Design of a Prototyping Press for 3-D Monolithic Compliant Mechanisms

by

Thomas J. Slowe

Submitted to the Department of Mechanical Engineering in Partial Fulfillment of the Requirements for the Degree of

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ABSTRACT

The Precision Systems Design and Manufacturing Lab at the Massachusetts Institute of Technology has the need for a metal forming device capable of applying a plastic deformation to two-dimensional sheet metal templates of up to 1/8-inch thickness and 8-inch diameter in order to transform them into prototype three-dimensional monolithic compliant mechanisms. These mechanisms have applications in industrial positioning as they are highly accurate and free from normal performance reducers such as friction, wear, and backlash. This thesis presents the design of a prototyping press capable of achieving the deformation required to produce the 3DMCMs from their 2D templates. The prototyping press that is developed herein utilizes a multiple-piston, hydro-pneumatic cylinder to deliver up to 5,000 lbf over a 4-inch stroke. The press offers force sensing to within 10%, displacement sensing to within 0.005 inches, and rate control centered around a 6-inch per minute average rate. It is powered by a compressed air supply at up to 100 psi and motion is controlled by a single electrical solenoid shut-off valve.

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Chapter 1

INTRODUCTION

1.1 Motivation

The Precision Systems Design and Manufacturing (PSDAM) Lab at the Massachusetts Institute of Technology has the need for a device capable of plastically deforming sheet metal structures up to 0.125 inch thickness in an accurate and repeatable manner. The device must be able to form two-dimensional shapes into prototype three-dimensional monolithic compliant mechanisms (3DMCMs) via basic metal forming processes. The prototype 3DMCMs are currently formed from their two-dimensional templates with an arbor press controlled manually by an operator. The operator is the only source of control in the current system and the mechanisms produced are each unique in their exact geometry. The current method of prototype production does not meet the needs of lab.

1.1.1 Three-Dimensional Monolithic Compliant Mechanisms

Three-dimensional monolithic compliant mechanisms are a relatively new development being introduced by the MIT PSDAM Lab. They are formed in a low cost, two step process. First, a pattern or template is cut out of a two-dimensional sheet of metal, generally steel or aluminum. The template is cut using a low cost manufacturing process such as an abrasive water jet in the case of the PSDAM lab. The two-dimensional template is then plastically deformed by some forming device, the design of which the work of this thesis will focus. The forming is generally a simple bending or stretching operation used to extend the two-dimensional template into the third dimension.

Prior investigators [1] have utilized 2DMCMs for the purpose of alignment, specifically optical alignment, but 3DMCMs offer a distinct performance advantage: significantly larger work volume. What is more, it is thought that 3DMCMs can be created to achieve nanometer resolution over their larger work volume, thus maintaining the accuracy of the 2DMCMs already in existence. Thus, the prototype 3DMCMs being produced by the PSDAM Lab hold exciting possibilities for industrial application.

The application being investigated currently by the lab is the usefulness of 3DMCMs in positioning equipment. The manufacturing industry is constantly seeking positioning equipment that can deliver improvements in precision at the lowest possible cost. Positioning equipment affects industry's bottom line by constraining the performance of machines performing even the most basic manufacturing processes such as measurement, parts alignment, and assembly. Operations that utilize 3DMCMs for positioning will notice the virtual elimination of performance reducers such as friction, backlash, and wear, which are not present in single component mechanisms yet plague all multiple component positioning mechanisms [2].

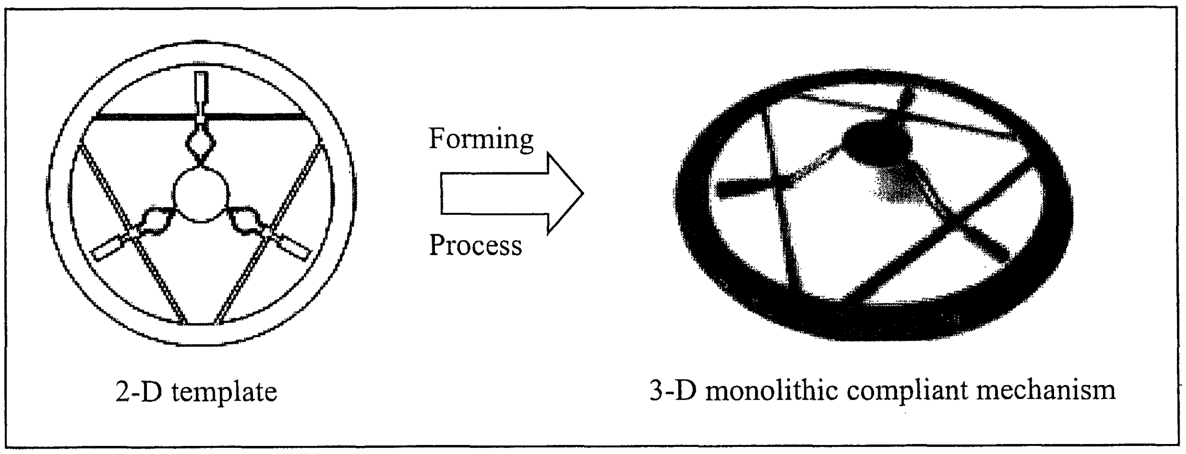


Figure 1.1 - Diagram showing 2D template and corresponding 3DMCM [2]

1.2 Thesis Scope and Organization

This thesis aims to present the design and qualification requirements of a device which solves the need of the PSDAM Lab for a forming mechanism that will assist in the production of prototype 3DMCMs. The device described herein will apply a controlled plastic deformation to the two-dimension sheet metal templates, thus turning them into 3DMCMs. As this thesis is meant to address the theoretical design as well as the practical application of the metal forming device it focuses on, considerations for construction, use, and maintenance of the device are also within the scope of this thesis.

The first chapter of this thesis will briefly outline the need for the device described herein. A short background on the industrial need for 3DMCMs and the laboratory need for a forming device is presented in this chapter.

The second chapter addresses the prior art in metal forming technology. The topics of frame type and actuation method are addressed in detail sufficient to justify the mechanism design presented in the third chapter.

The third chapter proceeds immediately into the design of the metal forming device. Functional requirements are presented and various technologies that exist to meet these requirements are evaluated for their ability to meet the functional requirements. The design of the overall system and selection of components is presented with justification for each selection. The forming device and a control method for it are considered in this third chapter.

The fourth chapter concludes the thesis with a summary of the work accomplished herein, its potential for impact, and suggestions for future work to support the construction and utilization of the forming device that was designed.

Chapter 2

PRIOR ART

2.1 Overview

The device that is designed in this thesis performs a common function: the plastic deformation of sheet metal. While the functional requirements presented in the next chapter will determine a unique device to solve the problem presented, there is no need for the final system to be entirely novel. In fact, a goal of this design problem is to best use existing technology to meet the functional requirements of the metal forming task. This chapter will therefore present an overview of some common metal forming technologies already in existence from which the basic design of this device will be taken. Specifically, the types and uses of various frame types and actuation methods will be presented for consideration. The next chapter will then explain why a particular technology was chosen for the final design.

2.2 Frame Types

The basic frame types for sheet-metal forming equipment are commonly known as the C-frame structure and the gantry frame structure. Figure 2.1 presents a visual of these two distinct frame types.

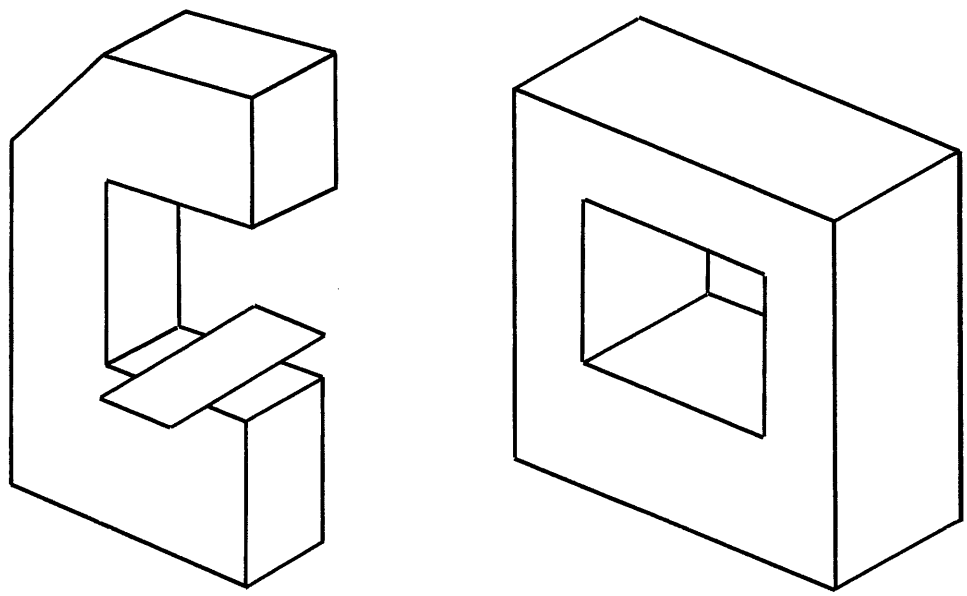


Figure 2.1 - Geometric representation of C-frame (left) and gantry frame

The C-frame is commonly used on equipment producing a relatively low force or on equipment to which easy access from all directions is required. C-frames are not as stiff as their gantry frame counterparts, but they offer more convenient workpiece accessibility since the front, back, and both sides of the workpiece are accessible from many angles [3]. C-frames might typically be used for applications where the geometry of the workpiece varies frequently and where the workpiece is discrete and not wound on a continuous roll or other continuous feed arrangement.

A major drawback to the use of a C-frame for metal forming operations is the lack of stiffness compared to a comparably sized gantry frame machine. Operations involving large compressive loads, such as metal forming, will suffer inaccurate and often inconsistent results due to the lack of rigidity of the system. Also, because the frame is not symmetrically supported about the workpiece, thermal expansion of the machine frame will cause non-uniform effects on the workpiece and may be hard to predict and account for. Both thermal expansion and forceful deformation of the frame from compressive loading will create linear and angular displacement errors as the C-frame will arch forward or backwards depending on the stimulus (e.g. heat or force). Figure 2.2 illustrates this deformation.

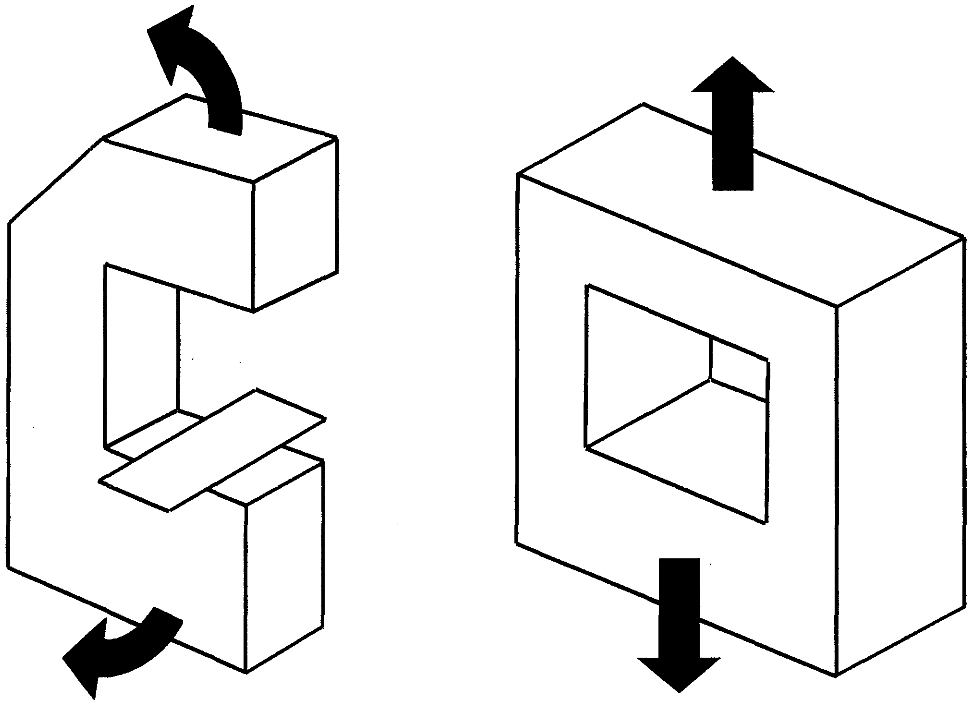


Figure 2.2 - Schematic showing direction of force-driven deformation

To improve upon the drawbacks of the C-frame geometry, a gantry type frame may be used. The gantry type frame is symmetrically supported with either solid sides (double column configuration) or posts in each of four corners (pillar configuration). Because the gantry frame supports the structure of a machine in a symmetrical manner with respect to the workpiece, deformations resulting from the thermal expansion of constituent materials are more uniformly distributed and are easier to predict, measure, and account for through a control algorithm, for example. Forceful deformations of the structure are also less severe due to improvements in structure stiffness resulting from the use of a gantry style frame. These features of the gantry frame type commonly make it the preferred frame geometry for high load equipment as well as continuous material flow equipment such as metal forming machines that accept a rolling input.

2.3 Actuator Types & Configurations

While there are many choices of actuators available for use in metal forming machines, there are three general classes of actuation schemes that are typically seen on this type of equipment: mechanical actuation schemes, screw-type actuation schemes, and fluid actuation schemes. For the purposes of this work, an "actuation scheme" refers to the combination of the true actuator (e.g. an electric motor) and the associated mechanism that enables the press motion (e.g. a power screw). Each class of actuation schemes can be distinguished by the factor that primarily limits its function. The common factors affecting performance of a metal press are load, stroke, and energy [4]. Each factor limits a different class of equipment as explained below.

Mechanical actuation methods are primarily stroke limited and do not require the use of fluids for their power. Within this category are found two main configurations: crank-type mechanisms and eccentric-type mechanisms. These actuation methods depend on an electric motor to power a flywheel for energy. Mechanical actuators are capable of delivering very high forces, but deliver maximum load only at the bottom center of the actuator's stroke due to the geometry of the design. Also, the load being applied to a particular spot on the workpiece at any given time may vary widely from the applied load at another spot in the rotation of the flywheel. The inconsistency of the applied load of the mechanically actuated machine over its stroke makes it undesirable for applications requiring a long power stroke, but ideal for applications where high force is required only at the end of the stroke, as in many small deformation operations such as punching.

Screw-type actuation methods are another means of actuating a metal forming machine. Like other mechanical presses, they also utilize a flywheel to generate the necessary energy from an electric motor. In the case of screw-type metal presses however, the flywheel energy drives a power screw which then acts on the workpiece. Screw-type actuation schemes are useful for low production rates as they typically require intermittent motor use and frequent changes in the motor drive direction. Screw-type presses are also useful for production of precision components because of their ability to freeze at any position in the actuation cycle and to easily reverse.

Screw-type presses can provide a high-force output that is consistent in force distribution over the often lengthy stroke. Stroke length is not as tightly constrained with screw-type presses as it is with other mechanical presses due to the simple possibility of extending the screw length rather than redesigning a crank or eccentric cam system. Because screw-type presses derive their energy from a flywheel but are not stroke limited, they are said to be energy limited.

Finally, there are fluid-type actuation schemes. Fluid actuators utilize either liquids, gases, or a combination of both to produce a load-limited output. The most basic fluid actuator would be a single piston and cylinder which is actuated by applying pressurized fluid to one side of the piston and using this fluid to extend the piston outwards while fluid freely exits from the backside of the piston. Piston retraction is often accomplished by pressurizing fluid on the opposite side of the piston. Alternatively, a mechanical spring may be used to return the piston to its original position when the supply pressure is removed.

Hydraulic oil is generally used as the working fluid in high force applications utilizing fluid actuators because the use of air or any other gas is subject to many dynamic issues stemming from its compressibility. Control of the stroke speed on hydraulic machinery is easily accomplished by controlling the flow rate of the hydraulic oil at any point during the stroke. One limitation in the use of fluid actuators is the speed at which stroke reversal may be achieved. Because of the dynamic properties of fluids, rapid stroke reversal beyond a certain rate is not possible with a fluid actuator. Hydraulic actuators are therefore useful when a constant force is needed at a controlled speed and rapid reversal is not required.

Chapter 3

DESIGN OF THE PROTOTYPING PRESS

3.1 Functional Requirements

The functional requirements for the prototyping press can be broken into several more specific subsets. They would be: fixturing requirements, force requirements, speed requirements, and sensing requirements. Together, this set of functional requirements specifies the capabilities the prototype press must have. While these functional requirements do not determine a unique design, their specification does ensure the final design will satisfy the needs of the PSDAM Lab.

3.1.1 Fixturing Requirements

Fixturing refers to the location of two or more components with respect to one another. In the case of the prototyping press, the main concern is the location of the two-dimensional template with respect to the tooling of the press. The following list of points prescribes the relationship intended between the template and the tooling.

- When the tooling comes into contact with the template, the two pieces should be in parallel planes.

- The tooling should be centered with respect to the center of the template and there should be some clearance between the two pieces before the tooling comes into contact with the template.
- The template should be able to be held rigidly in place while constrained in all six degrees of freedom. The tooling should only be able to cause movement of the template via deformation of the template material; relative motion of the entire template to the press should not be allowed during the forming operation.

Furthermore, the fixturing should provide a means of securing parts of the template even against deformation. This requirement is necessary to help ensure that deformation takes place only at the locations on the template where it is desired. The fixturing must be able to meet these requirements throughout the duration of the forming operation and for each forming operation it performs. That is to say, the press must provide fixturing that performs repeatedly in order that the performance of 3DMCMs formed on this prototyping press may be fairly compared against one another during testing.

With "fixturing" now defined above, the geometric capabilities of the prototyping press will be specified. The press must be able to fixture a template no thicker than 1/8-inch, and no larger a device than would fit in an 8-inch diameter circle, centered under the centerline of the travel of the press tooling.

3.1.2 Force Requirements

The PSDAM Lab determined that the press should be able to perform its basic forming operations on common steel and aluminum alloys up to 0.125 inches in thickness. In order to determine the force requirements of a prototyping press capable of meeting this need, a few steps were taken. First, a worst-case template was specified. Then calculations were performed to determine the force required to deform the worst-case template as if it were enduring a simple punching operation. The justification for this process and the calculations performed are specified in Appendix A. The result was a force requirement of 5,000 lbf (2.5 tons).

3.1.3 Speed Requirements

The speed of the press stroke should be slow enough to be easily observed visually and to not introduce any unwanted shock loads to the template during the plastic deformation process. A speed of 6.0 inches per minute would allow typical deformation (1.0 inch) to be completed within approximately 10 seconds and provides the desired speed characteristics. The speed should, however, be easily varied within a reasonable range at the discretion of the investigator.

The return stroke of the press is not specified as its precise value is unimportant to the investigators needs. The main consideration in designing the return stroke should be safety as it should be quick and easy to remove the load from the press tooling and reverse its motion in case the operator becomes endangered by the motion of the tooling.

3.1.4 Sensing Requirements

The prototyping press should have sensing abilities that meet the criteria set forth in this section. First, the press should be able to sense the force being applied to the template at all stages of motion. Press force should be able to be monitored to ensure consistency between trials as well as to allow the application of a predetermined force to the template and observe variations in this force during the pressing operation. Press force should be measurable to within an accuracy of 10% to allow for estimation of optimal press force conditions under various circumstances.

Secondly, the prototyping press should able to sense the depth to which the template has been deformed as it is deformed. Throughout the pressing process, the depth of deformation should be accessible to the press operator for operations decisions and data collection. Displacement measurement must be accurate to within a few thousandths of an inch in order to make reliable comparisons among structures produced with the press.

In both instances, the sensors used on the prototyping press should allow for easy collection of data via electronic methods. The investigator should not be made responsible for recording real-time data manually. Automated data collection, both of force and displacement data, will allow the investigator to compare results of various trials and decide upon optimal operation conditions.

3.1.5 Repeatability Requirements

All of the aforementioned requirements are compounded by the need for repeatability in the performance of the prototyping press. As the press will be designed to accommodate one template at any given time, the performance of the press must be nearly identical during each use to ensure each individual template specimen is acted upon in a predictable manner that allows for comparable observations to be made. Also, once a desired configuration is discovered for a given template design, system parameters should be able to be controlled such that multiple, nearly identical mechanisms can be produced by the press for testing purposes.

Table 3.1 - Functional requirements of the prototyping press

Fixturing (capacity)	0.125 inch thickness, 8.0 inch diameter
Force	5 000 lbf (2.5 tons)
Speed	Average 6.0 inches/minute, variable
Sensing Accuracy	Force 10%; Displacement 0.005 in.

3.2 Concept Evaluation

In order to narrow down the field of candidate technologies, careful evaluation of each technology presented in Chapter 2 was carried out. The actuation schemes presented Chapter 2 represent the practical choices for actuation of a material press with the general load requirements to which this press will be subject. In comparing this set of technologies, a list of criteria was developed. The list reflects the criteria that were important to the investigators of the PSDAM lab. These criteria are: force control, position control, speed control, force sensor compatibility, position sensor compatibility, size, weight, cost, and safety.

3.2.1 Evaluation Parameters

To begin with, the actuation scheme must be able to meet the basic functional requirements set forth in the earlier part of this chapter. Namely, the actuation scheme must provide the required force, speed, sensing, and fixturing capabilities in a comprehensive prototyping press. While there could be countless other capabilities that may be desired, the functional requirements determine the minimum performance standards the machine must achieve.

Additionally, the actuation schemes being considered must be evaluated in terms of other factors that do not truly belong to the set of functional requirements. These factors are cost, size, weight, and safety. Cost is an important factor in almost all design projects and requires no additional justification here. Size and weight were important in this design challenge because the device that is designed herein is intended for laboratory use where space is at a premium. An excellent prototyping press that consumed half of the available lab space would not be a good solution to this problem.

Safety is also of paramount concern in a laboratory environment. The prototyping press may be used by individuals not specifically trained in its use and must not pose great risk of injury to such operators. While training on the operation of the press is recommended, the reality of a laboratory setting is that training may not be provided so intrinsic safety is important.

3.2.2 Actuation Scheme Evaluation

Table 3.2 presents a summary of comparisons between the actuation schemes considered for the prototyping press. The text that follows explains the reasoning behind the grading of the schemes shown in the Pugh chart.

Table 3.2 - Evaluation of actuation technologies

Criteria	Fluid Power			Mechanical Power		
	Hydraulic	Pneumatic	**Hydro-Pneumatic**	Power Screw	Crank	Eccentric
Force Control	0	-	**0**	-	-	-
Position Control	0	0	**0**	0	-	-
Speed Control	0	-	**0**	0	0	0
Force Sensor Compatibility	0	0	**0**	-	-	-
Position Sensor Compatibility	0	+	**0**	0	0	0
Size & Weight	0	+	+	-	-	-
Cost	0	+	+	+	-	-
Safety	0	+	+	+	0	0
Totals	0	+2	**+3**	-1	-5	-5

The mechanical means of actuation will be considered first. The most commonly found mechanical actuation schemes used on industrial presses involve the use of an electric motor and flywheel to generate the large amount of power required by the press. The end effector is generally either a crank or some type of eccentric that creates a very large force at the bottom dead center of its stroke. Actuation schemes utilizing this type of mechanism are common in industrial applications where repetition is the goal. Such machinery can very repeatably utilize a desired force to induce a desired displacement in a template material.

The problem with this actuation scheme is that it does not offer adequate flexibility in its operation. While very useful for producing large quantities of identical parts, this technology cannot easily be adapted for the type of one-off production the PSDAM Lab requires. In order to properly adapt this technology for the small scale production of prototype mechanisms, the entire crank or eccentric mechanism would need to be easily reconfigurable and this is not a very practical solution to the adaptability problem. Additionally, the size, weight, and cost, of these actuators are not favorable because of the inertia they need to generate before impacting the template.

The remaining common mechanical solution is the power screw. This is not a common option in high volume industrial settings, but is not uncommon for smaller laboratory pressing and clamping needs. The reason it is not popular in large scale production is the speed at which the power screw acts. To produce the required deformation via screw advancement and then reset the screw by reversing the direction of spin, a significant amount of time is required. Still, this remains a possible solution for this laboratory need since high speed is not required. A remaining drawback may be the safety aspect. If some part of the operator were to become caught in the path of the power screw, the screw could not be as quickly reversed or disabled as a hydraulic circuit can by dumping the pressure supply.

Fluid power presents a number of attractive benefits as well as some drawbacks. Hydraulic actuators are an industry standard for fluid power presses. Hydraulic power can produce very high forces with relatively simple actuators, namely hydraulic cylinders. Press speed can be easily controlled by varying the flow rate in or out of a hydraulic cylinder and there is no hard limit on the amount of force that can be designed into a system. What is more, press speed and force can be easily adjusted by adjusting either the flow rate or the input pressure respectively.

Some detractors to using hydraulic power in a laboratory setting however, are the size, weight, cost, and safety of some of the required components. Since an electric motor typically drives a hydraulic motor which then outputs a high pressure fluid, a significant investment in actuators is required. Also, the high pressure fluid poses a safety hazard in the event of rupture in any of the fluid paths. Fluid also is bound to leak from the system at some point since the system cannot be entirely sealed to prevent such losses. While the losses are not a significant source of cost or detractor from performance, they do create an unclean environment that is not desirable in a laboratory.

Pneumatic press machinery, while not popular in industry, offers some advantages to hydraulic machinery, but also suffers some disadvantages. Pneumatic circuits can be cheaply supplied with power from a compressed air source that may typically exist in a laboratory setting. Also, pneumatic equipment offers the advantage of being able to exhaust to the atmosphere rather than be required to reclaim its spent fluid as hydraulic equipment must.

The detractors to pneumatic equipment are most noticeably the force and speed control. While input force can be manipulated by changing the input pressure, air is very compressible and so the result of changing the input pressure while performing an operation may be nonlinear and more difficult to predict when compared to hydraulic equipment. Also because of its compressibility, air is more difficult to use for speed control. While the flow of hydraulic oil can be easily controlled to produce a linearly variable restrictor, air flow suffers from more noticeable dynamic effects and cannot be as easily controlled in its flow to produce a simple speed control.

3.3 Concept Selection

The Pugh chart presented in Table 3.2 indicates the choice of hydro-pneumatic power to be a strong one for the given evaluation criteria. Hydro-pneumatic, as used here, refers to a press that utilizes a double acting cylinder with pressurized air on one side of the piston, acting to extend it, and hydraulic oil on the other side of the piston, resisting its motion.

In this case, pneumatic pressure acts on the piston and is responsible for the extension of the piston and tooling, thus providing an energy source that is affordable, convenient, clean, and reasonably sized for a laboratory environment.

More importantly however, the detractors of pneumatic technology, mentioned in section 3.2.2, are neutralized by the use of some hydraulic circuitry. The oil on the back side of the piston can be precisely metered during its outflow to allow for precise and repeatable press speed control. Furthermore, the hydraulic oil provides a resistance to the cylinder air during extension so the force exerted is controlled with respect to the dynamic effects of rapidly expanding air in the press. Finally, the oil is useful in that it provides a hard stop for the piston when the operator wishes to cease the pressing operation. As oil is very incompressible, the amount of overshoot when the operator signals a stop is minimized. Pneumatic machinery would suffer unacceptable overshoot due to the compressibility of the air used to power it.

3.4 Press Design

In this section, the layout of a hydro-pneumatic circuit that meets the functional requirements of the prototyping press will be presented. An explanation of the function of each component in the scheme of the machine will be included along with practical considerations that were given when choosing the components.

3.4.1 Circuit Function

A schematic of the hydro-pneumatic circuit designed to meet the functional requirements previously laid out is found in Figure 3.1. The circuit utilizes a combination of hydraulic and pneumatic components to produce a hybrid power transmission circuit capable of achieving a controlled output force and displacement from a cylinder actuator.

Beginning at the top of the figure, there is a regulated source of compressed air. The pressure of the compressed air is measured by the pressure gage located downstream of the air source. After leaving the air source, air encounters a 4-way, 2-position directional control valve. The directional control valves positions are associated with either extending or retracting the piston in the cylinder shown at the bottom of the figure.

When in the position to extend the piston, the directional control valve allows air to flow to the left side (Figure 3.1) of the piston. The piston remains locked in place however, until the shut-off valve is opened by solenoid action. The shut-off valve is normally closed, preventing cylinder motion until a current is applied to the solenoid. If current is now applied to the solenoid, the shut-off valve opens and the hydraulic oil residing on the right side of the piston (Figure 3.1) is forced from the cylinder by the pressurized piston and passes through the flow control.

The flow control allows the speed of the press to be adjusted to a desired rate during the extension phase of the press cycle. The hydraulic oil leaving the cylinder, after passing through the shut-off valve and the flow control, enters the air/oil tank. During the extension of the piston, this tank acts as nothing more than a storage reservoir for the excess fluid exiting the cylinder. The tank accepts the oil and outputs an equal volume of air through the silencer vent located on the top side of the tank.

The aforementioned sequencing describes the process of piston extension. The process of piston retraction essentially proceeds in reverse, but with minor changes. To achieve piston retraction, the directional control valve must be reversed. Air from the compressed air source now flows under pressure into the air/oil tank.

As pressurized air is added to the tank, an equal volume of oil must exit the tank. The oil exits the tank and proceeds freely to the shut-off valve, bypassing the flow control through the non-return valve. Assuming the shut-off valve solenoid is energized, the oil flows through the shut-off valve and into the right side of the cylinder. The pressurized oil acts on the piston, forcing it to retract into the cylinder. The air on the left side of the piston vents to the atmosphere via the other silencer. This completes one full cycle of the cylinder circuit.

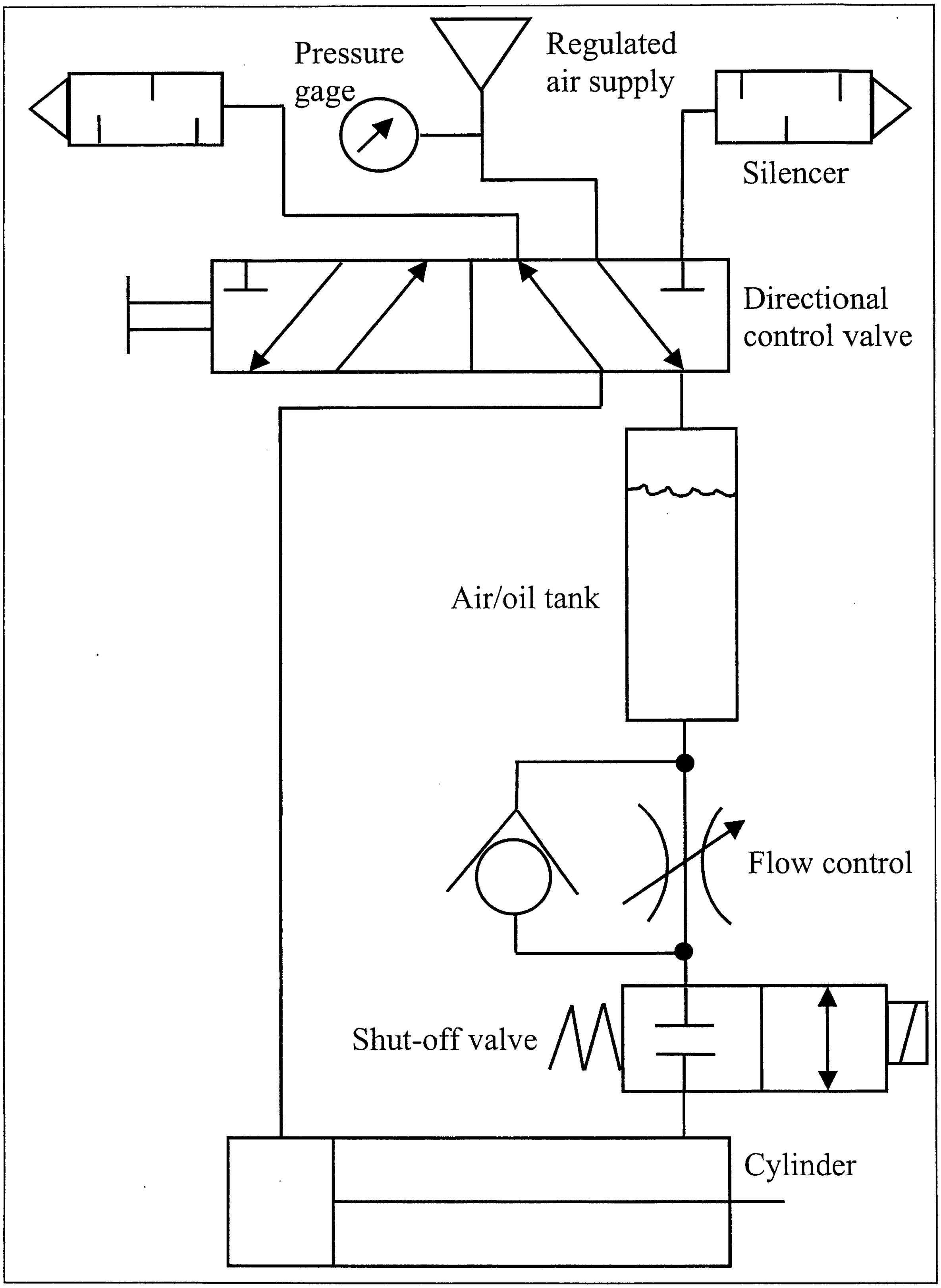


Figure 3.1 - Hydro-pneumatic circuit layout of the prototyping press

3.4.2 Component Descriptions

The key component in the circuit described above is the cylinder. The cylinder is the actuator that applies the force to the template, inducing the desired deformation. Though it is depicted as a simple double-acting cylinder in the diagram, it is actually a slightly more complex actuator.

The cylinder chosen for the prototyping press contains multiple pistons in separate chambers, attached to a common shaft (Figure 3.2). This type of cylinder utilizes integral channeling to carry pressurized air to and from the separate chambers. Since pressurized air acts on all piston simultaneously and all pistons act on a common shaft, the effective piston surface area is raised, reducing the diameter of the cylinder required to produce a given output force.

Output force of a fluid cylinder actuator is simply the product of the input pressure and the piston surface area contacted by this input pressure. In the case of the actuator chosen for the design of this device, three stages were used to produce a total of 56.4 sq. in. of effective contact area, thus producing 5,640 lbf when actuated by an air source at 100 psi.

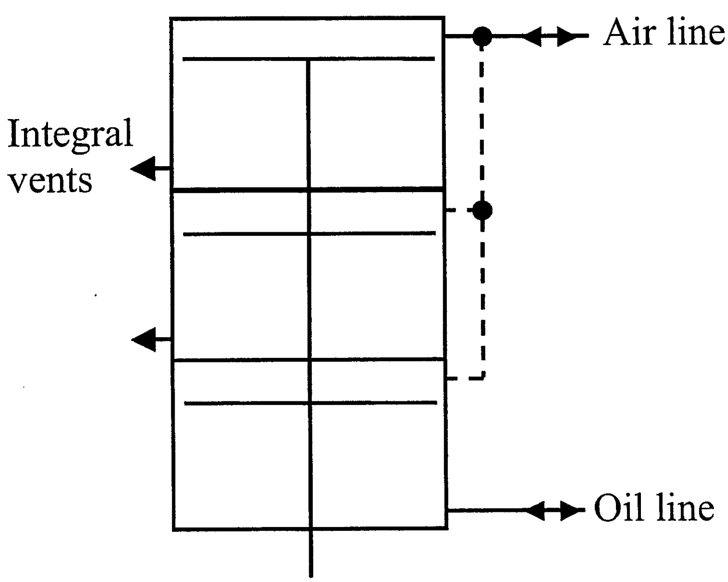


Figure 3.2 - Diagram of multiple piston cylinder

The specific actuator system chosen for use in the prototyping press is a slightly modified version of a unique air press manufactured by Fabco-Air, Inc. Detailed specifications for the actuator and its associated hardware can be found in Appendix B. Because the actuation system chosen is intended for press applications and is sold as a system, complete with the infrastructure required to adequately support the actuator for pressing applications, information regarding the design of the structural components is also reserved for Appendix B. The remaining components are specified to the degree necessary in Appendix C.

3.4.3 Displacement Sensor Selection

One component not addressed thus far is the displacement sensor. The functional requirements state that the prototyping press must be able to assess its position to a resolution of 0.005 inches for control purposes. The most common industrial applications of sensing technology to fluid pistons involve known position sensing. That is, one known position is detected during the piston's stroke, sending a signal to the controller that a known position has been reached. This is common practice in ensuring repeatability of manufacturing processes for large scale production. There are many ways of achieving this type of sensing including inductive, capacitive, and magnetic methods.

However, in the case of the prototyping press, continuous position sensing is required. The operator needs to know where the piston is at all times during the piston extension cycle so any desired position can be achieved by interpreting the sensor data and de-energizing the solenoid powering the shut-off valve, thus closing the valve and halting the piston's motion. The aforementioned sensing options (inductive, capacitive, and magnetic) do not help achieve this result as they only sense at or about a known position.

Two methods exist that can easily be employed to produce continuous position sensing at a reasonable cost. Each method involves the attachment of a device that runs in parallel to the piston, detecting piston motion only through the mechanical association of the piston to the sensing device. One such technology uses a linear potentiometer and the other, a linear variable differential transformer.

LVDT technology offers the advantage of higher accuracy and robustness as compared to the LP technology. LVDTs are make measurements without contacting the moving element, thus vastly improving wear life. While attractive over LPs for these reasons, LVDTs cost, on average, twice as much as LPs capable of a similar stroke length [5]. Because the performance derived from LPs is adequate for the stroke range dictated by the prototyping press, an LP will be used to measure the position of the piston at any given time.

Linear potentiometers function on the elementary principle of electrical resistance. The electrical resistivities of a moving shaft and a fixed element in the device are ascertained. Linear motion of the shaft causes a small wiper (usually made of a precious metal) to move relative to conductive element which is fastened to the casing of the linear potentiometer. A current is passed through the moving shaft, passes through the wiper, and then through the remaining portion of the fixed element. Based on the voltage difference seen by the device, the position of the wiper and thus, the piston, can be determined. Position can be determined to within 0.1% of the full stroke depending on the particular LP chosen [6].

In order to utilize an LP for displacement sensing of the piston, a rigid plate must be attached to the end of the piston. The LP should be mounted in parallel with the cylinder and have its open end attached to the same plate. When the piston moves, the connecting plate forces the LP to move in unison with it and an accurate displacement measurement can be taken.

Chapter 4

CONCLUSIONS AND FUTURE WORK

4.1 Conclusions

This work has presented the design of a prototyping press for making three-dimensional monolithic compliant mechanisms from two-dimensional templates. After presenting and evaluating a range of design options, the best suited technologies were chosen for incorporation into the device. While the technologies are not novel, they represent an effective means, both in terms of function and cost, of achieving the desired end result. Table 4.1 summarizes the final design choices that characterize this prototyping press.

The device designed herein is capable of inducing a desired plastic deformation in a sheet metal template of up to 0.125 inch thickness. The device accomplishes this by developing a force of up to 5,640 pounds from a compressed air source delivered at 100 psi. The device requires only the air source and a laboratory type electrical power source for its operation. Deformation is measured continuously during the process and used as feedback by the operator to control the device.

Table 4.1 - Final design characteristics

Frame type	Hybrid C-frame
Power source	Hydro-pneumatic
Actuator type	Multiple-piston cylinder
Displacement sensing	Linear potentiometer
Force sensing	Air pressure gage
Speed control	Hydraulic flow control
Directional control	Manual, directional control valve
Master control	Single solenoid shut-off valve

The prototype press design contributes to the goals of the PSDAM Lab at MIT by providing them with a device capable of forming prototype 3DMCMs from 2D templates in an accurate and repeatable manner. The investigators in the PSDAM Lab benefit from this improvement to the current manual deformation process by gaining the ability to create prototype structures that are essentially identical for testing purposes. In all engineering research, it is important to strive to isolate experimental variables from factors intended to remain constant so as to properly attribute experimental results. The prototyping press contributes to this need as well by allowing the investigators of the PSDAM Lab to separate the plastic deformation process from their group of desired experimental variables. Such consistency in and measurement of the plastic deformation process was not possible before the design of the prototyping press developed in this work.

4.2 Future Work

While this thesis completes the work of designing the prototyping press, a few practical considerations should be mentioned pertaining to the construction of the press for laboratory use. The cylinder and associated infrastructure is purchased as a disassembled unit and requires only basic assembly before use is possible. The cylinder should be arranged in the laboratory in a vertical orientation.

Also, the air/oil tank used for the storage of the working hydraulic oil should be oriented vertically. This latter consideration is especially important as a horizontal arrangement may allow the press to function, but air would easily be allowed to mix with the working hydraulic fluid causing the fluid to become compliant due to the compressibility of air. When the hydraulic fluid becomes compliant, it loses its ability to act as a hard stop for the press and performance of the prototyping press is significantly affected. Accuracy and repeatability requirements cannot be met if this is allowed to happen. Similarly, the air must be purged from the bottom chamber of the cylinder before it is put into use to avoid a similar effect.

When assembling the hydro-pneumatic circuit, care should taken to make fluid conduit lines only as long as is absolutely necessary and to minimize the number of hard angles in the course of the line. Unnecessary line length and sharp bends can contribute to significant energy losses in the lines and reduce the performance of the system accordingly.

In order to maintain the system, the oil level should be checked for consistency at regular intervals during use and the elements in the exhaust silencers should be replaced if a back pressure is detected. If consideration is given to the items mentioned in this section, the prototyping press can be expected to perform as outlined in this work.

REFERENCES

[1] Anderson, Gordon A.B. "A Six Degree of Freedom Flexural Positioning Stage," MS ME thesis, M.I.T., Cambridge, MA, 2003.

[2] "Mecha-gami: Formed and Reconfigurable Compliant Mechanisms" *MIT PCSDM Lab* webpage. <http://psdam.mit.edu/research.html> [accessed 19 May 2004]

[3] Kalpakjian, S. and Schmid, S.R., *Manufacturing Engineering and Technology.* Prentice-Hall, Upper Saddle River, NJ, 4th edition, 2001, pp. 432-434.

[4] Kalpakjian, S. and Schmid, S.R., *Manufacturing Engineering and Technology.* Prentice-Hall, Upper Saddle River, NJ, 4th edition, 2001, pp. 357-359.

[5] OMEGA Engineering, Inc., One Omega Drive, Stamford Connecticut 06907-0047. P.O.Box 4047. (800) 848-4286. www.omega.com.

[6] Honeywell Sensing & Control, 11 West Spring Street, Freeport, IL 61032. (800) 537-6945. <http://content.honeywell.com/sensing/contact/default_sc.asp>

[7] Kalpakjian, S. and Schmid, S.R., *Manufacturing Engineering and Technology.* Prentice-Hall, Upper Saddle River, NJ, 4th edition, 2001, p. 396.

[8] Dowling, Norman E., *Mechanical Behavior of Materials.* Prentice-Hall, Upper Saddle River, NJ, 1999, p. 120.

[9] Fabco-Air, Inc., 3716 NE 49th Ave., Gainesville, FL 32609-1699. (352) 373-3578. www.fabco-air.com.

[10] McMaster-Carr. P.O.Box 440, New Brunswick, NJ 08903-0440. (732) 329-3200. www.mcmaster.com.

[11] MSC Industrial Supply Co., Inc., 75 Maxess Rd., Melville, NY 11747-3151. (800) 645-7270. www.msc-direct.com.

Appendix A

JUSTIFICATION OF THE PRESS FORCE REQUIREMENT

A.1 The Worst-Case Template

The worst-case template was determined to be made of 2024-T4 aluminum alloy. This alloy is a common engineering material with an ultimate strength just higher than that of common AISI 1020 steel, as rolled. The shape of the structure is not as important as the amount of perimeter material presented to the punch is, so a specific shape was not specified. However, the amount of perimeter material was specified to be 0.75 inches, an amount almost double that of the templates currently being utilized by the PSDAM Lab (0.5 inch total perimeter length).

A.2 Punch Force Calculation

Once a worst-case template was decided upon, a calculation was performed to determine the punch force required to shear the worst-case template. This calculation was selected as it was determined to be a conservative estimate of the force that would be required by the machine. Since the template material does not actually reach shear, but rather is deformed into a three-dimensional structure, the punch force for a shearing operation provides an upper bound to the force required by the prototype press. Punch force was determined according to the following calculation:

$$F = 0.7 \cdot T \cdot L \cdot UTS \text{ [7]}, \tag{A.1}$$

where F is the punch force required, T is the thickness of the material, L is the length of shear (or perimeter of material presented to the punch), and UTS is the ultimate tensile strength of the material. For the worst-case template, the parameters had the following values:

$$T = 0.125 \text{ in.},$$
$$L = 0.75 \text{ in.},$$
$$UTS = 69 \text{ ksi [8]},$$

yielding a punch force requirement of F = 4,528 lbf. This requirement was increased to 5,000 lbf (2.5 tons) to provide for losses associated with the system.

Appendix B

SPECIFICATIONS FOR HYDRO-PNEUMATIC CYLINDER

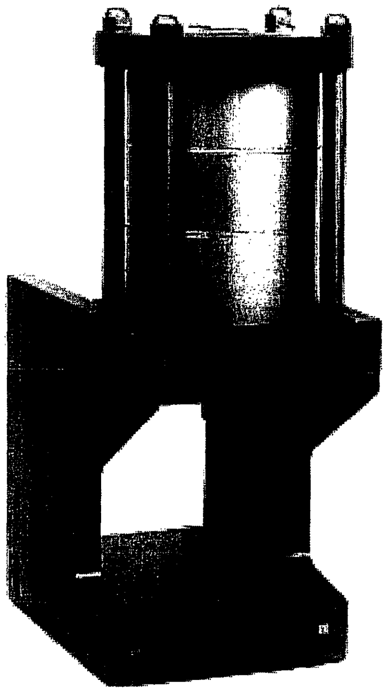
B.1 Multi-Piston Air Press

The cylinder and associated infrastructure utilized in the design of the prototyping press is manufactured by Fabco-Air, Inc., of Gainesville, Florida, USA. The design utilizes a modified version of the Fabco-Air Multi-Power Air Press. Specifically, the model utilized would be referred to by the Fabco-Air part number F55-Dx4-HS-V.

This model is based around a 3-stage multi-piston cylinder with modifications to accommodate the hydraulic oil utilized in the lower portion of the cylinder. The following two pages are excerpted from the Fabco-Air Multi-Power Presses catalog, number FP16. The material is provided courtesy of Fabco-Air, Inc [9].

Multi-Power® Air Presses

Provisions for operator protection are always the full responsibility of the user



Shown: Model F55 - Ax 1 - 10

Combining the muscle needed for production with the precision required for laboratory use

Fabco-Air applies the unique Multi-Power® Principle to a precision framework and base, providing you with the ultimate in a powerful, precision, compact, air-powered bench press for production or laboratory use.

How it works

The power cylinder uses multiple pistons attached to a common shaft. Each piston is isolated within its own chamber by means of baffles integral with the outer cylinder wall. Special internal porting allows air pressure to simultaneously energize all pistons –
enabling output forces in excess of 5 tons to be easily reached!

How it's built

The power cylinder has all the standard Multi-Power® features plus beefed up construction to meet the rigors of press type applications–

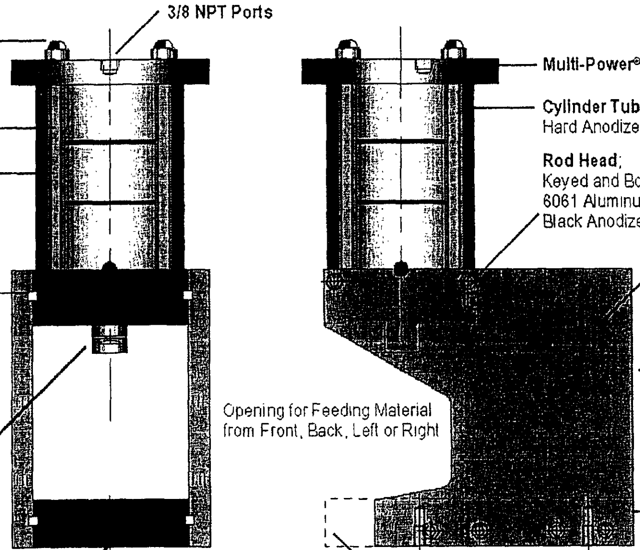
- Hard chrome plated stainless steel ram
- Extended rod bearing length
- Duralon® rod bearing
- 3/8 NPT ports are standard, with generous internal passages for air flow to allow high cycling speeds
- High strength aluminum frame plates are machined on the edges and have precision-located keyways
- Plated steel keys mate the cylinder head and base plate to the frame plates, providing accurate alignment and rigid construction

The keyed and bolted, high-strength construction provides you with precision and long press life unobtainable from any other "C" frame or post type construction.

It's shipped ready for fast set-up

You get your Power Cylinder completely assembled with all ordered options. The Frame Plates, Base, Keys and Frame Bolts are packed un-assembled.

Only a few minutes are required for you to bolt the keyed components together.



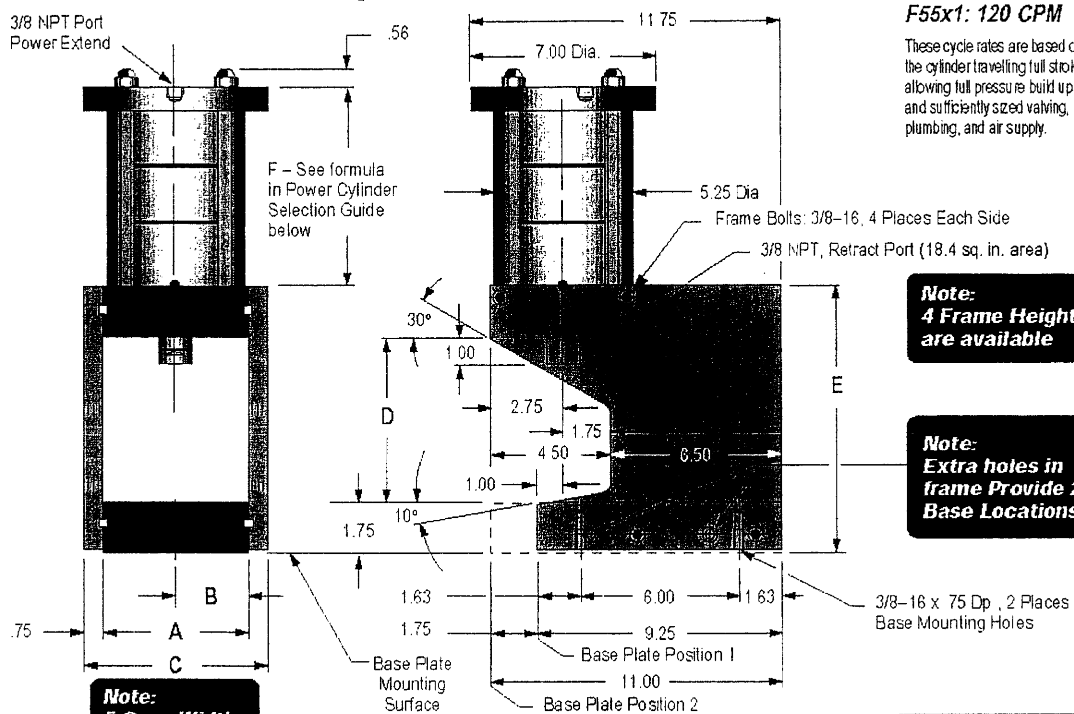
Multi-Power® Air Presses

Provisions for operator protection are always the full responsibility of the user

Multi-Power® Press Configurations

SPEED
F38x1: 140 CPM
F55x1: 120 CPM

These cycle rates are based on the cylinder travelling full stroke, allowing full pressure build up, and sufficiently sized valving, plumbing, and air supply.



Note:
4 Frame Heights are available

Note:
Extra holes in frame Provide 2 Base Locations

Note:
5 Open Widths are available

Dimensional Data

Open Widths Available

DIM	5-1/2	7	9	10	11
A	5.50	7.00	9.00	10.00	11.00
B	2.75	3.50	4.50	5.00	5.50
C	7.00	8.50	10.50	11.50	12.50

Frame Plate Height

DIM	10	14	16	18
D	6.31	10.31	12.31	14.31
E	10.00	14.00	16.00	18.00

Power Cyl. Weights (lbs) for 5-1/2" Open Width

Power Cyl. No.	0 Stroke Weight	+ Weight Per inch
F19	14.4	0.9
F38	16.5	1.5
F55	18.6	2.1
F72	20.7	2.7
F93	22.8	3.3

For other Open Widths
if 7", add 1.7 lbs.
if 9", add 3.9 lbs.
if 10", add 5.0 lbs.
if 11", add 6.1 lbs.

Weights (lbs)

Frame, Keys & Bolts	
10"	13.6
14"	18.0
16"	20.0
18"	22.0
Bases	
5-1/2"	8.6
7"	11.0
9"	14.1
10"	15.7
11"	17.2

Power Cylinder Selection Guide & "F" Dimension Formula

	F19	F38	F55	F72	F93
Formulas for "F" Dimension	Stroke + 2.22	2 x Stroke + 3.35	3 x Stroke + 4.47	4 x Stroke + 5.60	5 x Stroke + 6.72
Total effective piston area †	19.6 sq in	38.0 sq in	56.4 sq in	74.8 sq in	93.2 sq in
Maximum operating Pressure	150 psi	150 psi	150 psi	150 psi * 110 psi ‡	120 psi * 90 psi ‡
Force @ max. oper. pressure	2940 lb	5700 lb	8460 lb	11,220 lb * 8,228 lb ‡	11,184 lb * 8388 lb ‡

Force output = effective piston area x operating pressure
Force for retracting ram = 18.4 sq in x operating pressure

† For –AS Option, area is reduced by 1.2 Sq. in.

*For open widths: 5-1/2" & 7"
‡ For open widths 9", 10", 11"

Specifications subject to change without notice or incurring obligations

Appendix C

SPECIFICATIONS FOR OTHER COMPONENTS

C.1 Air Source, Regulator, Pressure Gage, and Silencers

A particular compressed air source is not specified in the design of the prototyping press because the possibilities for air supply will vary with the location of installation. Air may be provided by a laboratory air system, an ample sized compressed air tank, or a dedicated compressor. The minimum performance specifications for the air source follow here. The source must be capable of delivering dry, filtered air at a minimum rate of 3.0 cfm at a minimum pressure of 120 psi. These requirements allow for losses in the system and ensure that the press will not be underpowered or underfed at any time during its anticipated operation. Typically the press will be run at 100 psi and consume less than 2.0 cfm.

The air source must have a regulator placed inline with it, capable of accepting the highest rated output pressure of the air source and regulating it down to a range between 5 and 150 psi. The system is not designed to handle pressures in excess of 150 psi and the operator of the prototyping press must ensure the air coming out of the regulator does not exceed this pressure.

The air delivery system must also include a pressure gage capable of indicating pressures in the 5-150 psi range. This gage may be included on the regulator if the regulator is visible to the press operator for data collection purposes. If this is not the case, a pressure gage should be installed downstream of the regulator where it is visible to the press operator.

A suggested regulator with pressure gage is available through McMaster-Carr, part number 4959K51 (page 855 of the online catalog available at www.mcmaster.com) [10].

The silencers should provide significant noise reduction from the pressurized exhaust release while not causing a significant back pressure. (N.B. 2 are required.)

A suggested air silencer is available through MSC-Direct, part number 65083040 (page 3969 of online catalog available at www.msc-direct.com) [11].

C.2 Shut-off Valve

The shut-off valve should be a 2-way, 2-position, solenoid actuated type, normally closed by mechanical means (e.g. spring return). It should be open to flow in either direction only when the solenoid is energized by a 24 VDC current.

A suggested shut-off valve is available through McMaster-Carr, part number 4635K823 (page 387 of online catalog).

C.3 Flow Control

The flow control valve should allow fine manual adjustment of flow rate from a full-open position to a full-closed position. Adjustment level should be easy to detect visually. The flow control valve must also allow for unrestricted flow in the reverse direction so as to allow rapid raising of the press piston when the directional control valve is reversed.

A suggested flow control is available through MSC-Direct, part number 09634924 (page 4021 of online catalog).

C.4 Air/Oil Tank

The air/oil tank must provide adequate volume to contain the oil displaced from the last stage of the cylinder as well as a small volume of air necessary to allow pneumatic actuation of the return flow of hydraulic oil. A volume of at least 250 cu. in. is recommended. Baffles should be installed inside the top and bottom of the air/oil tank to prevent pressurized air or oil entering the tank from mixing with the reserve fluid in the tank. Such mixing causes the hydraulic fluid to become more compressible (spongy) and decreases its performance.

A suggested model is available through MSC-Direct, part number 08391161 (page 4011 of online catalog).

C.5 Directional Control Valve

The directional control valve should be a manually controlled valve with a 4-way, 2-position, 5-port configuration. The valve should allow easy manual access via a toggle knob or switch. The valve should be manual return, remaining in a given position until physically switched by the operator.

A suggested model is available through McMaster-Carr, part number 2700K12 (page 870 of online catalog).

C.6 Displacement Sensor

The displacement sensor should be a linear potentiometer with a range of 4.0 inches, capable of accuracy to 0.1% of the full stroke.

A suggested sensor is available from Honeywell, part number SLF-04-N-6000-B-6-A (online catalog available at www.honeywell.com).