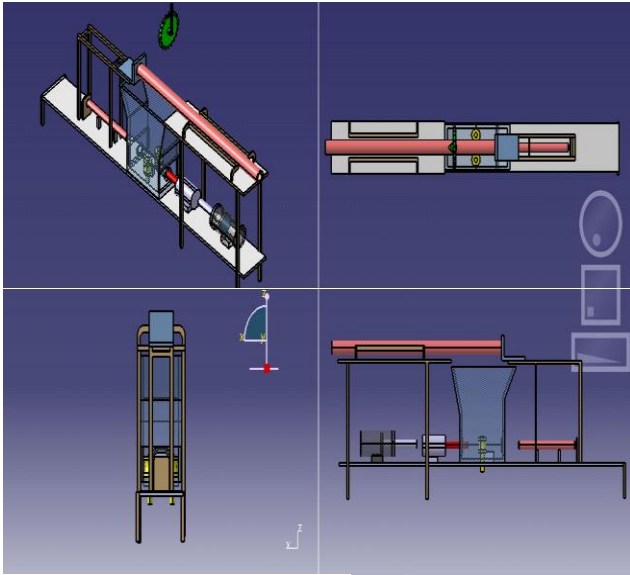


Fig.3

When the sleeve will fall down inside the hopper after getting cut by the cutter, then the pneumatic cylinder will get actuated and it will move forward to make its forward stroke. The piston of pneumatic cylinder will push the sleeve to insert it into the rod.

WE have used rotors for exact alignment of sleeve to match the axis of sleeve and rod. The arrangement of the sleeve and rod is shown in the following view for inserting the sleeve into the rod. The purpose of the rollers is to support the sleeve and minimize its degree of freedom in only one direction.

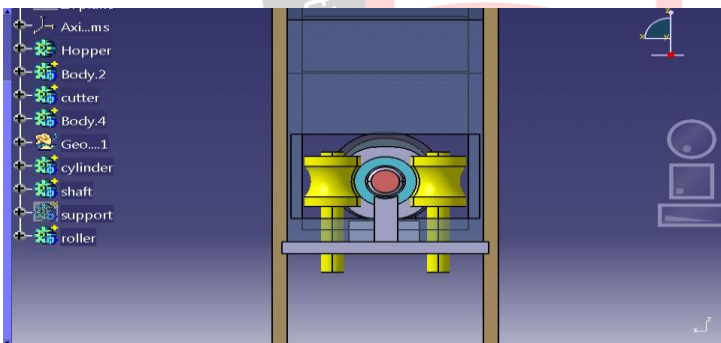


Fig.4

Conclusion:

The use of automation for sleeve insertion process will increase the production rate, maintain uniform consistency in the products and maintain good hygiene, which will indirectly increase the efficiency of the production. Fatigue causing activities will be eliminated due to lesser human intervention. The lead time of the product will also be decreased and will directly affect the on-time delivery performance of the company as lead time reduction is one of the key factors in all competitive companies. The capacity of the product can be varied by easily scaling the product for required capacity due to which it can be used for small to large scale production as per the vendor's requirement. Since the cost of Sleeve Inserting Machine is lesser it is affordable to small vendor which are the main targets for manufacturing the machine.

References

- Akshay Kumar M. Shinde, Design & Manufacturing of Cork Fitting Machine International Research Journal of Engineering and Technology (IRJET)
- Gil, L. Cork composites: A review. Materials 2009, 2, 776–789.
- Amol More, Modification and Implementation of Lower Link Bush Press Machine 2nd International Conference on recent trends in Engineering science and management.
- Mangesh Saraf, Application of microcontroller for accurate control and safety of SPMA Case study on sweet making
- A. Ola, G. A. Mekanjuola, E. S. A. Ajisegiri, O. Aremu and T. M. A. Olayanju(2014), Development of manually operated dough dicing machine for cold extrusion, *JNSET*.
- Riaz M N(2000), extruders in food application, *CRC press publication*, pg 3 to 18
- Richards W hartel, Denis heldman(2012), Principles of food processing, pg 273 to 279