GENESIS OF A MACHINE

A PRODUCTION PAPER CUTTER

Thesis by

Robert Dewey dePencier

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ABSTRACT

The evolution of a new machine is described, from an idea conceived in 1960 to a prototype operative in September 1964. The emphasis is on the decisions, methods, criteria, and results incorporated in the machine rather than on analytical aspects of the design. The primary criterion guiding the design was to provide a machine more economical for users to own.

A description is given of the machine, PC 64, which is a new completely hydraulic 42 in. x 4 in. capacity production paper cutter of the guillotine type, weighing less than 3000 lbs. Introduction of the essentials of a guillotine cutter and analysis of a 1960 commercial machine are used to establish reference limits for the new machine. Results of cutting tests performed to determine the most advantageous cutting angle are included.

Important new features of PC 64 include a single shear straight-line motion knife drive powered by a single hydraulic cylinder, and a hydraulically driven and controlled backstop. Optional automatic control for the backstop is accomplished by means of notched program bars affording trimout cuts to .015 in. It is shown that for any cutting angle selected, the height, width, and length of PC 64 cannot be reduced further.

The machine is evaluated by comparison with commercial cutters marketed prior to and during the project. From the comparison it is concluded that features of PC 64 have commercial potential and that the primary criterion has been met.

Photographic materials on pages 11, 48, 49, 162, 163 and 181 are essential and will not reproduce clearly on Xerox copies. Photographic copies should be ordered.

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I. INTRODUCTION

Purpose of the Thesis

This thesis is based on a design project carried out in the Engineering Design Department of the California Institute of Technology between 1957 and 1964. The project was directed at the problem of the precision cutting of rectangular sheets of paper as required in printshops, stationers, and in paper production factories. By 1960 it had been decided that the guillotine cutters then in use for this purpose could be economically replaced, by means of the application of accumulated technology, with a new machine which would be more economical for printshops and other users to own. In 1960 the author was given the problem of making such a new machine a reality and on September 22, 1964 the completed prototype, PC 64, was operative and the author's share of the project was accepted as finished.

Material included in the thesis deals primarily with three aspects of the project: identification of problems found from study of previous designs and other problems which arose during the design of PC 64; approaches, methods, and decisions employed in the design of PC 64; and a critical presentation of the solution as offered by PC 64 and by a number of commercial machines of 1964. Emphasis is on those factors which are not generally assigned to a specific engineering discipline; in this way it is intended that features of more general interest may be isolated. Moreover, the analytical work usually equated with certain specialties, though widely used following the initial synthesis stages, generally was elementary in nature. Except for specific results of significance to later evaluation of the machine, analysis is excluded.

Some experiments were performed where knowledge sufficient for making decisions was lacking, and descriptions of these are included.

A primary aim of the thesis is to add to the literature the statement of an interdisciplinary engineering problem of significance, a new solution to that problem and its subproblems, and an account of the methods used to evolve and select that solution. By the recognition and description of the methods applied to such problems, efforts to advance knowledge of general methods may be aided. A secondary aim is to include sufficient material to facilitate the appraisal, maintenance, and improvement of this cutter PC 64 so that the project may eventually prove to be profitable.

Papercutting - The Problem

One application of production paper cutters considered in this work is in paper mills. Output from modern paper mills is in the form of continuous sheets wound on mandrels. These rolls are usually slit to a number of smaller widths as they are wound on mandrel-mounted hollow cores in a second operation. In this project the cutting of rectangular sheets is considered, and these are produced in another operation which laterally cuts the slit rolls at spaced intervals. These sheets of paper or light cardboard are generally oversized and are stacked in piles called "lifts". The lifts are then "jogged" so that at least two edges are roughly in line in preparation for cutting on the guillotine cutters which are used to reduce them accurately to smaller sheets. In this application, guillotine cutters are often modified to suit the particular requirements of large orders of similar stock.

The more general application of guillotine cutters, and the more demanding one, is to be found in printshops and similar activities. This is due primarily to the fact that a strip of waste as narrow as .015 in. often must be removed by means of "trimout" cuts from between multiple printed impressions. Moreover, as a result of the precision necessary, the edges of sheets must be jogged precisely so that all impressions on sheets in a lift will line up. In addition, because of the relatively small size of the orders, the varieties of stock and sizes that must be cut on a single machine, and the lack of maintenance staff, a precision general purpose machine of high dependability is required. It was the more general and demanding problems of printshops that were considered in this work, for it was concluded that a design which satisfied these could be adapted easily to the less demanding needs of a particular paper mill.

The geometry of the paper cutting problem for which guillotine cutters are a solution is shown in Figure (a).

Planes 1, 2, 3 and 4 form two corners of a cube. The distance between the plane of cut (1) and the back reference plane (3) is adjustable to determine the size of parallel cuts. Square cuts are made by placing the jogged edge of a lift against the side reference plane (4). Plane (2) is the support plane.

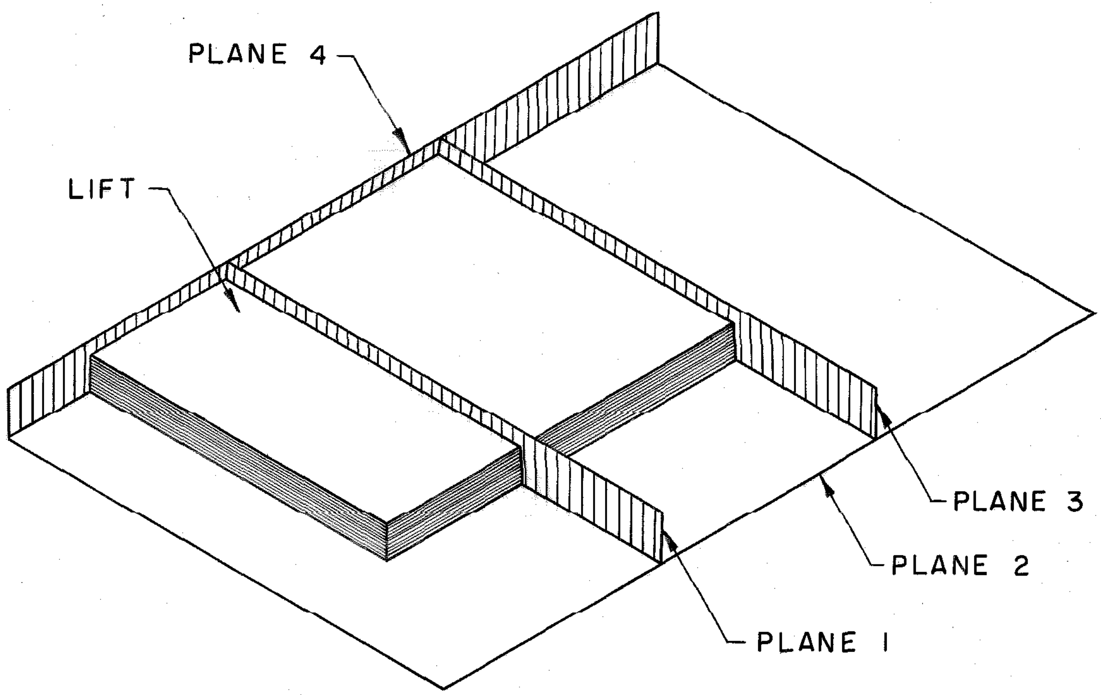


Figure (a) - The General Problem

Prior to a cut the lift is set up as shown in the figure, so that in the case of a trimming cut, the distance between planes (1) and (3) represents the portion to be retained, and for usual operation the length of lift in front of plane 1 represents the critical dimension. Some cutting device then separates the lift along plane (1), and the resultant strips are stored until all of the parallel cuts are complete. Usually it is necessary to rotate these strips through 90° and repeat the procedure to a new set of dimensions in the other direction.

When rectangular sheets are to be obtained in this way, i. e. -- by reducing the size of larger rectangular lifts, the following are required:

1. a means to make cuts in the plane of cut
2. a rigid support plane
3. a rigid back reference plane
4. a rigid side reference plane
5. a means to change distance between the plane of cut and the back reference plane.

These five points are taken as fundamental in the work to follow. They were considered as the minimum requirements of the problem. It should be noted that they are essentially independent of their means of accomplishment.

The only successful means of accomplishing these five points, so far as this work has determined, is represented by various designs of the guillotine paper cutter. There are certain features that determine the nature of these machines. In this work, these were envisioned as the components of figure (b), and were regarded as the necessary components of a guillotine cutter. Following this idea they are referred to as the components of a minimum guillotine cutter.

From this figure it can be seen that four of the five fundamentals of the problem are immediately satisfied:

1. KNIFE - a means to make cuts in the plane of cut
2. TABLE - a rigid support plane
3. BACKSTOP - a rigid back reference plane
4. FENCE - a rigid side reference plane

Not shown, but characteristic of all guillotine cutters is the fifth:

5. BACKSTOP DRIVE - a means to change distance between the plane of cut and the back reference plane.

In addition, due to the selection of the knife (1) and the resultant cutting forces, it has been found necessary to include:

6. CLAMP - a means to allow the cutting action without shifting the stock
7. BASE - a means to support the reaction forces of the cutting action selected.

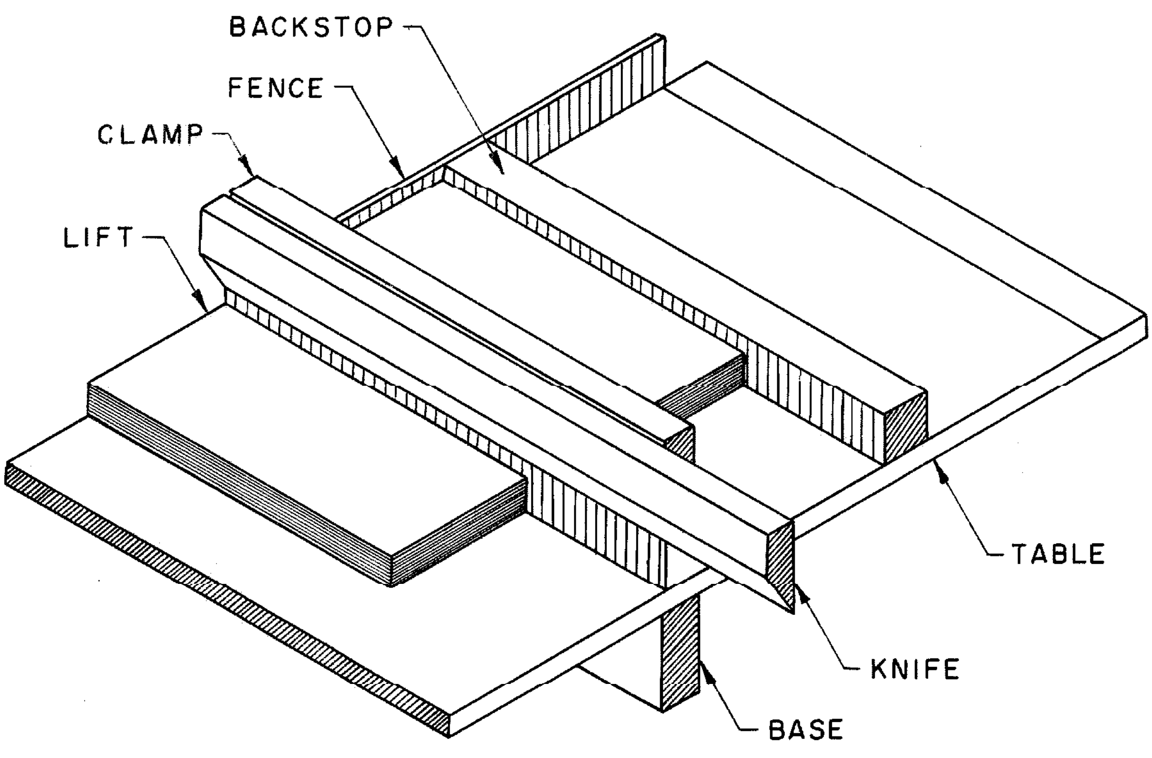


Fig. (b). The minimum guillotine cutter.

It is noted immediately that the need for (6) and (7) is questionable in general, as they are dependent upon the particular selection of (1). Nevertheless, if the nature of the solution is a guillotine cutter it appears necessary that these seven components be included. No new cutting action was found to eliminate the need for a strong clamp and base.

A new improved guillotine cutter was designed, for which the minimum cutter served as a set of necessary components. The design effort was devoted to the reduction of the requirements of these necessary components and to the addition of sufficient extra components to just satisfy the requirements of the main problem. Extra subproblems of printships that could be solved with little added expense were considered also.

By reference to figure (b), the general operation of a guillotine cutter can be described. In the idle position the knife and clamp are up and the backstop is either against a set limit stop and is ready to advance to a selected dimension, or it is locked at a selected position. In either case, the lift is slid onto the table and jogged against either the fence or the backstop or both, depending on whether a parallel or squaring cut is to be made, and whether the original lift is square. The backstop plane usually consists of edges of vertical fins or fingers which enter slots in the shoe of the clamp if the backstop is to be advanced for cuts narrower than the width of the clamp. In addition, if the operator is trimming books or similar objects, the segmented backstop feature may be used. This feature allows segments of the backstop to be adjusted so that different parallel cut dimensions can be established simultaneously. Once the lift is on the reference planes, unless the backstop is fixed for repetitive cuts it will be advanced by means of either a powered or manual backstop drive to a new position. The lift may be of any thickness from that of a single sheet to the capacity of the machine, represented by the clamp opening height.

Advance of the backstop may be accomplished by manual powered mechanical means, by manual control of a powered backstop, or by automatic control of a powered backstop. The backstop is advanced until it is stopped by a signal. In manual modes, the operator initially determines the position by observing the dimension between the backstop and the cutting plane on a readout device, or by lowering the clamp near the surface of the lift by means of a clamp drop device and then sighting along the front edge of the clamp shoe. If a sequence of cuts initially located in this way is to be repeated, either mechanical stops or signals on a programming device are established. These may serve as references either for automatic or for manual positioning. For trimout cuts it is necessary to space these signals as close as .015 in., but in programmed operation no two trimout cuts are adjacent. Following advance the backstop is locked in position prior to the cut and the rear reference edge of the lift is against it.

Two hand levers or buttons which when at least one is released will stop or reverse the motion of the knife and clamp are used to start the cycle. The cycle consists first of the descent of the clamp until it strikes the top of the lift and clamp force is applied. The shoe or bottom of the clamp is usually some combination of slots, segments, and flexible portions so that uneven stock will be clamped with approximately uniform pressure. Instead of using the cycle both to lower the clamp and to apply the pressure, the clamp can first be slowly lowered with the clamp drop mechanism to avoid shifts of the stock due to impact.

The knife drive then causes descent of the knife. The paper is totally cut after the knife enters a cutting stick of wood or plastic situated flush with the table beneath the line of cut. The knife consists of a bar holding the blade which can be removed for sharpening. The need to cut the last sheet against the cutting stick requires that the knife edge be precisely located at the bottom of each stroke. This is usually accomplished by means of the over-center action of crank-connecting rod linkages. Following penetration of the last sheet, either the knife goes up, or it goes up after the operator removes at least one of his hands from the interlocked cycle controls. To avoid riffling the stock the clamp applies some pressure until the knife is up, and then the clamp ascends. If the machine is in the automatic mode then at this stage the backstop advances to a new position. If not in the automatic mode the operator positions the stock for the next cut.

Apart from these external or useful functions given in this description of the general operation of guillotine cutters, there are many internal features which are determined in part by the means selected to accomplish the external functions. It will be seen in later work and in the photographs of commercial cutters that the various means selected have resulted in a wide variety of internal features.

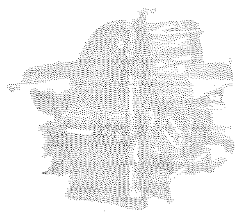
Examples of the progenitors of current guillotine cutters are included as Figure 1. All of these were offered for sale in 1960 and the three noted as "available" were available from the manufacturers as late as 1962.

Because the knife and knife drive have been noted as critical portions of guillotine cutters the machines of Figure 1 have been classified as "guided linkage" type and "frame" type machines according to the knife systems used. All of the models shown are characterized by electric motor-flywheel-clutch-crank-connecting rod-linkage drives as can be seen in Figure 1.

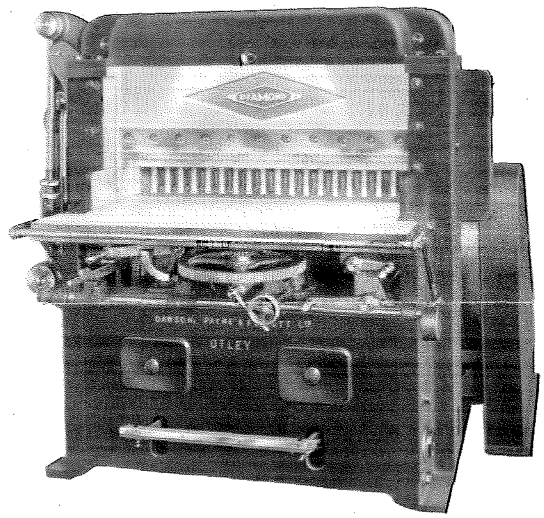
In the guided linkage types of machines, the crank throws at each end of the knife bar are unequal. Sufficient universal joints are included in the drive linkage so that a point on the bar can be guided to describe a definite path in the plane of the cut. Types of guiding employed are indicated by the angled corner of the knife bar of Motoya, the cast angled ramp of C and P, and the slot of Dexter, all situated at the left side of the machine. Due to the unequal lengths of the crank linkages used, the motions of all points of the knife are not identical to that of the guided point. A second motion, an angular velocity about the guided point, is imposed on the bar. Brochures describe the resultant motion of the knife as being advantageous and term it "double-shear".

The Seybold and Otley machines shown employ the "frame-

(a) = available

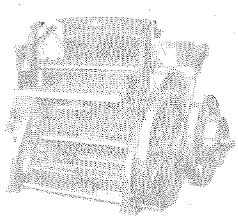


SEYBOLD

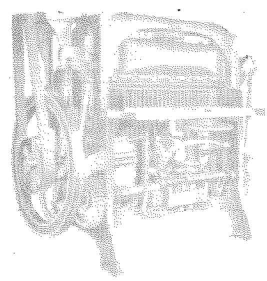


OTLEY (a)

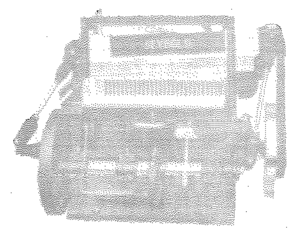
FRAME TYPE



DEXTER



SHERIDAN



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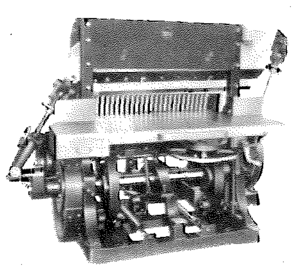
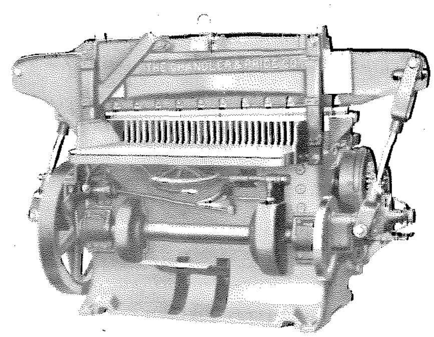


MOTOYA (a)



CHANDLER & PRICE (a)

GUIDED LINKAGE TYPE

Fig. 1. Progenitors of Current Machines

type" system for the knife drive. Two points of the knife bar are guided in parallel paths and the "double-shear" feature no longer appears. In one example of this general type, a four sided frame contains the clamp bar and the guide slots for the knife bar. Cam rolls in the knife bar engage the slots of the frame. In use, the clamp frame and guide slots descend until the clamp contacts the surface of the paper. Another frame then applies a vertical motion to the knife bar while allowing it to move sideways. This causes the knife bar to descend through the paper at the angle dictated by the guide slots.

The complexity, size and main features of these machines are clearly shown in the figure. An idea of scale is obtained if it is noted that the height of the tables is approximately 36-40 inches.

The decision to proceed with this project was based on technical knowledge, a scrutiny of machines such as those shown in Figure 1, and on a comparison of those machines with the fundamental features of the minimum guillotine cutter as shown in Figure (b). Figure (b) was regarded as representing the necessary components for any guillotine cutter and Figure 1 as representing the sum of the necessary plus the sufficient components for various operative commercial cutters. It was concluded that the sufficient components of these commercial cutters were excessive and furthermore were so varied as to indicate that no single solution had yet been accepted as best.

The general approach to the problem and subproblems was to determine the minimum problem statement, to advance a minimum solution, and then to converge slowly on the final solution by making additions which solved primary features of the basic, without introducing significant new, problems. Occasionally, the nature of some solutions offered additional features at little extra cost. These led to reexamination and sometimes to modifications of the initial statement of the problem. This was justified by the fact that all actions and decisions were guided by the rule that the overall result must be "more economical for printshops to own". The thesis includes a description of how this general approach was applied.

Content of the Thesis

The thesis is divided into several sections presented in the order that the work was performed. In the absence of direct experience with the papercutting and printshop industries, the predesign work described in section II served to clarify the problem. Former students had considered some subproblems and suggested a number of possible solutions. These were reviewed and showed the need for further studies and data related to the actual cutting action. From the students' work it was concluded that the machine should be completely hydraulic, and that cutting tests would be necessary. Resultant tests and studies are described and some criteria for further work presented. One of these which was used to establish the cutting angle as 21^{o}, includes a fundamental assumption incorporated in the design. This assumption was that the resultant cutting force lies along the direction of blade travel, or that by modification of the blade profile, this could be accomplished. The minimum guillotine cutter presented in the introduction established a lower limit for both the problem and the solution, and at this stage an upper limit for the solution and a more detailed problem statement was established by the description and analysis of a common 1960 machine. In this way the work described in section II established guide lines and limits for the future design.

From a number of alternative means considered by this time, section III describes some of the features considered advantageous for inclusion in the final design. These features were based on rough layouts, analysis, and a consideration of the interactions of various proposals. The work of section III established the essential subassemblies, and general features of the future machine.

The methods used in the final design, the decisions made, and the criteria on which they were based are represented by the assembly drawings and description of the prototype, as presented in section IV. Consideration of the features described should be based on the observation that two distinct sets of functions are represented: those necessary to the operation of a guillotine paper cutter, represented by external features of various subassemblies; and those found necessary for operation of the subassemblies chosen. The design presented is aimed at accomplishment of the former with the introduction of as few of the latter as possible. In the main the subportions described are closely related to the minimum components as presented in figure (b), and these can be recognized. The results of graphical calculations of cutting performance are included in this section, primarily to provide data so that operation of the machine will indicate if the assumption related to the cutting angle is correct. If it is seriously in error, several simple changes to the machine are available.

A photograph of the resultant machine PC 64 is included for comparison with the minimum machine of figure (b), the progenitors of figure 1, the benchmark machine of figure 6, and other machines of 1964 shown later in the report. Results of comparisons with these other machines have been encouraging.

Section V describes difficulties encountered when the machine was demonstrated for the first time. It is noted that all of these difficulties might have been foreseen and did not concern the basic concepts included in the design. The primary problems were shock from the forces involved at the bottom of the knife stroke, and complications in the backstop hydraulic circuit.

In Section VI an evaluation of the machine has been included. It has been noted that the author's share of the project was completed when the machine first cut paper. For this reason, most of the evaluation is based on design experience and estimates of performance and improvement, not on extensive performance tests of the machine. Performance has indicated, however, that the assumption concerning cutting angle is not grossly in error, and that the main problem to be considered is the effect of shock, related to hydraulic pressures and geometry during reversal of the knife at the bottom of the stroke.

Twelve commercial machines of 1964 are described by means of overall photographs and photographs of specific details of interest, as well as by a table of data available from brochures. In this way section VI simplifies the assessment of this machine. From the evaluation it was concluded that no significant features had been overlooked, though some could definitely be improved. The improvements offered are generally related to refinements of specific details, not to major changes.

Finally, section VII advances several conclusions concerning the machine and the methods used to design the machine. Fundamental of these was that PC 64 with several small changes could accomplish the purpose for which it was intended, i. e., it could economically be supplied to printshops as a machine more economical to own than the cutter of 1960. The same conclusion was reached with respect to the competitive machines of 1964.

II. PREDESIGN WORK

2.1 A First Scrutiny of Existing Machines.

Between 1958 and 1960 a number of design students at the California Institute of Technology examined existing paper cutters and advanced proposals for an improved machine. The major components were studied independently. These are:

Knife - blade, bar, drive.

Clamp - bar, drive.

Backstop - bar, drive, control, readout.

Table

Control Circuits: Hydraulic
Electrical.

It was assumed that primary power supply would be electrical, but no restrictions were placed on the means of force and power transmission within the machine.

These detailed studies indicated further consideration of some component configurations as well as the immediate rejection of others. While reviewing this work it became clear that certain recurrent assumptions required substantiation from the technical literature or from tests. In particular, it would be necessary to determine the optimum cutting angle.

In 1960 the author was assigned the problem with the stipulation that a commercial model of the machine should result. Economy of construction was a prime consideration, and was to be based on an expected immediate production of 100 cutters per year. No rigorous plan of attack was used, but the pattern will emerge in the description of the work.

There are two points of view from which a problem of this sort may be approached. In one case, the problem is: "Given these parts, how, with current technology, can they be produced most economically?" This is the role of the technician. In the second case the question is: "Can current technology offer a solution to the minimum statement of the problem? If so, what components and technology will yield the most economical solution?" This is a primary role of the design engineer. In this study, the emphasis is on the design problem.

2.2 A Review of Former Work

From examination of the work done by former students it was found that opinion was in favor of making the machine all-hydraulic. Some existing machines had a hydraulic clamp, successfully introduced to the industry by Lawson in 1947. No other large component substitutions had evolved in the former work although changes in the power linkages were suggested.

The following specific recommendations had been recorded. Following review of these, attention will be turned to the fundamental features involved in the cutting of paper.

General

Duplication of the existing functions of present machines was the minimum acceptable scope of the new design. Furthermore, the wide use of hydraulic power including preservation of the hydraulic clamp feature was considered to be advantageous.

Ideas for the desired modes of operation were included, mainly centering around the fundamental two: manual and automatic. The advantage of supplying in addition a "clamp only" or a "knife only" mode was noted. The former would be for punch press use and the latter would have common application for creasing cardboard.

Safety was a problem not solved on existing cutters, but no important suggestions were offered. It was noted that standard precautions such as interlocks in the relay circuits should be included so that both of the operator's hands were in use to initiate and continue the cycle in the manual mode of operation. The presence of a "window" to the right of the stock and below the blade was noted. In spite of many alternatives considered, no solution covering that danger region was offered.

Knife and Clamp

Some layouts of the castings and core shapes that could be used for a knife or clamp bar had been made. Each of these showed curved longitudinal vertical section for these members, when bending strength was allowed to dictate their external profile.

The importance of the linkage used for the knife bar had justified further investigation of the motion and geometry involved. Extensive examination had yielded a possible knife drive in the form of a 4-bar linkage. This attempt had been aimed at having similar motion at both ends of the bar, a small shear angle, and a provision for precise bottoming.

Throughout some of this effort, it had been assumed that power could be transmitted to the knife linkage by means of a rotary hydraulic actuator. This offered a compact motion that could be used to provide a precise bottoming location by means of the over-center action of a crank.

The knife blade itself should be lightened, as the current 100 lb unit was expensive and dangerous to handle, but no acceptable details had been developed.

Various types of clamping mechanisms had been described. These included a simple torque tube arrangement, toggle types, unwinding of roller chains from sprockets, and two two-bar linkages joined to the clamp at the top and to the base at the bottom. The primary problem for the clamp was to guarantee similar deflections at the two ends.

Backstop

1. Drive

The T square type of backstop angular control as used on other cutters, was not desirable, as it required a large extension in the rear of the table which increased the overall length of the machine. This had led to a recommendation of a backstop squared, as well as driven, from the underside of the table. The squaring was to be accomplished by means of steel tapes running on four pulleys set in cantilevered dual ball bearings mounted on the underside of the table casting. Runout or eccentricity of the pulleys was to be controlled as it would affect the geometry. The steel tapes were to be wrapped in bandsaw fashion, one around the two left-side pulleys, and one around those on the right. The size of the pulleys was such as to allow connecting these tapes to each other at a single point under the center line of the table. Subject mainly to the spring constants of the bands, two points at the edges of the table would move in similar directions, a distance equal to the travel of the common connections. These two points were then to be connected to the backstop beam itself by means of bridge blocks held at right angles to the plane of the table by teflon pads. An inherent feature was the necessary extension beyond the backstop front edge of at least one radius of the pulleys used. No simple way was seen of reducing the diameters of these horizontal pulleys to less than one half the width of any table.

It was evident that something better than a simple hydraulic cylinder must be used to drive the backstop. The inherent requirement of a simple cylinder that its extended length is much in excess of the stroke would require a machine length more than twice the paper length. This had led to the idea of a "doubler", to produce a useful motion equal to twice the useful stroke of the hydraulic cylinder. There was the possibility of using a tripler or other multiplier to reduce the stroke length required even more.

2. Control

A backstop speed and position control was to be incorporated. Naturally, in this machine a hydraulic control was to be used. This was to be a valve mounted on the backstop drive which decelerated the backstop as a desired position was approached. Because this would not guarantee a stop at the given location, with acceptable error, a change was added to this classic hydraulic deceleration valve. At a predetermined distance from a required location (say within .002 inch), the flow to the cylinder was to be diverted to exhaust. This would guarantee the functions required, and in addition, would define a maximum leakage rate that could be tolerated. Seven or eight layouts for valves which included these two features had been preserved.

Manual

For manual control, a lever near the operator would allow him to change the length of a cable. This cable was wrapped around two pulleys mounted on an arm moved by changes of the cable length. The motion of the arm was transferred to the valve bobbin. Readout of position could be done with a steel tape, spring-loaded to correct for sag.

Automatic

For automatic control a seeker, mounted on the end of a lever which transferred motion to the valve bobbin, could engage a notch. As the backstop advanced and transported the valve body with it, the seeker was stopped by a notch and subsequent motion would be controlled by the resultant motion of the valve. The backstop would then stop at a location fixed with respect to that notch.

Notched Bar Programs

It had been suggested at the same time that the notches for programmed stops could be made on steel bars or strips, loaded radially on edge in an assembly rotatable from the front of the machine. For repetitive cutting schedules these programs could make the location of the backstop an automatic function. The seeker could be made to rise automatically from one notch and look for the next after completion of a cutting cycle. Each strip could be supported independently, and should be spring loaded to eliminate slack errors, without requiring precise manufacturing tolerances. Several layouts describing ways of mounting approximately 12 of these "capstan bars" had been prepared.

Trimout Cuts

Trimout cuts, the removal of the small strip of waste between printed impressions, had been considered. A difficulty associated with spacing of notches was seen when it was recognized that the minimum spacing of two adjacent similar notches would be limited by their shape, whereas the size of needed trimout cuts is limited only by the minimum cutting dimension of about 1/64 inch. Of use in the solution might be the additional fact that trimout cuts are always separated by a cut of relatively large dimensions.

Alternating Program Selector

In cutting, say 8 - 1/2 in. by 11 in. sheets from large lifts, the most efficient practice would include cutting it one way completely into 11 in. strips, i.e. using the 11 in. program, then rotating these strips and cutting the other way according to the program for the 8 - 1/2 in. width. If this cycle were to be repeated, a program "alternator" could be provided. An operator would then be able to expect the correct one of two alternate programs if he stuck to this cutting sequence. Such an alternator had been suggested, and the design of a mechanical alternator that would rotate the whole program capstan after each cycle had been explored briefly.

2.3 Results of Paper Cutting Studies

First scrutinies of existing paper cutters were based on little knowledge of the actual cutting actions and forces involved. However, it was observed that not only were factors such as massive size, deflections, acceleration forces, and forces and velocity of cut interdependent, but also that these factors had dictated many of the criticized features of the machines. It was concluded that none of these features could be rationally redesigned until two subjects had been studied: factors affecting cutting action, and forces and energies of cut, for various assumed parameters.

Prior to experimental work, some observations and limits were established: it was first noted that paper, a fibrous material, is cut by the application of energy to a small area. Though various means could be investigated, some of which could be particularly applicable to cutting both curves and straight lines, this investigation was limited to knife blades. Current applications can be considered in terms of two types of blades; those curved in the plane of cut, and those straight in the plane of cut.

Curved-blade devices include lever-action cutting boards, slitter rolls, and most scissors. In all cases cutting forces are small, the cut is made at a single progressive point, and the angle between the opposing blades is approximately constant at the point of cut. For most effective cutting with these devices the blades are ground with small "teeth" on the edges. The thickness of material that can be cut is limited by deformation of the stock and by the dictates of continuity where the stock is only partially cut.

Straight blades have been applied so far as is known only to guillotine cutters, the fundamental features of which are presented in the introduction. It was observed that relative to the plane of the paper, the edge of the blade always had some horizontal as well as vertical motion. These led to the terms "sawing" and "chopping" forces respectively and to the possibility that the forces of sawing could be varied by superimposing small teeth on the edge of the blades. As opposed to the case of cutting at a point, it was expected that due to relative motion between teeth and stock, these teeth would have to be sharp on all edges to avoid material removal, i. e., "sawdust". This, coupled with the fact that vertical forces were the dominant reason for the 13, 000 lb weight of the current machine, led to the idea of an almost total elimination of the chopping forces. It seemed possible that a horizontal band-type machine could be built with the described toothed edge such that at high operating speeds not only the chopping but also the sawing forces could be reduced to negligible size. The main difficulties would be to restrain the blade laterally and to prevent side motion of the cut portions.

Based on these initial considerations, it was decided that both the proposed band-saw and the proven guillotine action should be investigated further. The work was approached with the idea of obtaining a sound qualitative idea of the actions involved. Only approximate quantitative values were sought because it would be impossible in the time available to determine cutting values for all the varieties of paper which the machine would be required to cut. Instead, after several checks of other varieties, Ditto paper was selected as sufficiently representative to be used for the prototype machine.

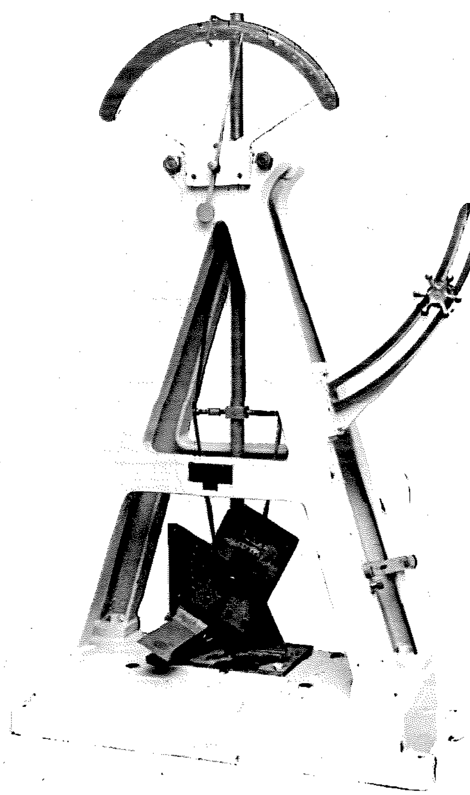
With these considerations in mind, band-cutting and guillotine cutting were studied.

A. Band Cutting (Band Knife Test Machine)

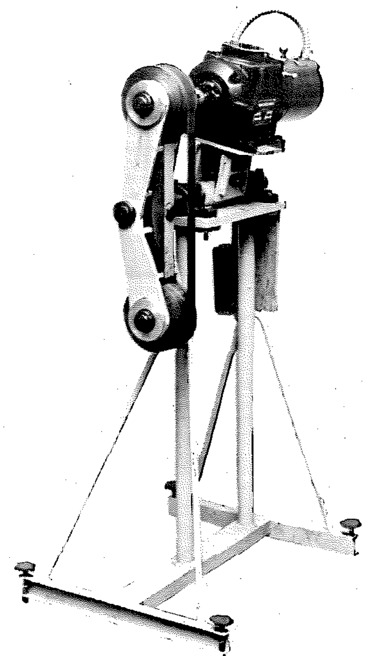
This investigation was a preliminary examination of the energies, speeds, and qualities of cut obtainable by band cutting. The fundamental difference between this type of cutting and common band sawing results from the blade edge being parallel to the surface of the thin lamina to be cut: as long as the lamina is free to move once cut, no material removal, hence no "set" of the teeth is required or desired.

To obtain an initial idea of the utility and requirements of the method, the edge of the blade of an 18-inch band cut-off saw was ground to a knifelike cross-section. A small fixture to hold ream-sized samples was clamped in the vise of the machine and test cuts were made. The speed of the blade and the cutting speed were slow, but the resultant cuts were exceptionally smooth.

Due to this encouraging result, and the availability of a variable speed drive unit, a band cutting machine was built as shown in Figure 2 according to the design of Figure 3. Provision for wide bands, variable high rigging tension, easy blade removal, variable band lengths, variable speeds, speed measurement, torsional stiff-



DROP KNIFE



BAND KNIFE

Fig. 2. Test Machines

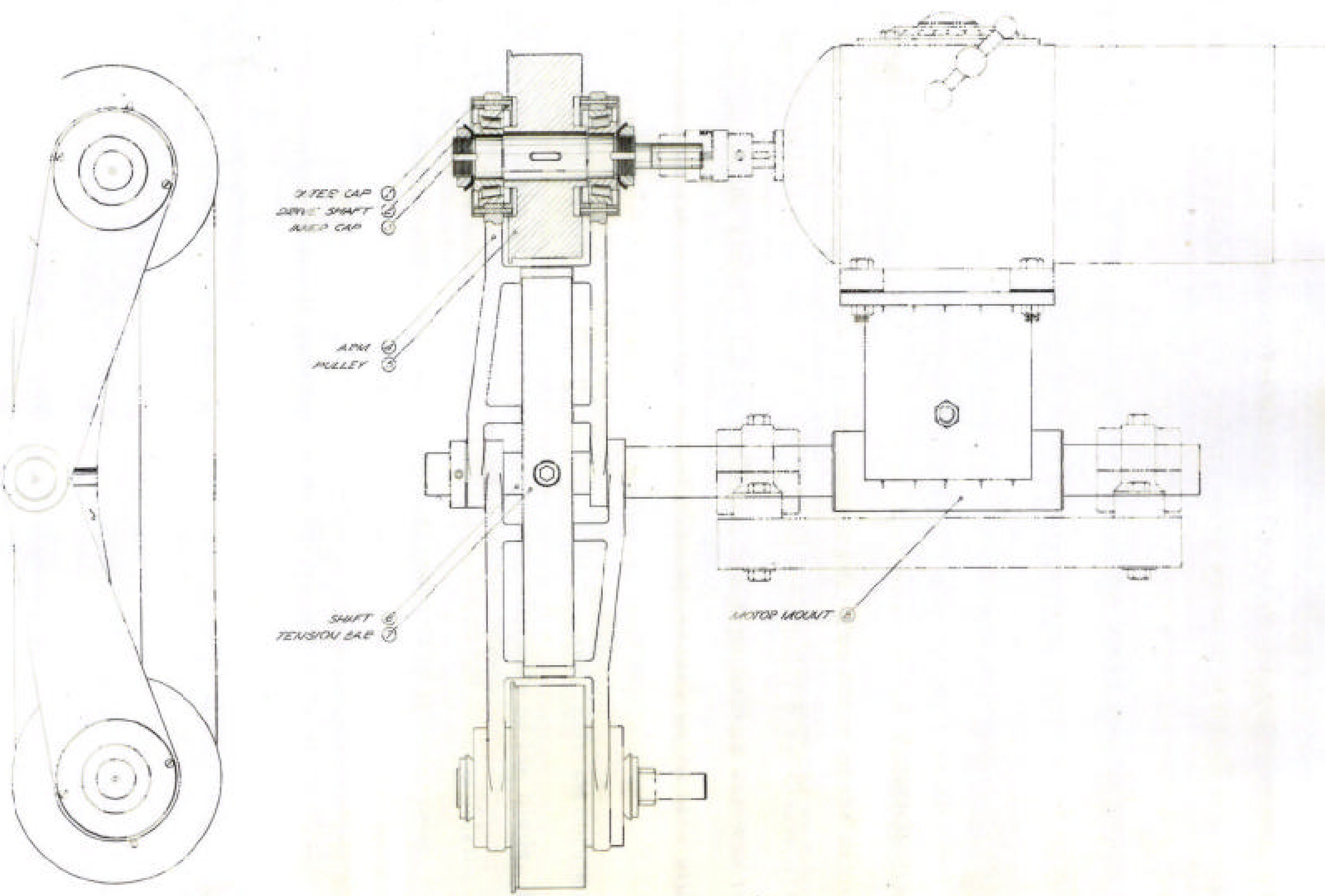


Fig. 3. Band Knife Test Machine Assembly - BK

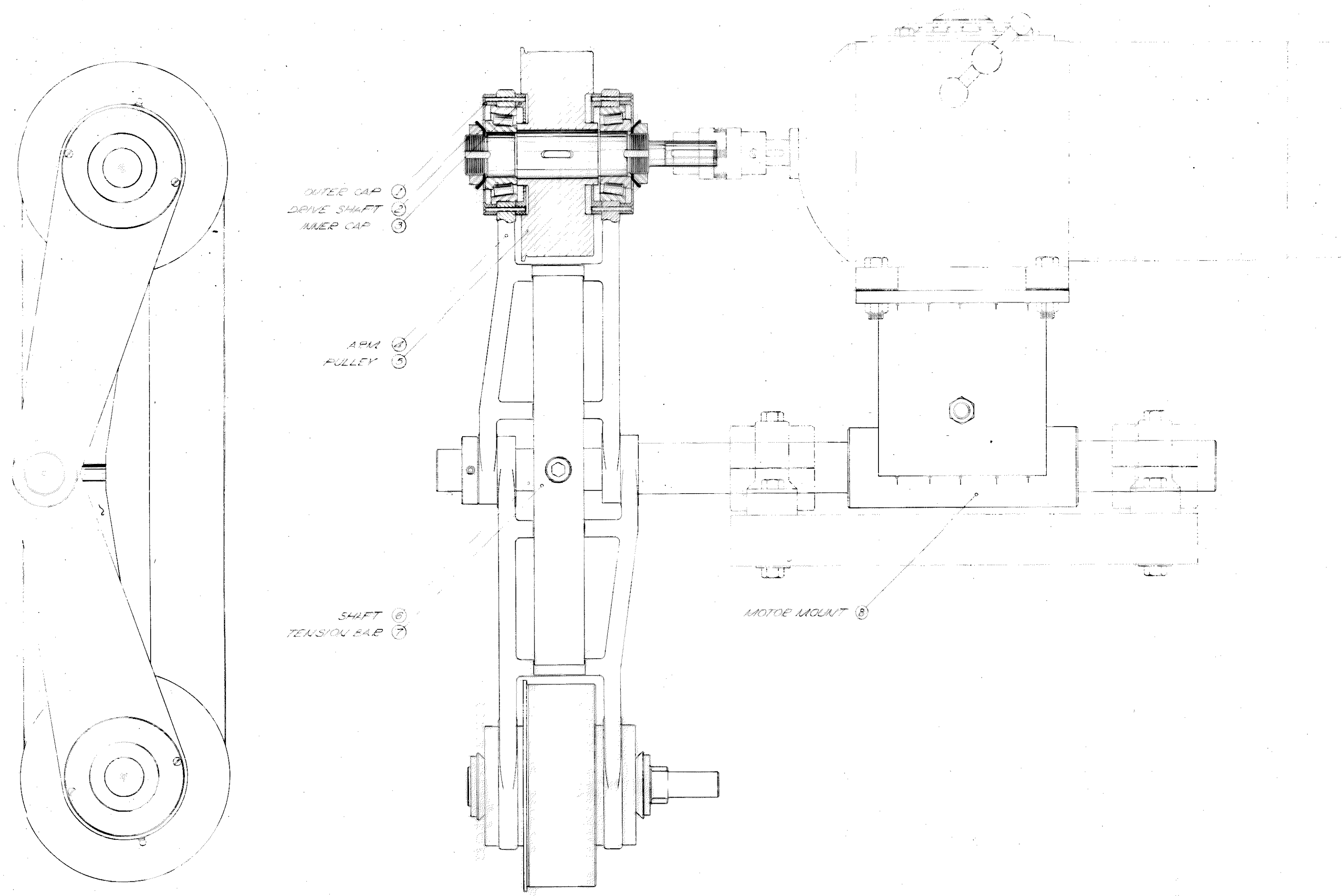


Fig. 3. Band Knife Test Machine Assembly - BK

ness of the assembly, extra side stiffness of the blade, stock-holding fixture, and force measurement, were some of the primary requirements of the design. Measurements of force and speed were to be used for energy determination. Except for extra side stiffness, stock-holding fixture, and force measurement, these were accomplished in the design shown in Figure 3.

Figure 3 is generally self-explanatory. By applying strain gages to the shank of the screw at the center of the TENSION BAR (7), the tension in the blade can be measured. The force of the cut could be obtained from the moment in a long externally hinged arm used to support the stock. Later a slotted vee-type backup bar was to be mounted on the tension bar to apply the necessary extra side stiffness to the blade.

In tests without the backup bar, cuts were again smooth but because of the lack of torsional and side stiffness of the blade, square cuts were not obtained. No adequate solution for this problem, or that of motion of the cut stock was obtained, and it seemed that much development would be needed before the idea could be included with confidence in the new machine. At the same time, tests on the guillotine type of cutting, given below, were extrapolated to show that sawing required much more energy per cut than did the combination sawing and chopping used in the guillotine action. For these reasons, further work on the band-cutting method was stopped.

B. Guillotine Cutting (Drop Knife Test Machine)

The investigation of guillotine cutting action began with a scrutiny of existing machines, as described in the introduction. Based on the information acquired, figure (a) shows a typical front view of the guillotine action, looking toward the plane of cut, and figure (b) shows a typical section perpendicular to the plane of cut. From these, significant parameters were defined and their effects noted:

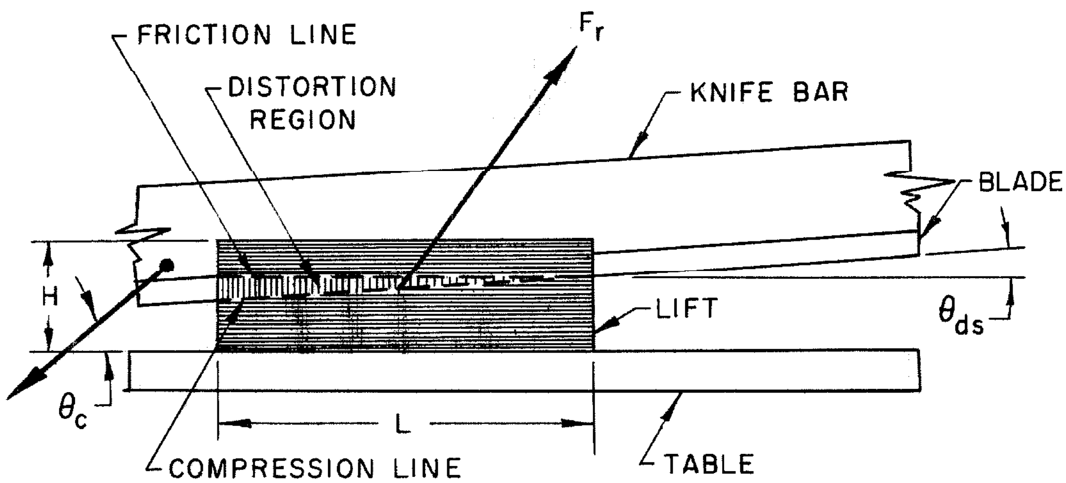


Fig. (a) - Front View of Guillotine Action

$\underline{\theta_c}$ - the angle of cut, represents the angle between the path of a controlled point near the left end of the knife bar, and the plane of the table. This angle was observed to be about 40°-60° and, depending on the design, may vary during the cut.

$\underline{\theta_{ds}}$ - the angle of double shear, is the angle between the edge of the blade and the plane of the table, and is claimed to be an advantage by those manufacturers who include it. Its value varies from a maximum of about 5° to zero for the last sheet cut.

<u>The Friction Line</u> represents the boundary between totally cut sheets, which only contribute frictional forces, and partially cut sheets.

<u>The Compression Line</u> represents the boundary line formed by the edge of the blade, between uncut and cut portions of sheets. The wedge shaped edge of the blade causes compression along this line.

<u>The Distortion Region</u> is bounded by the friction line and the compression line and includes all cut portions of partially cut sheets. As a result of the finite blade cross-section and the dictates of continuity, these portions are all distorted.

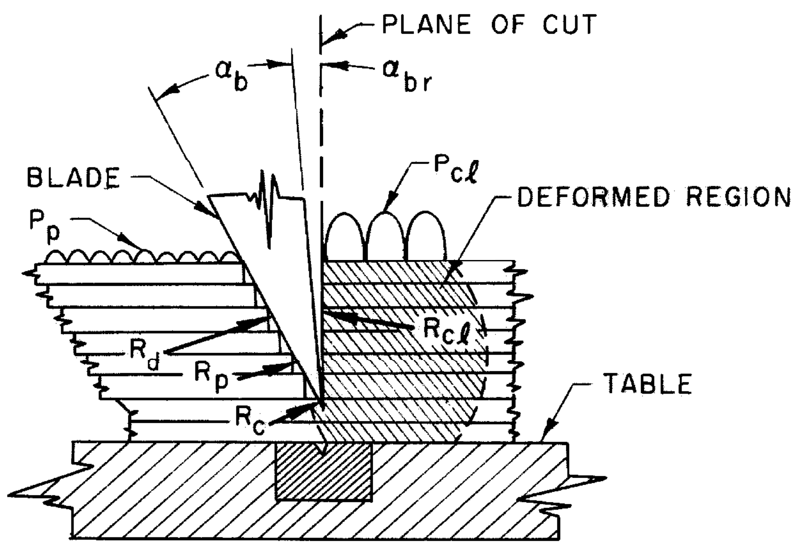


Fig. (b) - Vertical Section of Guillotine Action

<u>P_p</u> - the paper pressure on the cut portion of the lift is a result of the weight of the paper above. For the 4 in. height of stock considered, a maximum value of $\frac{1}{6}$ lbs per in^2 of stock was taken.

The maximum cut portion in front of the knife was assumed to be 24 in, so to serve as an upper limit for other calculations this pressure is equivalent to a maximum total force of 4 lbs per in. of blade length.

$\underline{P_{cl}}$ - the clamp pressure on the uncut portion of the lift is used to preserve the set portion of the stock, and to compress it prior to the applied knife forces. The machines are designed so that an approximately uniform clamp pressure is applied over a region extending from just behind the plane of cut to about 2-3 in. behind that plane. An assumed maximum value of 100 lb per in^2 of clamped surface or a total load of approximately 250 lbs per in. of cut length was taken.

$\underline{\alpha_b}$ - The included blade angle, is due to the need for blade strength and for moving the cut stock along the friction line. A value of this angle on commercial machines was observed to be about 30°.

$\underline{\alpha_{br}}$ - the blade back-rake or back-clearance angle, establishes clearance between the back plane of the blade and the plane of cut. On the cutters examined it was found to be zero.

<u>The Deformed Region</u> is due to clamp pressure. This region was assumed to extend below the clamp approximately as shown in Figure (b). The magnitude of the deformations would depend on the height of the cut portion, on the properties of the stock, and on any externally applied restraints.

$\underline{R_{cl}}$ - the reaction on the back plane of the knife, is caused by deformation of the clamp deformed region. The magnitude of R_{cl} varies with the deformed region and with the size of a_{br}.

$\underline{R_c}$ - the reaction on the blade edge, is due to compressive forces along the compression line. Along this line the blade edge stretches the uncut portion of a sheet by compressing the cut portion. This force varies with the blade angle and the sharpness of the blade, and with the thickness and elastic properties of the sheets cut. R_c was assumed to be at least an order of magnitude smaller than the compressive forces in the plane of cut.

$\underline{R_p}$ - the reaction on the front plane of the knife blade, is a result of the paper pressure on the cut portion of the lift. From the maximum assumed 4 lb total value of P_p per in. of blade length this force would be negligible. In the event that the blade included angle a_b exceeds the friction angle of the paper, or cutting speeds are exceptionally high, it must be considered, however.

R_d - the reaction on the blade front plane due to the distortion region. Although this reaction was not evaluated a consideration of the buckling of thick stock restrained by the weight of paper above, indicated its effect could be considerable.

The motive for determining these parameters affecting the cutting action is recalled: criteria are to be established to reduce the major forces involved in the cutting action. Referred to the cutting plane these are considered independently as either in a vertical section perpendicular to the plane of cut, or in the plane of cut.

Forces in a Vertical Section (Figure (b))

Of the forces in a vertical plane perpendicular to the knife blade, it was considered that three could be reduced without experiment, and decisions were made to affect this in the design. A fourth, due to the clamp pressure P_{cl}, would be studied in tests of the machine. The three considered were: R_c, the blade edge reaction; R_d, the paper distortion reaction; and R_{cl}, the clamped deformation reaction.

R_c - Blade Edge Reaction

The major result of this force is a transverse bending moment in the blade considered as a cantilever. This moment dictates the blade material strength and hardness, and the size of the included blade angle.

The magnitude of R_c can be reduced by a reduction of α_b, the blade angle (assuming the blade is sharp), and by the reduction of the vertical cutting forces as described below. Since reduction of α_b is compatible with reduction of R_p but results in a reduction of the cantilever beam strength, a compromise was made. It was decided that an included blade angle of 30^o, similar to other cutters, would be used to avoid extensive testing.

R_d - Paper Distortion Reaction

The major effect of this force is to cause binding and bowing of the knife bar. R_d tends to increase as the knife descends, and can cause the knife to stall. This force dictates the lateral bending stiffness of the knife bar and sometimes results in the location of a third bearing surface at the mid-point of the bar.

The magnitude of R_d could be reduced by reduction of the area of the distortion region, the thickness, width and rigidity of the material cut, and the weight of the material above the distortion region. Of these only the area of the distortion region can be changed freely. Since the angle θ_{ds} determines the distortion angle, and this angle of double shear is reduced to zero for the last sheet cut, it was concluded that θ_{ds} was non-zero for some reason other than an inherent requirement of guillotine cutting of paper. Shock reduction or a particular linkage design are possible reasons. To reduce R_d it was decided that a double-shear angle of zero would be used in the design.

R_{cl} - Clamped Deformation Reaction

The major effect of this force is identical to that of the paper distortion reaction R_d but in the opposite sense. It might be argued that the two could be made to cancel, but this is impossible since cutting conditions are not subject to the necessary control for such a procedure.

The reaction can be eliminated if the restraint of the back plane of the knife blade is removed. To do this, it is necessary to have a non-zero back-rake angle α_{br} based on some estimate of the deformation properties of the stock to be cut. It was decided that, to reduce R_{cl}, an angle of back-rake of 1° would be used.

It has been seen how criteria for reduction of forces normal to the plane of cut were established. It was recognized that due to friction and the effect of the wedge-shaped blade, these forces also would contribute components in the plane of cut. The calculations necessary to describe these effects were omitted for three reasons: the magnitudes of these forces, as shown by cross-sections of knife bars of existing cutters, are significantly smaller than those occurring in the plane of cut; the magnitudes are further reduced by the small coefficient of friction between the surfaces involved; and any tests to determine values for use in the design would necessarily yield the effects of all force components acting in the plane of the cut. It will now be shown how the problem of reduction of forces in the plane of cut was approached, and what assumptions, data, and criteria resulted.

Forces in the Plane of Cut

The motive for determination of the forces acting in the plane of the cut was based primarily on one assumption and one observation. It was assumed that all the forces acting in the plane of cut could be represented adequately by a single force F_r as shown in Fig. (a). The observation was that the single most dominant factor dictating the massive size of guillotine cutters was the size of the vertical component of that force. The resultant force configuration acting in the plane of cut is shown in Fig. (c), and for discussion, the resultant F_r has been resolved in two ways.

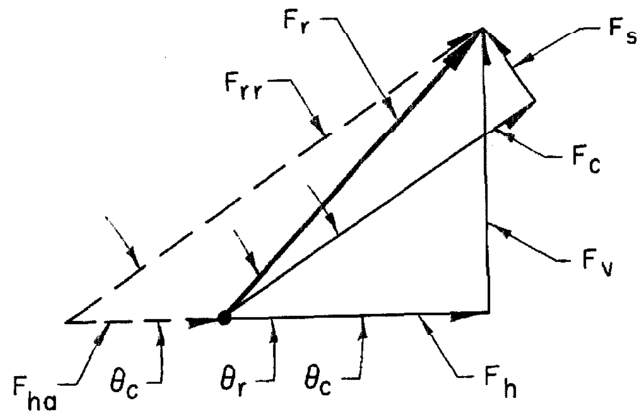


Fig. (c) - Actual Blade Forces

where

θ_r = angle of force of cut with respect to table and blade edge

θ_c = angle of direction of cut with respect to table and blade edge

F_r = resultant force of cut (lbs.)

$F_c = F_r \cos(\theta_r - \theta_c)$ = component of cut force along direction of cut (lbs.)

$F_s = F_r \sin(\theta_r - \theta_c)$ = component of cut force perpendicular to direction of cut (lbs.)

$F_v = F_r \sin(\theta_r)$ = vertical component of cutting force, i. e., "chopping" force (lbs.)

$F_h = F_r \cos(\theta_r)$ = horizontal component of cutting force, i. e., "sawing" force (lbs.)

L = length of lift cut (in.)

H = height of lift cut (in.)

A = LH = area of cut

The forces F_c and F_s represent the resolution of the cutting force along axes parallel and perpendicular to the path of the cut. Since F_s is perpendicular to the motion of the blade, it does no work during the cutting process. It is desirable that F_s be reduced for two reasons; first, it increases the vertical component of the cutting force; and second, it must be balanced by forces guiding the blade along its path.

The force F_c supplies the energy to cut the paper and will be, in general, a function of θ_c. The vertical component of F_c contributes the major portion of the vertical cutting force and it is desirable to reduce this F_v as much as practical.

Because F_s is zero when θ_c is 90^o, and because F_s would be small for a sharp cut when θ_c is near zero, it was thought that this force would be small in comparison to F_c, and therefore the essential forces of the problem are as shown in Figure (d).

If F_s should prove to be a problem in the operation, it was thought that it could be eliminated by the addition of teeth to the blade. If teeth were added to the blade, as represented by the forces of Fig. (c), the horizontal component F_h could be increased by an increment F_{ha}. Assuming that this could be done relatively independent of changes to F_v, then by a particular selection of teeth, the force F_{rr} at an angle θ_c would result. The effect of this addition would be to cause a new resultant F_{rr} equivalent to an increased cutting force at the angle of cut θ_c. Although the energy of cut would then be correspondingly increased, θ_c might thereby be chosen small enough to allow a net reduction of F_v, and in addition, the knife drive could consist of linkages in pure tension directed exactly along the line of cut, the forces being as shown in figure (d).

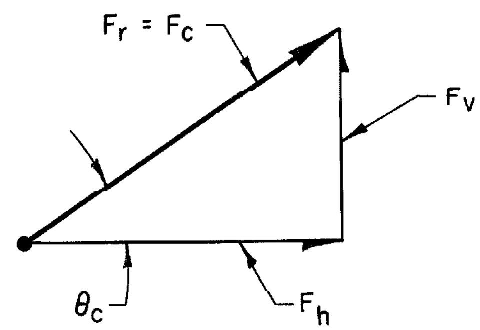


Fig. (d) - Assumed Blade Forces

There is a practical limit to the decrease in θ_c, due to the resultant increase in knife bar length and the concomitant decrease in blade rigidity, and overall machine width. Mainly based on a consideration of machine width, this was taken to be about 20^o. A further criterion that would be applied to the selection of the angle θ_c would be the energy of cut required. No applicable published data were found, and a test was suggested. It was decided that such a test should be made, and should include $\alpha_{br}=1^o$, $\alpha_b=30^o$, $\theta_{ds}=0^o$ as was envisioned for the final machine.

The test was to be used additionally to determine the assumed blade forces as shown in fig. (d) where:

F_c/in. = component of cut force along direction of cut per unit length (lb/in)

F_v/in. = $F_c \sin \theta_c$ = "chopping" force of cut per unit length (lb/in)

F_h/in. = $F_c \cos\theta_c$ = "sawing" force of cut per unit length (lb/in)

E = energy of cut per unit area of cut (in-lb/in^2)

Drop Knife Test Machine - Test Results

An impact test machine proved adaptable to the requirements of the test to determine E for various values of θ_c. As shown in the drawing of Fig. 4 a fixture added to the pendulum contained a blade and one added to the platen contained a provision for clamping the samples which were limited in width to 4 in. The blade was designed so that α_b=30°, but α_{br} was zero, a slight simplification. Due to the small deformation region of the test samples this simplification was expected to yield energy values only slightly higher than would result with α_{br} = 1°. The main disadvantage for this application was the presence of some double-shear, due to the slight arc of the cut direction. The effect of this was minimized to an acceptable level by limiting the depth of the cut. Operation and measurement were similar to that of the original impact machine.

By means of the two fixtures, the positions of the blade and of the paper can be adjusted from horizontal to vertical in increments of 9°. In this way the blade edge remains parallel to the surface of the paper but the angle of cut θ_c can be varied between 0° and 90°. The pivots for angular adjustment are located such that cuts on a

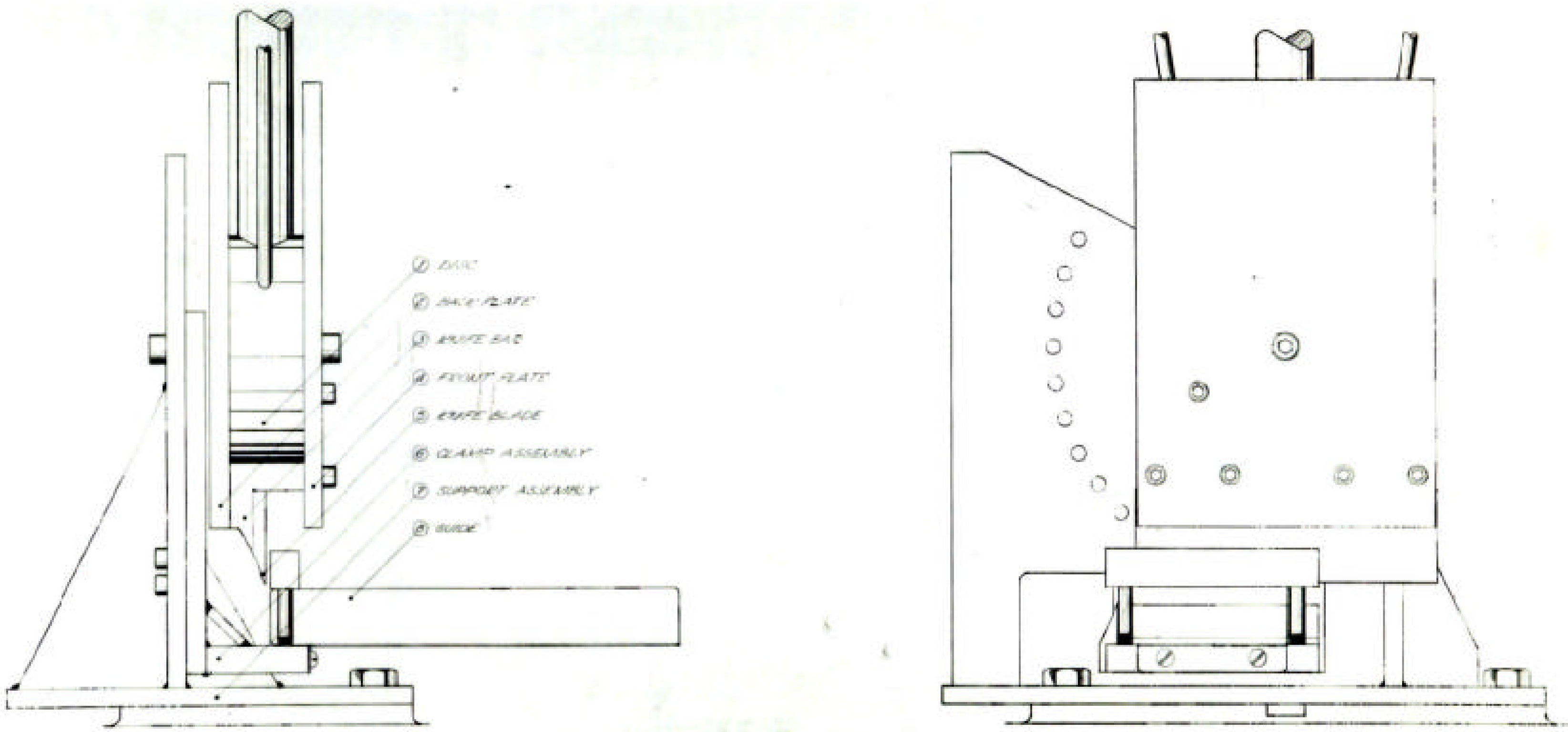


Fig. 4. Drop Knife Test Machine Assembly - DK

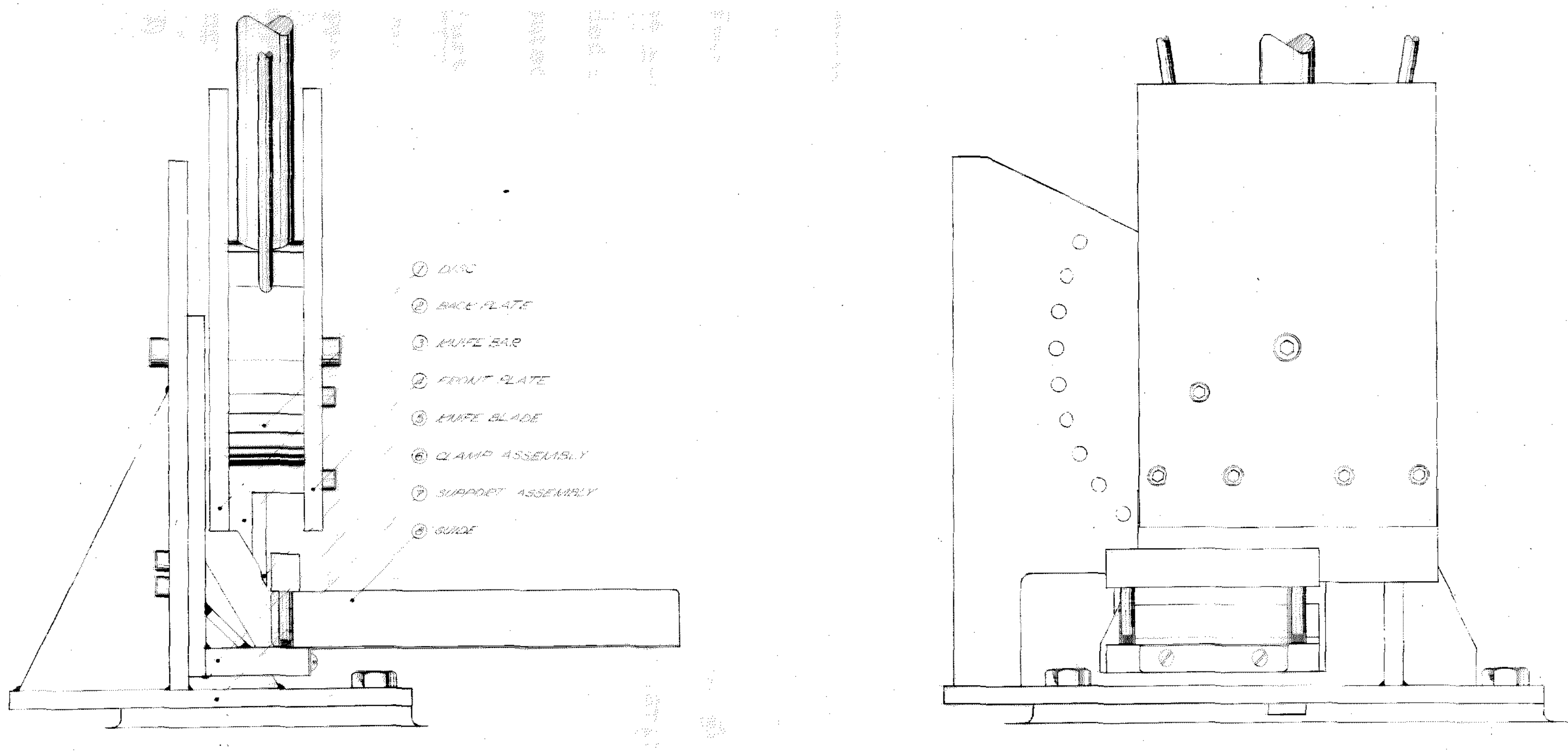


Fig. 4. Drop Knife Test Machine Assembly - DK

standard depth of lift occur for the position of the pendulum where potential energy variations with angle are least. The moving fixture is designed so that the center of mass remains at a constant distance from the main pendulum pivot. Both of these features were adopted to reduce calculations and experimental readings.

Initial Test

It was established by means of the drop knife test machine that, between a high value of $E=250$ lb-in/in^2 for the "chopping" angle ($\theta_c=90^o$) and another high of $E=280$ lb-in/in^2 at an angle as close to "sawing" as the test device allowed ($\theta_c=9^o$), the cutting energy E had a minimum. This minimum occurred at an angle θ_c of approximately 25^o and had a magnitude of $E=150$ lb-in/in^2.

This angle contrasted sharply with the 45^o or greater angle used by some guillotine cutters. It was seen also that large energy would be needed if sawing action alone was employed. This latter point contributed to the rejection of the band idea as noted above.

Further Tests

During the summer of 1962, Mr. B. Saltzer employed the drop knife apparatus to provide more design data. He tested many samples of Ditto paper. The conclusions submitted in his report included the following:

(1) High velocity cuts required less energy. The reduction was not a large fraction of the total energy for the samples considered.

(2) The minimum cutting energy required for each sample tested occurred at an angle θ_c of approximately 25^o.

(3) A local energy minimum occurred at an angle θ_c of approximately 70^o.

The data had revealed a minimum in the energy curve and located it with reasonable consistency. Force measurements on an existing hand-powered cutter had given energy values of approximately the same magnitude as those included in the report. Thus the presence of a minimum, the location of the minimum, and the values of cutting forces and energies as shown in the data were accepted for use in the design.

In order to demonstrate the way in which the test data were used the curves of Figure 5 are included. Results of ten sets of data for Ditto paper are plotted. It was uneconomical to test all variations of speed, stock, blade condition, clamp pressure, etc. Since all curves exhibited the same qualitative aspects, namely the location of the minimum and local minimum, the results were simply averaged. A line was drawn through these averages as shown in the figure.

A smooth design curve was then drawn through this curve as is shown also on the figure. From this curve, values were used to calculate and plot as shown, $F_{c/in} = E\sin\theta_c$ and $F_{v/in} = E\sin^2\theta_c$ according to the assumptions and description given above and on Figure (d).

It can be seen from Figure 5 that a cutting angle θ_c of 20^o requires a vertical component F_v/in of 25 lbs/in. For an angle of

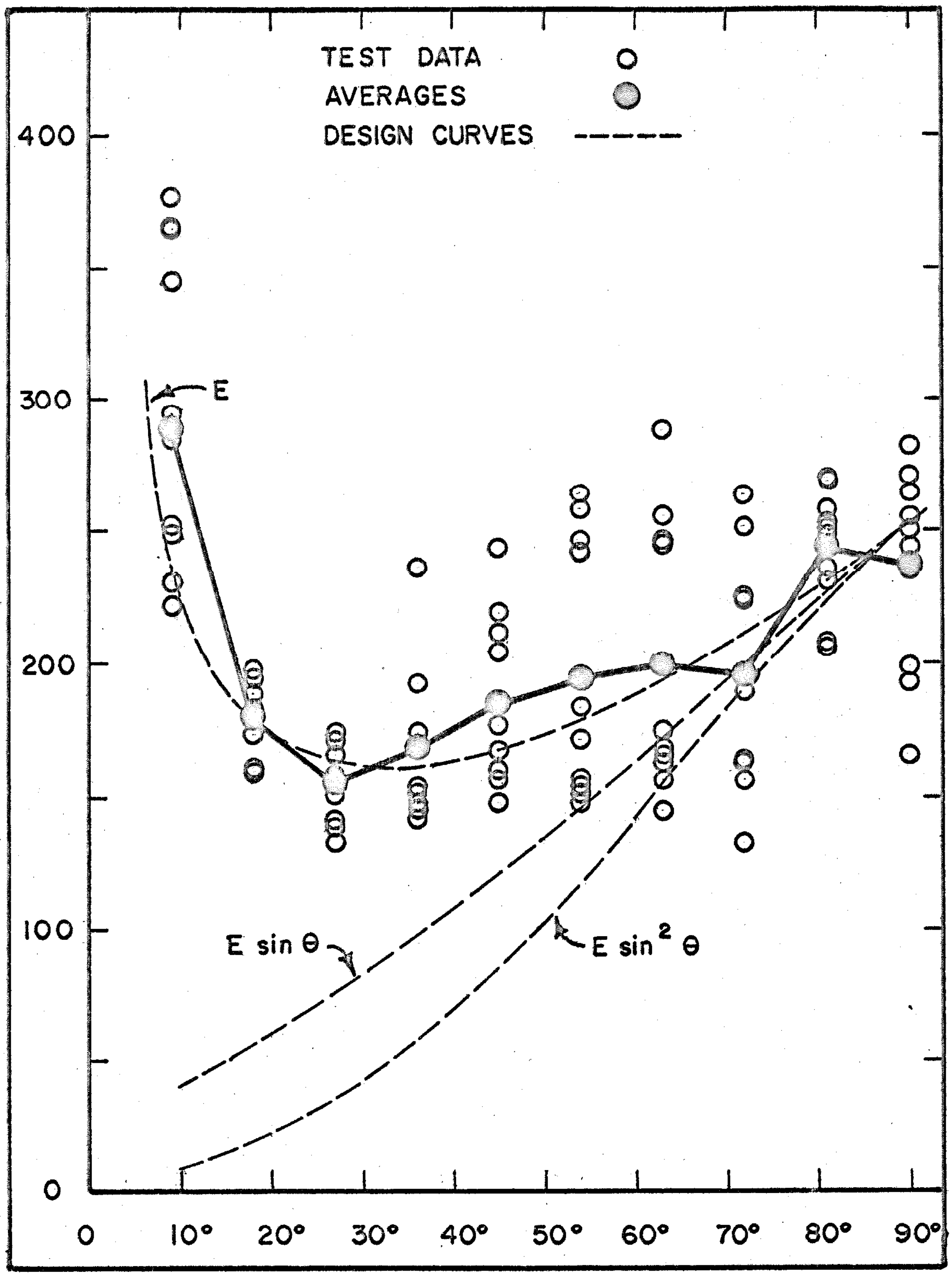


Fig. 5. Energy of Cut - E lb-in/in^2 vs. Angle of Cut -θ_c ^{o}

50^{o} the vertical component would be 100 lbs/in. It was concluded that the effort to establish criteria for the reduction of F_V had been justified. Because for values of θ_c below 20^{o} the energy begins to increase rapidly, and also the width of a guillotine type cutter would tend to be excessive, the gain represented by a reduced θ_c of approximately 20^{o} was considered to be adequate.

On the basis of the above considerations and data three further criteria were established for use in the design:

(1) The line of resultant cutting force should be established approximately parallel to the direction of the cut.

(2) The optimum angle for cutting is between 20^{o}-25^{o}.

(3) The force of cut per unit length, F_c/in, is approximately 75 lb/in.

With the results of these studies as a guide, one of the paper-cutting machines available in 1960 will be examined.

2.4 A Commercial Machine (1960)

The Lawson paper cutter commanded a large share of the 1960 U.S. market and embodied an acceptable industrial solution to the paper cutting problem at that time. In this study it was used initially as a standard of required performance and later for comparison of machines and solutions considered. It was manufactured by Miehle-Goss-Dexter, the "world's largest manufacturer of graphic arts machinery" and one of their salesmen discussed and demonstrated its outstanding features to the author. Figure 6 shows overall photographs of the cutter and representative details are shown in Figure 7.

For purposes of discussion the features of the Lawson cutter are grouped into four categories: the knife; the clamp; the backstop; and general features.

Knife

The knife blade and its bolted attachment to the knife bar are seen on the front view in Figure 6. Knife drive linkages are internally connected at the ends of the knife bar as shown in the "knife linkage" view of Figure 7. Pivot points for the drive linkages are situated in the base casting, and can be seen in the rear view. On one of these is an adjustable connecting bar for setting the knife linkage bottoming position.

Partially visible is the flywheel and electric clutch assembly on a single shaft. For rigidity, a bracket has been added so that both ends of the shaft are suspended. Power to the flywheel is supplied from the motor by a multiple v-belt drive. A crank on the clutch

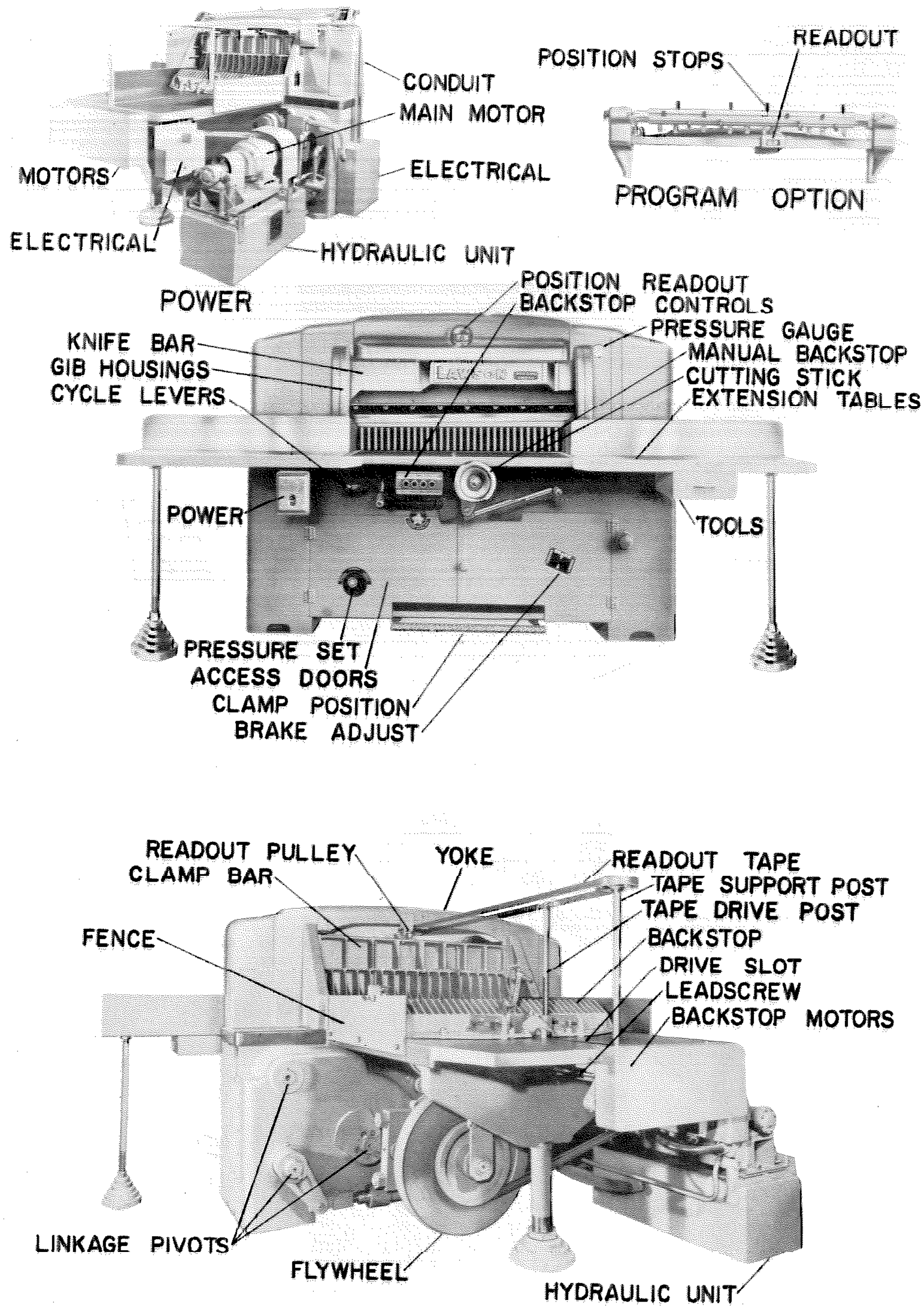


Fig. 6. Lawson Hydraulic Clamp Papercutter (1960)

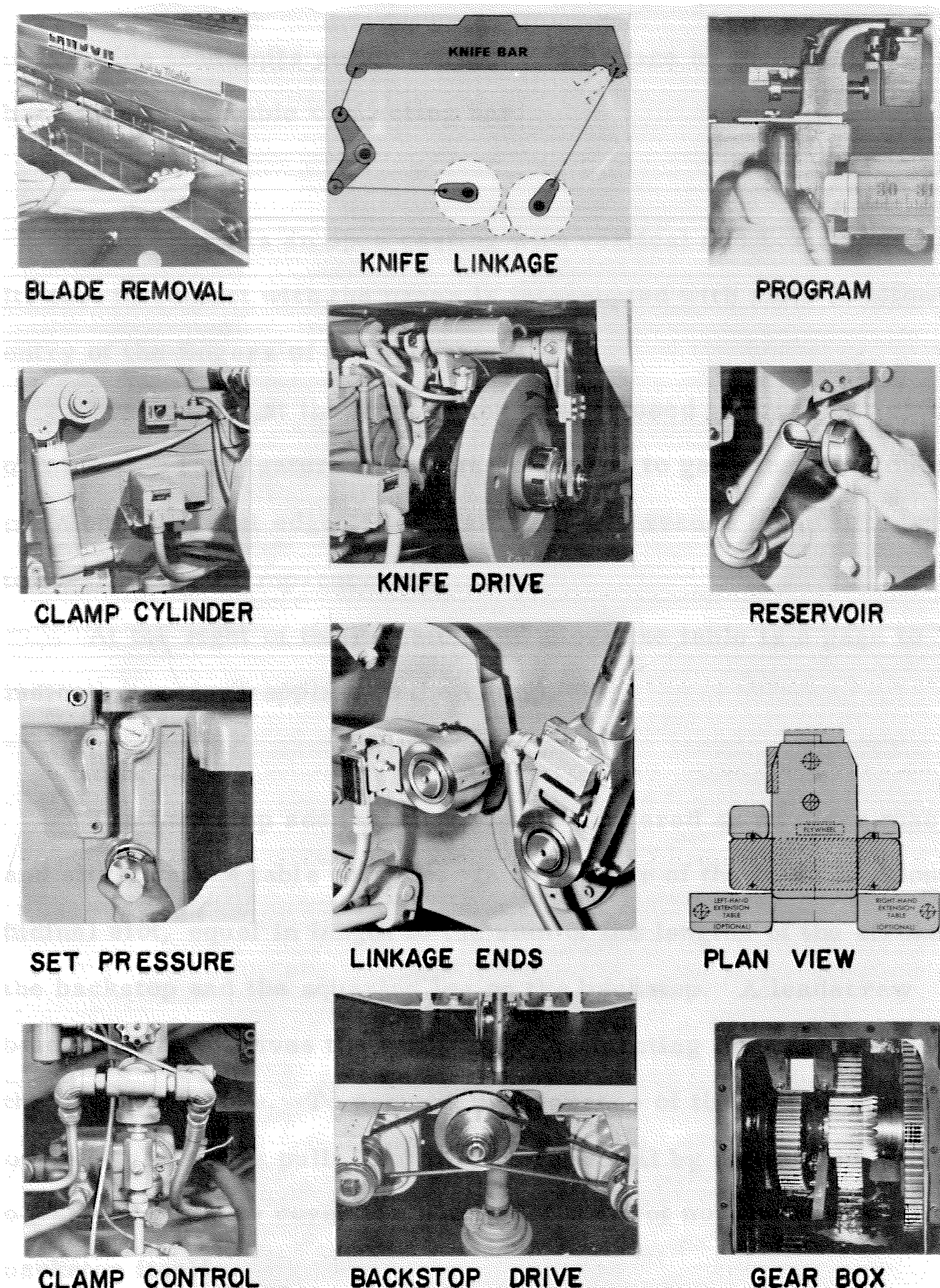


Fig. 7. Features of Lawson Papercutter (1960)

output side transmits power to the knife linkage by means of the horizontal adjustable connecting bar.

Clamp

The clamp is an open casting with vertical and horizontal ribs. Its foot in contact with the paper is interrupted with slots to allow entry of the fingers of the backstop.

The treadle at the front of the base is used to lower the clamp on command. This feature allows the operator to gage the location of a cut along the front edge of the clamp, and to secure bulky lifts prior to starting the clamp-cut cycle.

At the right of the machine and above the table is a gage to indicate pressure applied to the clamp.

Backstop

The backstop seen in all views, is fingered as on older machines and slides on the table surface. In the center of the table is a longitudinal slot, equal in length to the sum of the lengths of the stroke of the backstop and the squaring leg on the backstop. A leadscrew below the table drives the backstop by connecting through this slot to the backstop above. To prevent interference of the slot, a steel band or "slot closer" is pulled out of a storage coil by the motion of the backstop beam, to cover the portion of the slot not occupied by the backstop itself.

At the rear center of the table is a box which covers the electric motors and transmissions associated with the backstop drive. The transmission allows selection of two different speeds for the motion of the backstop.

The backstop is divided into three sections which can be set at different distances from the knife. Thus one cycle of the blade trims to three different dimensions, simplifying the trimming of books and other similar objects. In addition, the angle of the backstop plane with respect to the plane of the table can be adjusted by means of eccentric mounted cam rollers that ride the table.

By means of a readout lens set at eye level in the front of the machine, the operator can read, on a continuous steel tape rigged between two pulleys, the dimensional location of the backstop. The support for one pulley of the tape readout device is formed by a vertical post situated at the rear center of the table. The main casting just behind the lens forms the support for the other pulley. A post on the backstop moves a point on this readout tape.

The drum type object, or "program option" of Figure 6, is the memory device for positioning the backstop. One of six programs can be selected to produce a preset cutting sequence. Programmed positions are recorded by means of opaque tabs either set in slots or on bars removable for storage. By detecting interruption of a light beam by these tabs, two photocells on a travelling head dictate first that the slow advance motor be used, then that the brake be applied. To assist in accurate setting when the program option is included, extra pulleys allow the backstop motion to drive a readout lens along a fixed scale situated below the stops of the memory unit.

General Features

Two side fences are included for guiding the stock at right angles to the cutting plane.

On either side of the front of the main table are extension tables for storage of stock before and after cutting.

Suspended from the tables are the motor start box, two handles for running the cycle, extra operation buttons, a center wheel attached to the lead screw for manual advance of the backstop, and a tool drawer.

Lower doors at the front of the base are to facilitate maintenance and to cover internal mechanisms.

A massive welded rear leg permits height adjustment by means of screw threads at the foot.

The hydraulic unit of Figure 6 offers the best view of the motor. At one end is situated the drive belt to the flywheel and to the other is attached the hydraulic pump. The whole unit is mounted on the oil reservoir for the clamp hydraulic system.

Five electrical boxes are mounted on the machine; these house relays for the control of the backstop motors, the electric clutch, and other electrical devices. Conduit interconnections are required for the AC voltages employed.

Specification Sheet

The cutter is a 46 in. model. Quotations from the specifications sheet suggested some immediate improvements.

(1) "Backgage goes back 53 inches."

This is out of a total table length of 102-1/2 inches in front of the operator. If the typical distance of 24 inches is allowed for "reach" in front of the blade line, 25 inches are consumed behind the full back position of the backstop. Since this length is used mainly to provide a track for the squaring leg of the backstop tee, a different system should be sought.

(2) "Knife length of 58 inches."

For a cut length of 46 inches this allows a maximum of 12 inches lateral motion during the descent through a 6-1/2 inch lift. If all 12 inches are used, and a straight line knife motion is employed, the cutting angle would be 29°. Since neither is the case, it is estimated that a cutting angle of at least 40° is used in this machine.

(3) "Weight of electronic spacer machine minus weight of motor-operated backgage machine = 1000 lbs."

For this machine it should be possible to make this change with only the addition of a drum memory and a sensing device. This suggested that the electronic option had been an afterthought. Savings could be realized by providing for automatic options in the original design concept.

(4) "Gross weight of electronic spacer machine = 13,000 lbs."

This weight seems excessive and its reduction would be desirable. Apart from the reduction in manufacturing and handling costs, it should be possible to eliminate the expense of floor reinforcement where these cutters are installed.

Fact Sheet

Introduction

From the "fact sheet" available from the manufacturer of this machine two lists were prepared. The first refers to features termed as advantages or improvements. The second refers only to portions of the machine, and was regarded as consisting of statements of reduced inherent defects. A careful consideration of these facts and their implications served as a minimum statement of the problem and as a necessary substitute for direct experience in this industry.

(a) Advantages and Improvements

Knife

"assured straight cuts all the way across the sheet"

"smoother and exceptionally accurate cut"

"long shear action"

"positive double-end pull straightline knife action"

"a new double crank design"

"less linkage"

"fewer bearing surfaces"

"adjustable third or center bearing"

"over-sized adjustable gibs"

"knife adjustments are made at front of machine"

Clamp

"prevent draw when handling wire stitched signatures"

"for gaging the cut or taking the air out of bulky signatures"

"the lift is held firmly, and can't shift out of alignment throughout the cutting cycle"

"assures perfect alignment no matter at what point the paper is clamped"

"clamping pressure is easily adjusted to meet individual requirements of each job"

"instant clamp pressure as pre-selected by the operator"

"won't mar top sheet or disturb alignment of lift"

"smooth, gentle hydraulic, clamp action"

"the pressure gage is at eye level"

"clamp pressure knob is at arm's length on the front of the machine"

Backstop

"backgage 100 % square with the table"

"three piece fingered backgage" - useful for book-trimming

"backgage may be tilted"

"easy to read magnified measuring band"

Table

"air cushion device"

"built-in fluorescent light fixture over the table" "rear table is cross-ribbed"

"a single sturdy welded support"

Safety

"the knife positively cannot repeat"

"a heavy steel locking bolt is forced into place at the very beginning of each cycle"

"all moving parts are enclosed for maximum protection"

"requires that both starting levers be engaged for each cut"

"levers return to neutral position under their own weight"

"no need to rely on spring" for safety of operator's hands

"electro-hydraulic clutch"

"electro-hydraulic brake"

Machine Protection

"to prevent damage if knife digs into table too deeply"

"all moving parts are enclosed"

"prevent damage if knife is dull or hits a foreign object"

"improved overload device is incorporated"

"gears operate in a sealed bath of oil"

"the flywheel cannot cant because of machine settling or floor sag"

Accuracy and Rigidity

"maximum cutting accuracy"

"20% heavier than other cutters"

"massive construction"

Productivity and Capacity

"42 strokes a minute"

"operator ability to quickly move heavy lifts into correct position on the table" - (by means of the air lift feature)

"tons more paper cut per working day"

"minimum of down-time"

"easy to read measuring band helps keep production speeding along"

"6-1/2 in. clamp opening" - "greater capacity"

Convenience

"reduces operator fatigue"

"controls on front of machine"

"simple upwards flip of the two-hand operating levers"

General Features

"standardized to grow with you"

"modifications can be made right on your own floor"

"both ends of the drive shaft are fully suspended from the machine - not mounted on the floor"

"flywheel is located under rear table"

"fool-proof stick type gage"

"25 gallon hydraulic reservoir"

(b) Inherent Defects

"bowing" - (of the knife bar)

"wear" - (of the sliding surfaces, knife edge, and cutting stick)

"replacement" - (a necessary, sometimes dangerous, inconvenience of replacing worn cutting sticks)

"no-repeat" - (a reference to the fact that accidental cycles of these machines had caused loss of many arms)

"improved" - (a reference to the apparent fact that overloads as a result of foreign objects had damaged the machines due to inadequate overload devices)

"less..." - (a reference to the apparent acceptance that these machines were complex - it was applied to linkage)

"overheating" - (apparently overheating of the oil had caused discomfort and perhaps could have distorted the machine sufficiently to reduce accuracy)

"positive" - knife action - (due to long chains of force and pivots in the linkages used, the word "positive" apparently evolved to represent nearly identical blade position for each position of the blade drive)

"if knife digs into table" - (this possibility, which could easily damage the blade or the cutting stick apparently resulted from "non-positive drive")

"100%" - backstop square - (this need to use the adjective 100% indicated that squaring of the backstop was not easily done)

"sag" - floor - (reference to this feature further justified the attempt to reduce machine weight)

"settling" - machine - (this reinforced the conclusion for "sag" and led to the decision that relative positions of machine components should remain independent of the floor shape)

The air-cushion device used for handling of heavy lifts, should be explained, as it has been extremely successful. Small ball check valves are set in the table at approximately 12 inch centers. Low air pressure is piped beneath the table to each of these valves. When a heavy lift contacts these valves the released air causes the lift to "float". To allow for friction deceleration of the lift, the air pressure is removed during operation of the backstop.

Examination of the commercial machine in detail had given some understanding of the history of the design and had yielded a minimum statement of the problem, as well as an upper limit of the eventual solution to that problem. Having formerly identified a minimum guillotine paper cutter, the limits of the solution were defined.

It was felt, however, that at little extra cost, some extra functions could be added to a solution still between these limits. One such function was suggested by the clamp. It seemed inherently suited to adaptation as a wide throat low force punch press. Print shops so equipped could be able to cut corners on note paper, punch, or stamp with steel rule dies. At present extra machines have to be purchased. It would be necessary only to be able to operate the clamp independently of the knife, and provide a way to install die sets.

2.5 Analysis of the 1960 Machine

The existing machine was tested against the three working criteria presented in section 2.3, and a fourth criterion based on the properties of hydraulic systems.

(1) The line of the resultant cutting force is in the direction of the relative knife motion.

The design of the knife linkage shown in Figure 7 was not necessarily based on this assumption. If this assumption is applied, however, a component of the force to drive the knife acts normal to the ramp even for relatively steep angles of the cutting force. Until such steep angles are exceeded, say due to a dull knife, the path of blade travel will be controlled by just one side of the guide slot. It was concluded that the Lawson design neither supported nor refuted this assumption.

(2) The optimum cutting angle lies between 20^{o}- 25^{o}.

A cutting angle of about 40^{o} was seen in the figure. This presented an opportunity to reduce the size of the vertical forces and associated members.

(3) Blade side forces could be reduced by means of a back-clearance.

The back-clearance feature was not seen in brochures or on other machines although an extra third bearing to prevent knife bow had been noted. It was concluded that by the addition of back-clearance, the third bearing might be eliminated and the thickness of the knife bar reduced.

(4) Hydraulic power was to drive all portions of the machine.

Examination of Figure 6 showed that several advantages could be gained by the total application of hydraulic power. Only one motor and pump would be necessary, inasmuch as the backstop motors, geared transmission and leadscrew and the knife motor, clutch and flywheel could be discarded if hydraulic actuators were used instead. Thus the proven hydraulic clamp feature could be retained, shocks as the knife struck a lift would be reduced due to the resilience of the oil drive, and simplicity of construction would be possible. Further, it was noted that a low speed high force device like a hydraulic cylinder could be almost directly applied to drive the backstop, whereas the existing high speed low torque electric motor drive required conversion of force, speed, and control signals, and was not inherently suitable.

Some problems were foreseen with an all hydraulic machine. It would be difficult to supply a manually powered backstop to compete with the handwheel drive of the existing machine. Clamping of the backstop in a fixed position would be complicated by the absence of a leadscrew. Finally, an acceptable hydraulic drive for the knife was not available at this time. The problem of obtaining an adjustable, precise bottoming position for the knife blade remained. Efforts of former students to apply a rotary hydraulic actuator to the drive of a knife linkage, thus using over-centre action for location and deceleration, showed mainly that the associated details would be complex. Efforts by the author to apply a direct drive in the form of a hydraulic cylinder (linear actuator) had not yet revealed any way to quickly stop and reverse the vertical motion of the knife bar at a precise location.

III. PRELIMINARY DESIGN CONCEPTS

In this section the preliminary design concepts developed for the machine are discussed and features of the design at this stage are presented.

The writer attempted to determine the nature of all the solutions to subproblems of the overall machine prior to any attempt to finish individual subassemblies. In this way as many advantageous changes as possible could be made before it was necessary to finish the design. Furthermore, it was a firm policy that the effort committed to any single idea regarding the nature of an assembly should not prevent its rejection if such should prove advantageous.

At this stage two years of work had produced a large number of sketches, layouts, and calculations, each effort making some small contribution to the next. No final drawings or hardware had been produced but some specific recommendations for the nature of the final assemblies and details had been established. Based on the concept of a minimum guillotine type cutter as envisioned, this preliminary design indicated that the final machine would have some major distinctions over all other cutters seen. For any cutting angle selected, the design would result in minimum height, width, length, and base thickness. Other advantages of the design are pointed out in the following discussion.

A. Table

Only slight use was made of the possibilities of the table casting. It was regarded as a foregone conclusion, that it would be a fairly rigid slab. Extra transverse beam stiffness was to be supplied beneath the line of cut. One innovation was to be incorporated: instead of including a cutting stick, a small groove was to be made in the table under the knife line. This could be filled with a low melting point metal, such as cerrobase, and the surface could be scraped flush with the table to prevent any snagging of the paper. Repairs could be made by means of a wire heater set in the slot or by means of a household flat iron or a soldering iron. Alternatively a plastic could be used, to fill the slot, and supplied as a paste for ease of repair.

B. Knife, Base, and Clamp

A light-weight removable knife was obtained by designing a diamond-shaped cross-section with two cutting edges. Tapped holes allowed it to be mounted by means of bolts in a vee-slot machined in the knife bar.

A proposed design for the base and knife drive solved many problems. In this design wheels attached to each end of the knife bar were to ride ramps built onto the base. The ramps guided the blade in a straight line motion with the edge parallel to the table until it approached the bottom of the stroke. Here, valleys at the ends of the ramps provided a horizontal portion which controlled precisely the bottoming of the blade. Beyond the horizontal portion overshoot could be allowed for deceleration. As a result a hydraulic cylinder could drive the knife directly, thus the drive problem was solved. An additional advantage was that the last few sheets would be cut with a long, low slicing action, thus relaxing requirements on the cutting stick and blade sharpness.

The wheels would be mounted in adjustable eccentric bushings so that the knife bottom position could be set. The two ramps each had a double track. Rotation of the bar would be prevented by having one wheel ride each of the four tracks. The inner two sides of the ramps would be finished to keep the knife edge in a fixed plane.

"Force closure" guarantees a zero-clearance fit between two surfaces by providing a reaction between them. This principle was to be incorporated in the knife drive to allow the use of the ramps instead of slots. For a compact machine, the knife pull cylinder would be mounted to the right side end of the knife bar as the ramps thus knife motion sloped downwards from right to left, a custom on guillotine cutters. To counter the resultant tendency of the left end to jump from the ramp, an extra or "hold" force would be supplied, possibly by a spring. The magnitude and direction of this could be calculated when the cutting forces were known.

If analysis should indicate an advantage, the hold force could be made proportional to the cutting force. This could be accomplished by means of a small hydraulic cylinder to replace or supplement the spring initially proposed. The fact that pressure to this cylinder and to the knife cylinder would be the same would reduce ramp wear.

If the rod of the "hold" cylinder were attached to the end of the knife bar but the cylinder itself were inside of the reservoir, a problem would result since a slot would have to be sealed against dirt. It would be easier to seal an exit at a single point. Additional sealing advantages would result if the hold cylinder could be located above the reservoir level. Also it was seen that if a spring were to be used to supply an essentially constant hold force, its length would depend on its stroke. It was therefore decided that the hold force was to be provided by a cylinder, transmitted by means of a cable, and located by the position of a pulley. Sealing would then be at a fixed point. In addition, the position of the hold cylinder was no longer determined by the point of application of the hold force.

For the cutting of lifts located against the fence and with the assumed cutting force and direction, it was shown graphically that positive wheel loads could be guaranteed. If the forces were much larger, but in approximately the assumed directions, the blade would stall. If the angle were greatly increased the wheels would leave the ramp. Either case might result due to foreign objects under the knife, or a dull blade but the resultant stall or wheel jump would not damage the machine. Thus an overload device would not be needed.

Due to the knife drive selected, no superstructure or yoke above the table as seen on other machines was required. This region needed only to be made safe. Cover boxes, hinged to avoid pinch points, would be provided over the ramps and ends of the clamp and knife bars.

The rear of the base would carry pads and brackets to provide for vertical guiding of the clamp. The hydraulic reservoir would be included in the base casting.

The slot sizes on the foot of the clamp would be reduced in order to reduce its width for the same area of contact with the paper. The clamp assembly would consist of a cast beam with connecting rod links attached at both ends. A torque tube situated in the base would synchronize the cranks attached to each connecting rod and thus the displacement of the ends of the clamp bar. The torque tube would be rotated by a hydraulic cylinder situated above its midpoint or at some other location to be determined by deflection considerations. If large in diameter this tube could be thin-walled and light. Should this absorb too much volume, it could then be used as an extra reservoir or heat exchanger, or both.

Another advantage of the knife drive was that cutting and clamping forces would be carried directly into the base thereby reducing the need for stiffness, particularly torsional stiffness, throughout the assemblies. It would be sufficient to supply major beam strength and stiffness in the knife and clamp bar and in a portion just below these bars. In contrast, other systems, with a circuitous path between actions and the equilibrating reactions, require major overall stiffness to preserve geometry.

To reduce necessary overall stiffness further, and simplify installation, a three point support of the machine was desirable. Thus vertical motions of the corresponding points on the floor would not distort the machine. The rear table edge could either be triangulated to three bolted points on the base, or a leg could be supplied at the back of the table. Due to the short table length and weight distribution, it was felt that if preferred the machine could easily be designed to stand on the base alone.

C. Backstop

Drive

A manually powered backstop was discarded in favor of a powered backstop with manual control. The powered backstop would be made to compete in cost with the manually driven backstops of other cutters.

When the backstop had formerly been considered one particular problem seemed difficult to solve. Both the backstop beam and fence had to be above the table. To reduce the stiffness requirements of associated members the backstop should be driven from both ends and from beneath the table. This could be done by means of a single drive under the table center, equipped with a tee-shaped follower to connect to the two ends of the backstop. Others had provided a slot over the center drive to connect to the backstop bar. This slot was to be eliminated. In order to make the connections, a rigid bracket from the end of the drive tee and around the fence to connect to the backstop beam seemed necessary. This limited the clear space that had been available around the fence and capstan. It was solved finally by a sliding lamina connection between the fence and the table edge. Buckling calculations showed that the slot necessary was only 1/16 in. and would not interfere with the paper on the table.

In the beginning it was thought that the T square type of squaring required table length behind the rear backstop position. Thus other systems of squaring had been proposed before it was realized that an inversion of the T square solution had been overlooked. If the squaring function was performed under the table, i. e., associated with the drive, then the leg of the tee could protrude toward the front of the machine without adding length, because in front of the knife line were two feet of clear table length. This squaring system was accepted as better than all others proposed. The backstop features described would allow the design of a guillotine cutter of minimum length equal to the sum of the reach allowed for the operator, the travel of the backstop, and the thickness of the beam required to push the paper.

Since for any type of tee-squaring device a rigid leg would have to be provided, a way was found to use this feature twice. A hydraulic cylinder could serve as the rigid leg for the tee. This tee would carry the necessary speed and automatic control components for the drive. Sufficient space would be left at the top center of the base, the center line of the table and the center rear of the backstop to allow clearance for the additional plumbing.

<u>Control</u>

No great effort was applied to the actual backstop speed and position control valve unit at this time. This was postponed with the idea that the unit might be about fist size. It could contain both the position sensor and the speed control, and be attached externally to the end of the tee near the location of the capstan. Several solenoids would probably be included, to activate the position sensor and to retract it for program changes. Manual control would be by the pull cable system and if necessary a lever or pulley could be used on the valve to change the size or direction of the signal. The same basic control would serve also for the automatic option. The only change would be that a mechanical signal would come from the program bar, instead of from the operator. Thus the automatic feature could be added with little extra expense or weight.

Capstan

Much effort was applied to determining the location for the capstan or automatic control option for the backstop. It would contain notched steel program bars for sequential cuts. The main reasons for its final location on the fence are:

(1) Clear space for it was available because a lamina sliding between the fence and table could drive the backstop.

(2) The fence could be offered in two options, with or without the automatic spacer.

(3) Position information and feedback should travel the most direct route to the backstop control.

(4) The operator would be near the fence side of the machine to load paper.

(5) Program selection controls and backstop controls should be located near the operator position.

(6) Manual program selection and backstop speed control was best accomplished by direct mechanical means.

With the capstan mounted to the fence, and both situated at the left side of the table, selection of programs could be accomplished by an extension of the center line of the capstan through to the operator position at the front of the machine.

Automatic Programming

A notching punch could be included on the moving backstop assembly. Hydraulic power was available on the backstop, and due to the short punch stroke a diaphragm type of actuator probably would be used. This feature would enable a permanent program to be made on the first run of a repetitive job.

Readout

Readout of backstop position would be provided by a spring loaded reel of graduated tape. Sag effects could be reduced by the spring force incorporated in the reel. The addition of a single idler pulley would permit readout at eye level and observation of the tape would be by means of a lens at the front of the machine.

D. Hydraulic

A hydraulic circuit was prepared similar in essentials to the one finally incorporated. The final circuit is shown in Figure 8. Plumbing would consist of main assemblies mounted to the base, and connected within the reservoir. Manifold blocks would reduce connections. The motor-pump assembly and manifold block would be attached to the rear of the base and toward one side and would include the four-way valve.

On the other side and also attached to a manifold block would be the sequence valve and high pressure relief valve. From the outside of the manifold block connections to the backstop could be made. A foot operated valve to permit gravity drop of the clamp could be incorporated in that assembly. Since leakage is undesirable, and intolerable above the level of the table, as much of the circuit as possible would be situated inside the reservoir and all of it would be below the table level.

E. Electrical

Only a general conception of the electrical circuit was formulated at this stage of the design. In "power off" or idle conditions, all components must move to "safe" positions. Thus spring-loaded or normal positions of valves and relays should be such as to tend to raise the clamp and knife and move the backstop in the speed-controlled direction. Switches did not necessarily have to be standard. The decision to bring AC power only as far as the motor and operate all controls with a 28 volt DC power supply permitted the use of physically small electric circuit components and simplified the design of custom switches.

The relay box would be located in the left side cover box. This would allow for short cable connections to the backstop, base and control panel. In addition it would be easily accessible for any maintenance. It would also be possible to locate the operator control panel on the outside of the box, since the operator works to the left of the machine. To reduce down-time, instead of requiring skilled maintenance most of the electrical control system could then be replaced as a unit.

IV. THE FINAL DESIGN AND CONSTRUCTION OF THE PROTOTYPE*

This section includes assembly drawings and circuit diagrams of the prototype machine, PC 64. Descriptions have been added to show more clearly the functions of the assemblies and their elements, as well as specific considerations that led to their final forms. It should be recognized that at early stages in the design of the machine the form of a particular component was the result of opinions based on a large number of factors. The forms were not frozen into hardware until absolutely necessary. As was discussed in the Introduction, a formal procedure was not used in the design process because of the complex coupling of the functions and properties of the different components.

For convenience of presentation, the description of the resultant paper cutter is divided into portions corresponding to the major components and systems of the machine.

4.1 The Hydraulic Circuit.

The hydraulic circuit first proposed differed from the one finally incorporated and shown in Figure 8 in that the SEQUENCE SET solenoid assembly and the STROKE SET features were absent. The METER-IN backstop circuit first proposed was later changed to the METER-OUT system. Meter-out means that motion is controlled by

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* Photographs and overall assembly drawings of PC 64 are included in Figures 27, 28, 29, 30 and 31 of section 4.15.

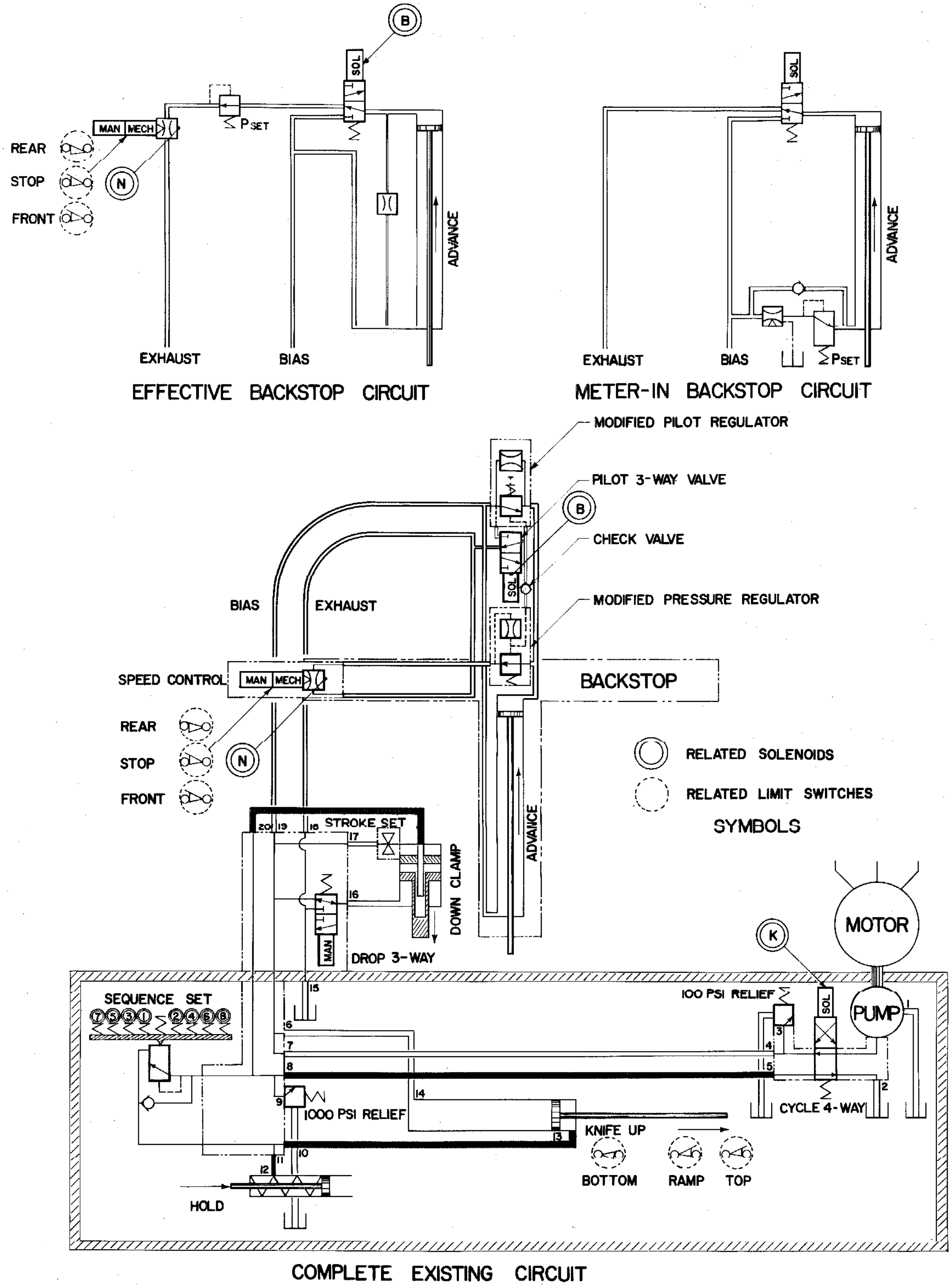


Fig. 8. Control Circuit - Hydraulic

metering the flow out of the cylinder.

The essential features of the hydraulic circuit shown in Figure 8 can be described as follows:

Base System.

Locations 7 and 8 in the circuit can be regarded as the principal sources. To each of these is connected a relief valve and a pipe to reservoir exhaust. The 100 psi relief valve for 7 is regarded as the source of BIAS pressure (LP). The 1000 psi relief valve for 8 is the source of high pressure (HP). A single positive displacement pump supplies both sources. A cycle 4-way valve acts so that if the flow of the pump is connected to LP, the HP line is exhausted, and conversely. Thus if all of the discharge cannot be expelled downstream against pressure resistance equal to or less than the setting of the relief valves, some of the flow is expelled through the relief valve. This acts as a protection for the pump and hoses, and also serves as a source of supply of constant pressure oil as well as of power loss when the relief valve is open. In addition, since an open-center 4-way valve is used, pump output is never totally blocked.

Because the backstop is stationary when the knife and clamp act, and since it requires considerably lower forces during an advance-clamp-cut cycle than do the clamp and knife, two operational modes were established. When the low pressure is supplied, the knife and clamp tend to rise, and the backstop circuit is supplied with bias oil. When the high pressure is supplied, the knife and clamp are powered and the backstop supply is removed. The spring return of the 4-way valve is such that both in the "cycle off" and "power off" positions, only bias is supplied. When the cycle buttons are depressed, flow to the knife and hold cylinders does not occur until pressure at the clamp builds up to the sequence set pressure. In operation this corresponds to a lowering of the clamp, application of force equivalent to the sequence set valve pressure, and then flow to the knife and hold cylinders to cause knife descent.

The drop 3-way valve was proposed to allow the operator to drop the clamp under gravity. This was in order to sight a line of cut, or to apply initial pressure so that subsequent full pressure application to the clamp would not shift or "draw" the paper.

Meter-in Backstop Circuit.

This circuit was considered prior to the effective backstop circuit (meter-out) that was built. Since control valves for both were later designed, it will be described briefly.

To reverse the backstop, bias would be applied to both ends of the cylinder and speed would be governed by pressure drops inherent in the piping. This is usually called a "differential" circuit. For equal pressures on both sides of the piston, the resultant force is equal to the pressure multiplied by the rod or "differential" area. Only a volume of oil equal to the volume of the piston rod need be supplied for a full stroke. In such a device the force is small, but the speed is high. Some increase of force is offered by the fact that pressure drops in the bias supply portions are reduced due to the small flow added.

For advance, exhaust is substituted for bias on the face of the piston. The input flow to the rod side of the piston is forced to pass through the variable orifice of the speed control. To make the size of the orifice correspond to a definite speed the pressure drop across it must be known for each orifice size, and be independent of load. For a meter-in control this is difficult to realize without using a complicated valve.

Quick Exhaust.

The exhaust line to the control valve can best be explained by introducing some special properties of the load that the valve controls, i. e., the backstop and the lift which it transports. The problem is to position the lift of paper within .002 in. as indicated by a mark on a program bar. Ideally, this should be done in the shortest possible time, by accelerating for a portion of the distance to a mark, and decelerating for the remainder. The valve considered for meter-in control of this machine is simpler than would result if that stipulation was included.

The action of the valve adopted is such that for spacings greater than 1/2 in. the backstop and lift tend to accelerate to a maximum speed. The speed remains constant until they are within 1/2 in. of the programmed stop location. At this point the backstop decelerates. The absolute value of this deceleration, divided by g, the acceleration due to gravity, can be called the equivalent coefficient of friction of the backstop.

For no sliding or overshoot of the sheets of a lift beyond the backstop, this value must not exceed the coefficient of friction between the sheets to be cut. In addition, it must not exceed the coefficient of friction between the table and the paper. Thus the maximum deceleration without appreciable overshoot corresponds to $\mu_{p_{min}} \cdot g$ and assumes $\mu_{pT} \geq \mu_{p_{min}}$.

where $\mu_{p_{min}}$ = minimum value of interleaf coefficient of friction,

μ_{pT} = coefficient of friction between table and paper,

g = acceleration due to gravity.

This clearly shows a theoretical limit to the speed with which a guillotine cutter can operate if the backstop is driven in this way. Future work should be able to avoid this problem, but the lift must not be free to slide under its own weight.

The main reason for dealing with this point was to explain the presence of the exhaust line from the speed control valve. For a large deceleration distance and relatively high speed, no overshoot could be tolerated. But for a very low speed and short distance this is not the case. Overshoot can be calculated, and an acceptable error allowed. At a calculated distance from the mark, the valve acts so that the small flow from the orifice to the input to the cylinder is diverted to the exhaust line.

Due to the low flow no line losses will be present, and both sides of the piston are at exhaust output pressure. If this is atmospheric, no net force is applied to the backstop. It decelerates according to its actual coefficient of friction, which should be designed higher than that of all paper to table values. If this is not the case, the backstop and the lift decelerate together. The resultant overshoot would then depend on the size of lift as well as the coefficient of friction of the lift.

A notable defect of this idea for this system is the lack of a locking feature. For example, position could be disturbed by jarring a lift on the backstop. One notable advantage of the idea is that instead of supplying a perfect seal at a precise position of the plunger, the valve is required only to open a port to exhaust.

4.2 <u>Base Assembly - B - Figure 9</u>

The base assembly connects all other components and is shown in Figure 9. For five positions on the ramp, and a number of resultant cutting forces, various knife bar configurations were investigated for this assembly. The properties of the drive chosen were determined mainly by the need for maintaining contact between the wheels and ramp during the cutting operation.

The necessary openings in the base were placed to allow the highest reservoir level without the use of gaskets. A center hole was left for a pin to support the knife, hold, and clamp cylinders.

Areas of the elevation view were allotted for access to attached subassemblies. A clearance between the left side table edge and the vertical ramp face was provided for a paper deflector device. The paper deflector would be a swing gate pivoted on the fence and descending with the knife. As opposed to other designs, this would leave no gap below the knife and thus prevent the scattering of trimmed strips or small labels.

A small area was provided for passage of a capstan extension rod for program selection from the operator position. If the capstan could not be located along this line, an offset drive would be required.

The casting was webbed or boxed where rigidity was required. The wall thickness determined by the bearing requirements of several pins was used throughout.

Requirements of other subassemblies dictated support for clamp guides, and depressions for the clamp cylinder, the clamp

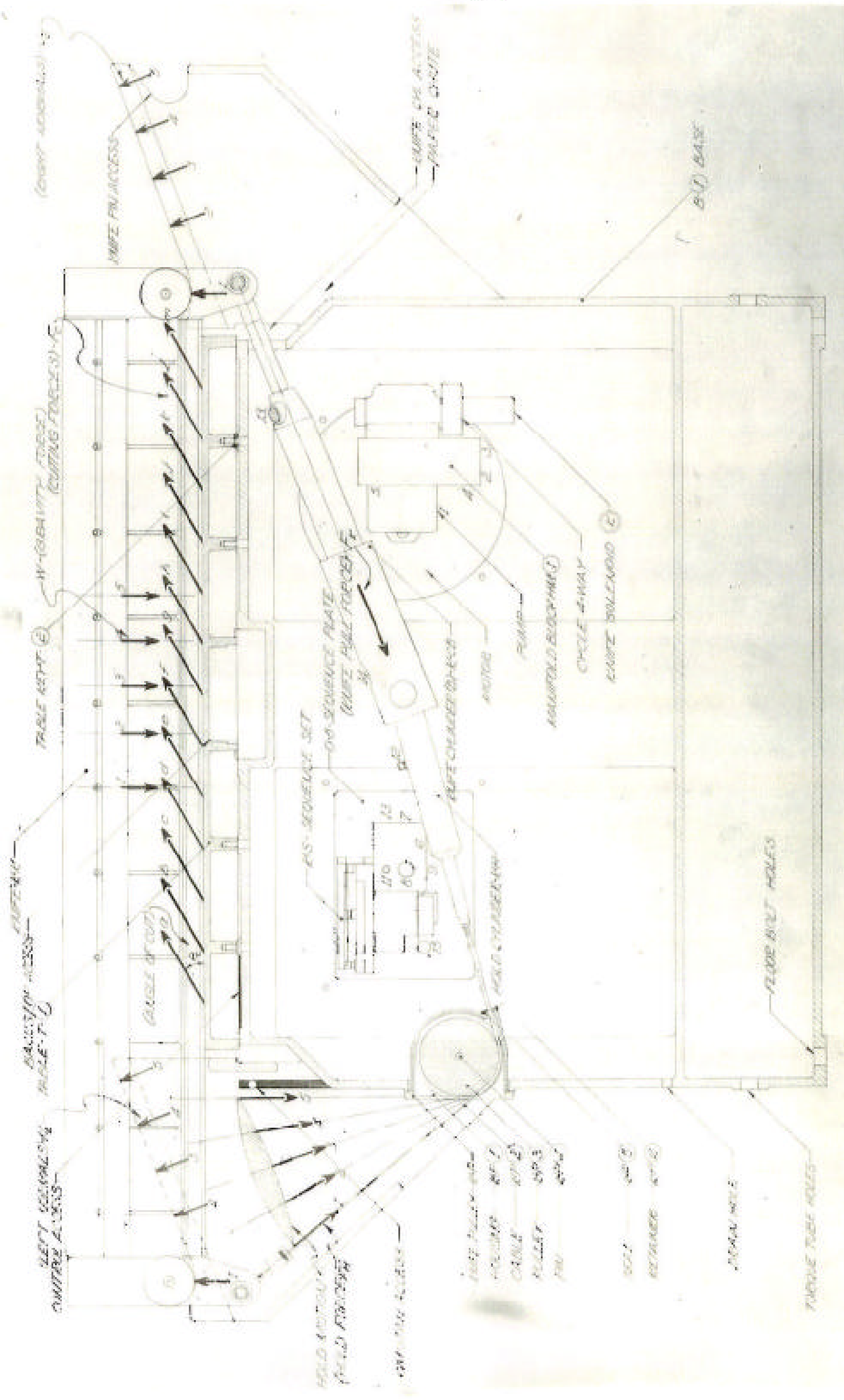


Fig. 9. Assemblies: Hydraulic - H, Knife - K, Knife Pulley - KP, Base - B, Table - T

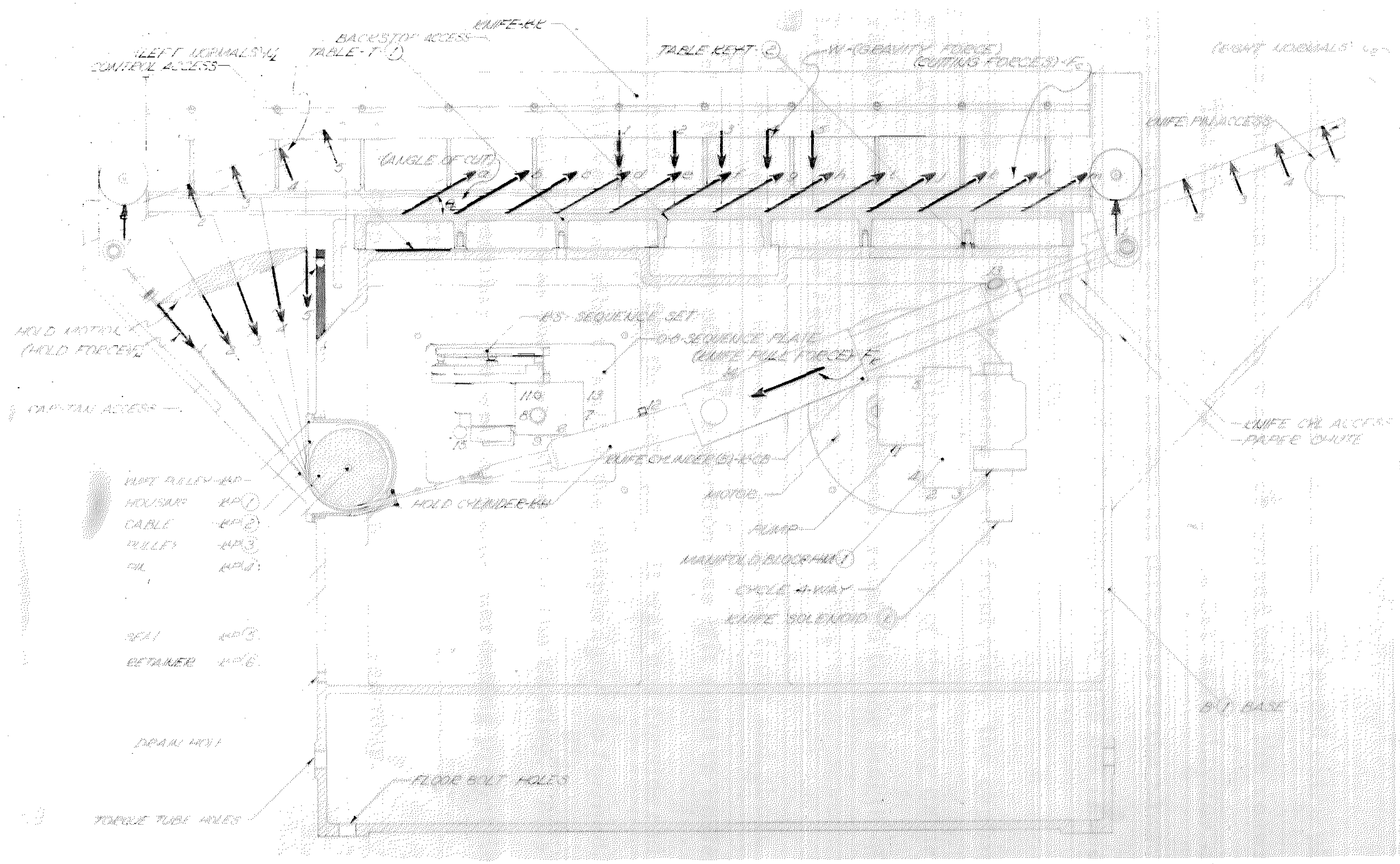


Fig. 9. Assemblies: Hydraulic - H, Knife - K, Knife Pulley - KP, Base - B, Table - T

pull rods, and the leg of the backstop tee. To avoid extra plates and seals, the clamp cylinder was to be housed outside the reservoir.

The Motor assembly, and Sequence Set assembly for clamp pressure control governed the overall thickness of the base. Due to the resultant volume enclosed, the torque tube was not needed as a reservoir.

Down to the vertical sides of the front view, the casting shape was governed by the established geometry of the swing cable and ramps. Torque tube pivot locations, related to the length of the clamp bar, influenced the remainder.

A large access opening was left at the front and above reservoir level. In the center of this, a portion was left solid to stiffen the box and provide bearing for the main pin. It was not considered necessary to install vertical ribs to stiffen the "diaphragm" so formed.

Meehanite was the metal chosen for the casting. This offered strength with ductility, as well as flame-hardening properties for surfaces subject to wear.

This completed the design of the base assembly.

4.3 Hydraulic Assembly - H - Figure 9

Two main subassemblies contain most of the purchased hydraulic components within the reservoir. In the first subassembly a low pressure relief valve and a 4-way valve are attached to the pump by means of a MANIFOLD BLOCK HM (1). The motor carries the pump and bolts to the base.

In the second, the SEQUENCE SET HS subassembly, is built around a standard sequence valve. Together with a high pressure relief valve it is connected to a plate which then bolts to the base. Later a block external to this plate was added to include a 3-way valve to drop the clamp, and ports leading to the backstop and clamp cylinders were added also.

By means of the Sequence Set feature the operator can set the pressure that is applied to the clamp before the knife is powered. Lights equally spaced over the paper are turned on corresponding to the clamp pressure setting required for that length of paper. Eight lights are illuminated in order from the left of the machine. Thus, in order to set the clamp pressure for a certain width of cut, the operator just turns on the lights over the lift to be cut. If eight small switches were added to the foot of the clamp, these could automatically set the clamp pressure as they contact the paper.

Manifold Block - HM - Figure 9

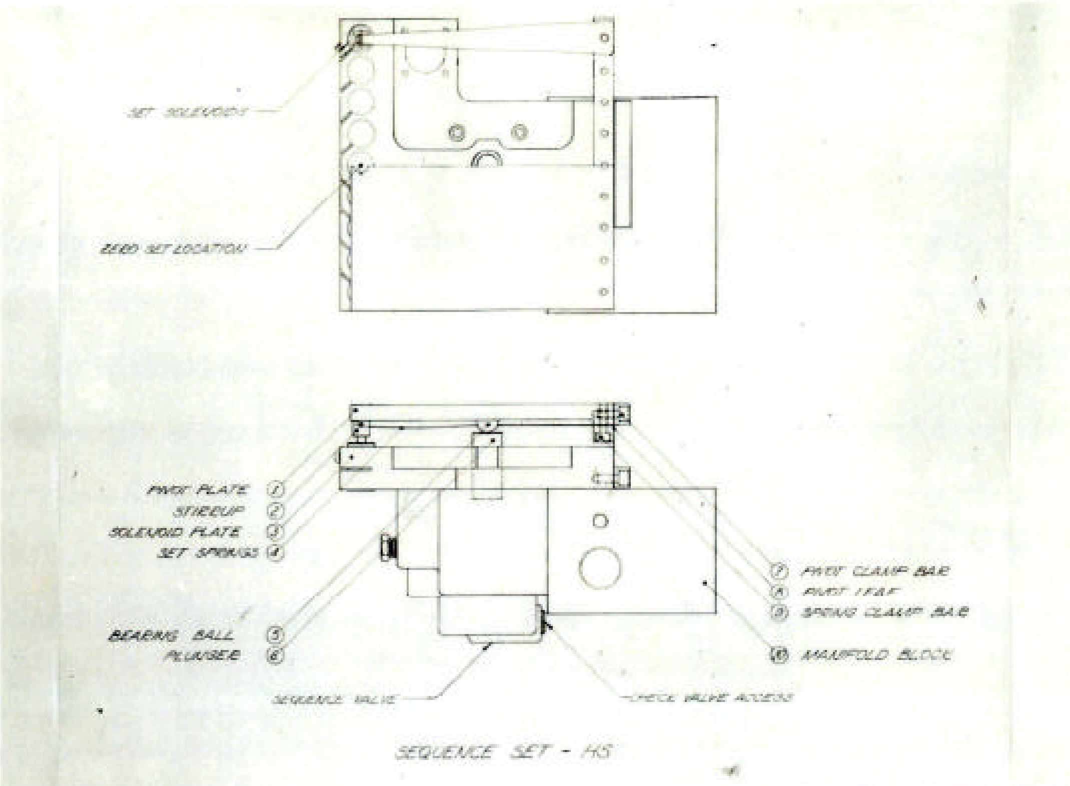
For a larger number of units this block would be cast. It had to be compact yet provide large ports and its length governed the overall thickness of the reservoir casting. The manifold block was designed by J. Ito.

<u>Sequence Set Assembly - HS - Figure 10</u>

The spring and cover plate of a sequence valve were removed to expose the valve plunger.

In place of the spring, an extension PLUNGER (6) was substituted and a PIVOT PLATE (1) on a PIVOT LEAF (8) also was added. Forces applied to the end of this plate are transmitted to the plunger by means of a steel BALL (5). Increase of these forces increases resistance to valve opening and thus increases the clamp pressure prior to allowing flow to the knife cylinder. A spring supplies a permanent small force and eight cantilevered SET SPRINGS (4) supply the additional forces. These are preloaded and attached to the plate, normally moving with it thus applying no force to the plunger. Solenoids can be switched to hold a number of these stationary. In that case their preloaded forces are applied. Since the solenoids only need to operate at zero air-gap they can be physically small yet apply relatively large forces. Clamp pressure can be varied remotely simply by powering various solenoids.

The assembly was designed by D. A. Morelli and B. Auksmann.



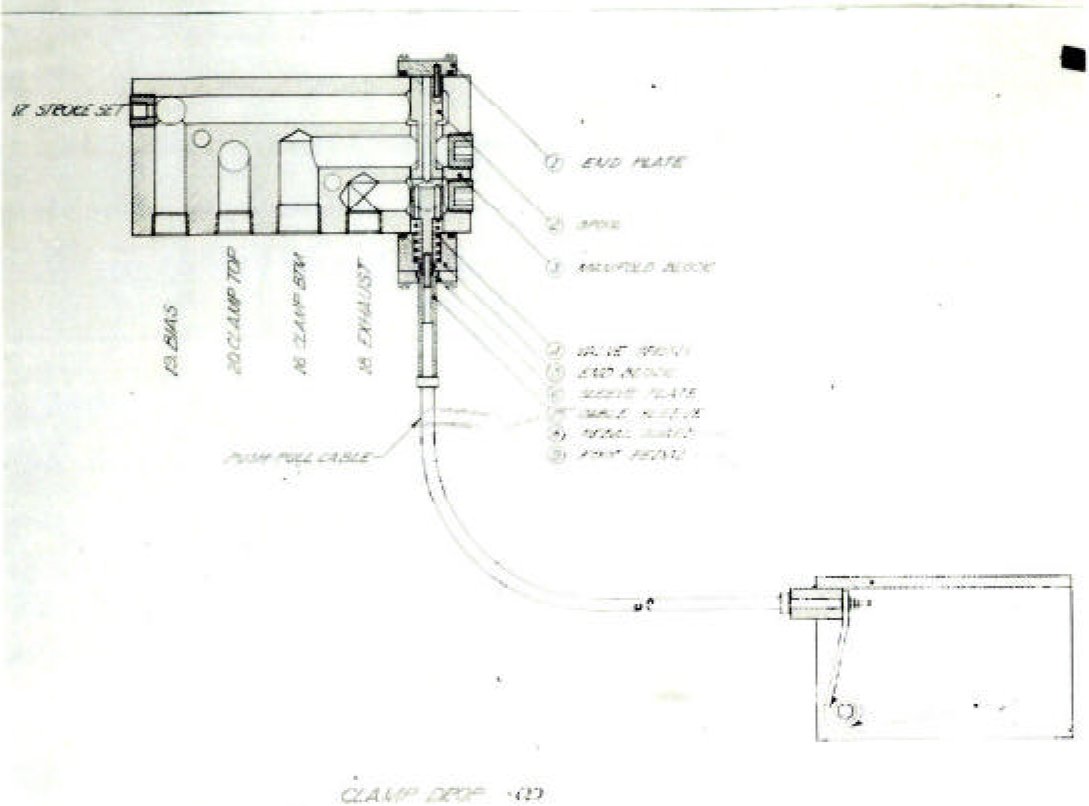
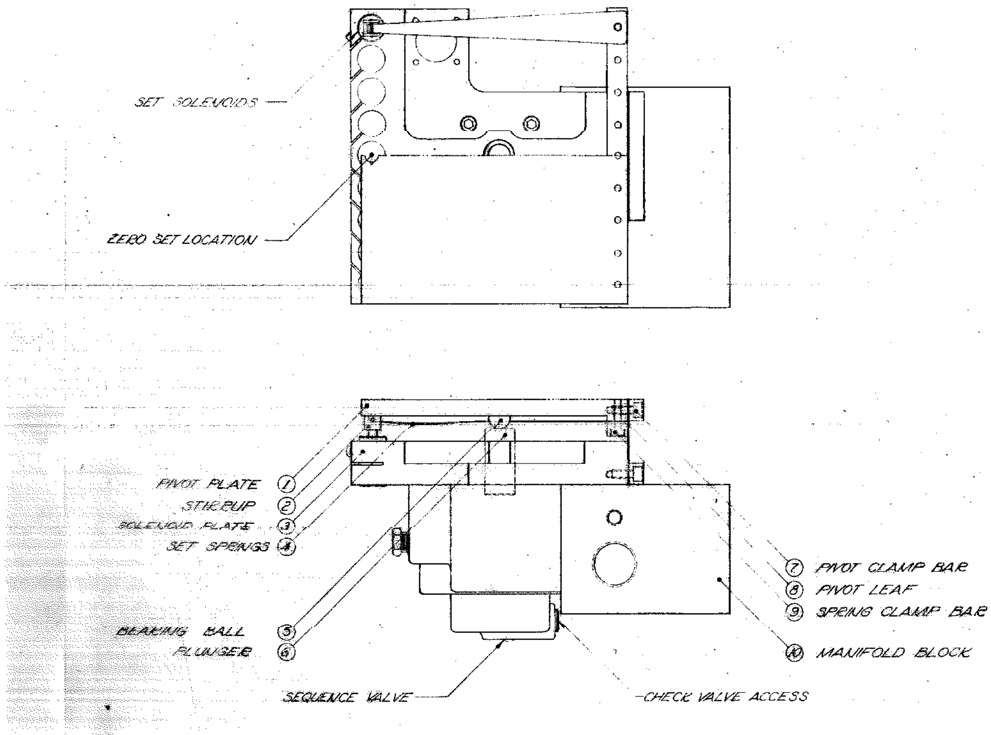
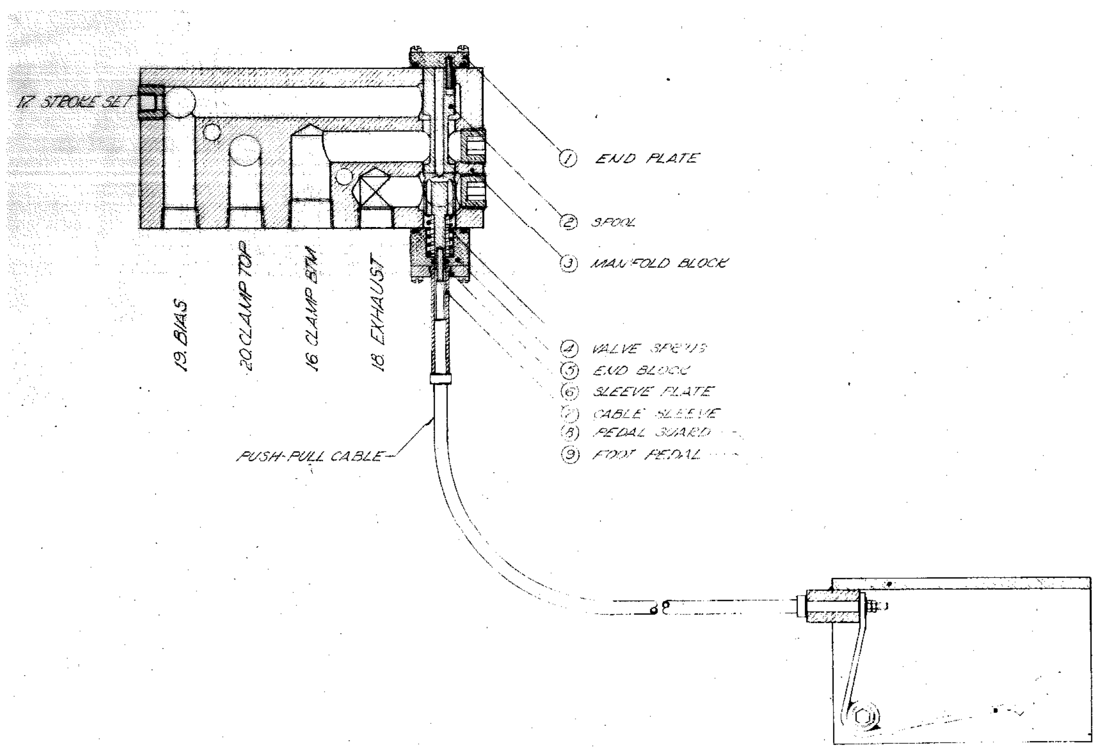


Fig. 10. Assemblies: Sequence Set - HS, and Clamp Drop - OD



SEQUENCE SET - HS



CLAMP DROP -OD

Fig. 10. Assemblies: Sequence Set - HS, and Clamp Drop - OD

4.4 Table - T - Figure 9

The cross-section of the TABLE (1) just in front of the knife line is shown in the Figure 9. It is fundamentally a rectangular meehanite plate with three cast-in transverse beams and eight cast-in longitudinal beams. In width, it is equal to the width of paper cut. The length behind the cutting line is equal to the length (42 in.) of cut that can be made plus the width (4 in.) of the backstop beam used to advance a lift of paper over its surface. The length in front of the cutting line is for working space and is common for the industry (24 in.).

Table width is fixed by the size of lift to be cut. By applying a T square type of backstop drive as shown in the Backstop Assembly of Figure 14 a decrease of length was made possible. If it can be accepted that the reach for the operator, and the backstop beam to advance the paper are necessary, then the length of the table has been minimized. This minimum table length was later used to dictate the maximum allowable length for other assemblies. As a result, a minimum overall length of the machine has been achieved.

For accurate positioning of a lift prior to a cut, the fence and table must be planes forming the corner of a cube with the plane of cut. Although sometimes the backstop plane is tiltable, for special types of cutting, it generally forms a corner of a cube with the fence and the table.

The table serves mainly to position these planes. All machined surfaces of the table are plane, and normal to the machined surfaces that they intersect.

Under the line of cut, four bolts inserted from inside the reservoir secure a transverse beam to eight finished longitudinal pads of the base. These pads are each about six in. long and lie in a plane normal to the plane of cut. Corresponding pads are machined on the longitudinal beams of the underside of the table. Thus when these tie down bolts are secured, the table under the line of cut is a plane. This plane is normal to the plane of cut.

Two key slots are machined on the top of the base and under the two tie-down bolts near the sides of the table. They are parallel to the plane of cut. Two rectangular TABLE KEYS (2) are used to locate these to corresponding slots on the underside of the table. Thus all necessary reference with respect to the plane of cut is established.

The backstop plane is made normal to the table and parallel to the line of cut by means of adjustable cam rollers that follow machined ways under the table center. This is shown in detail on the Backstop Assembly of Figure 14.

The base and the table form a rigid right angle. Since this is supported only by a leg at the center rear of the table, and two points on the base, the table is not subject to torques and therefore does not twist.

Since each of the longitudinal beams is held in a plane in the area under the knife, the plane of the table is largely determined by the stiffness of these beams. The ends of these are connected at the front and rear by transverse beams.

A number of holes for attachment of the leg, panel, fence, and side extensions were added when these parts were determined. The extension tables were bolted both to the base and to the edges of the table in front of the base.

In future models several changes should be made:

1. Less finish should be allowed for warpage of the casting. The 3/8 in. allowed on this casting was excessive and expensive to remove.

2. The top and bottom surfaces should be blanchard ground. Planing required excessive time and some distortion and grooves resulted.

3. The length of the mounting pads should be increased. This will add necessary extra rigidity.

4. More use should be made of the potential of the casting process. It is seen that this table design has the appearance of a weldment.

4.5 Knife Assemblies - K - Figure 9

The plan view of the knife bar had been determined approximately as shown. Dimensions for the main pin, piston rods, and cylinder diameters resulted from an analysis of the knife drive geometry for assumed values of angle of cut θ_c, locations of wheels on the ramps (1 through 5) and locations of resultant cutting forces (a through m) as shown.

Knife Pulley - K P - Figure 9

The diameter of the PULLEY (3) had been established and the pulley was located just above fluid level. Since the CABLE (2) ran close to the wall of the base casting, a thin-walled HOUSING (1) was provided. A cantilevered pivot PIN (4) was thereby avoided as the founder was able to supply aluminum castings with thin walls.

To make a symmetrical core thus requiring only a half core box, a clearance slot required for the cable above was repeated below.

To exclude paper dust, a felt SEAL (5) was used where the cable enters the reservoir. (The paper deflector included in the fence prevents fouling due to scrap falling between the pulley and the cable outside the reservoir.) Mounting bolts are larger than necessary to make most tapped holes in the base similar.

An aircraft type fibre pulley and cable were used.

Hold Cylinder - K H - Figure 11

Placement of the pulley dictated the stroke of the hold cylinder. Since this was small, a SPRING (4) in the cylinder could supply a reasonably constant extra hold force. Apart from modifying the drive characteristics for small cuts, this eliminates slack in the cable when the cycle is off.

ROD (3) and SLEEVE(5) diameters had been fixed by force calculations but the length had to be determined. For a low spring constant, the spring outer diameter was made as large as the sleeve would allow. From the various lengths and wire sizes considered one was chosen which produced the desired spring rate. After preload, this fixed the length of the hold cylinder.

Prior to machining, a layer of bronze was applied to the piston face. Thus the piston tends to score before the walls of the cylinder. A bronze bushing at the END (1) protects the rod seal from side loads, and also reduces scoring of the rod. Only a small welded-on fitting on the sleeve was required due to the low flow of oil.

A pin, put in the hole on the piston rod, can be used to remove spring force from the cable. The cable end has swaged-on threaded ends that screw into the rod. The cable length can be changed by using the wrenching flats on the rod end. A bushed eye on the other end of the cable is a slide fit on a pin at the end of the knife bar.

The diameter of this cylinder was much smaller than that of the knife cylinder which resulted in a simplification. Instead of, for example, providing castings for the lower ends of these cylinders, and

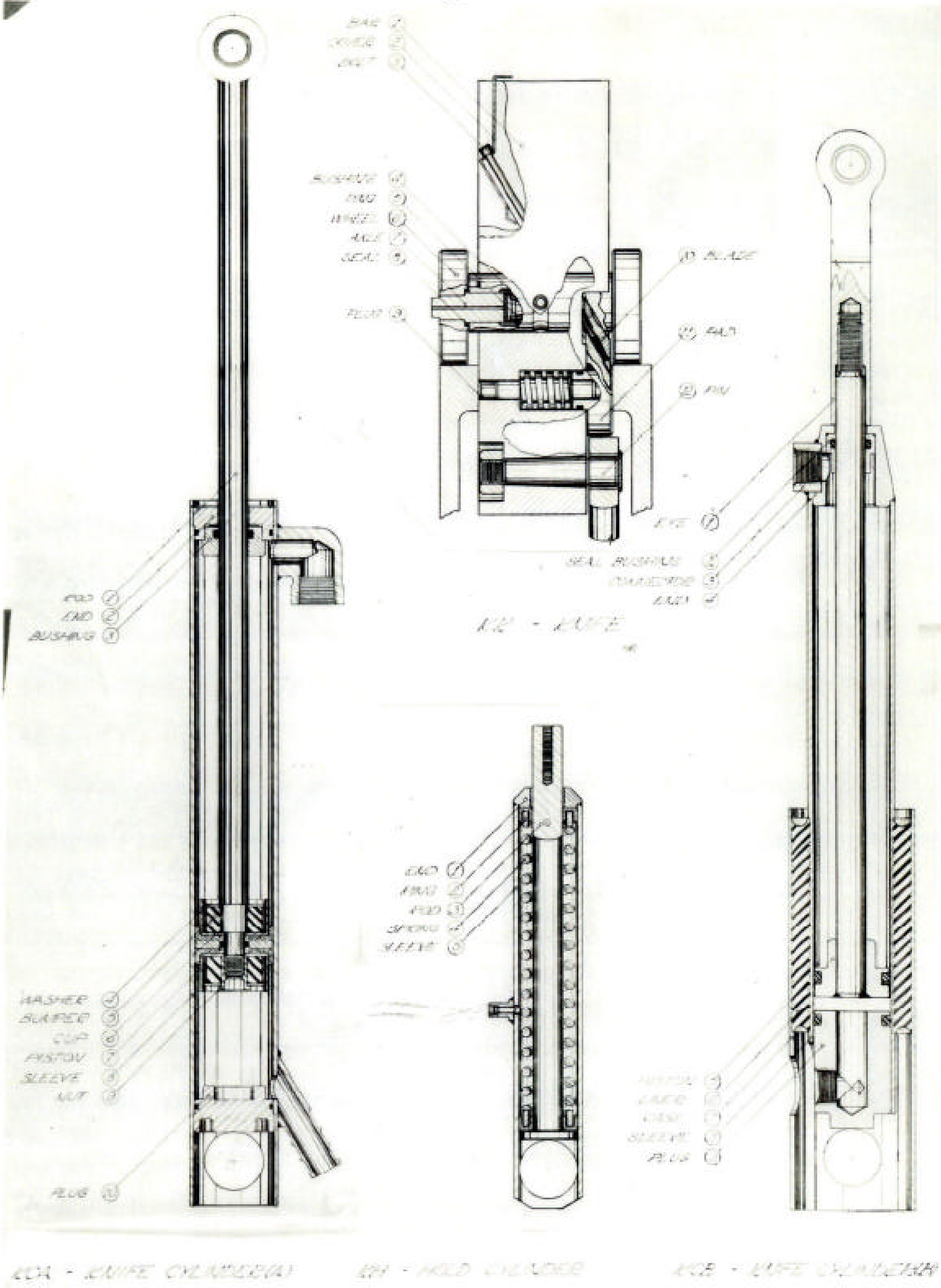


Fig. 11. Knife Subassemblies - KK, KH, KCA, KCB

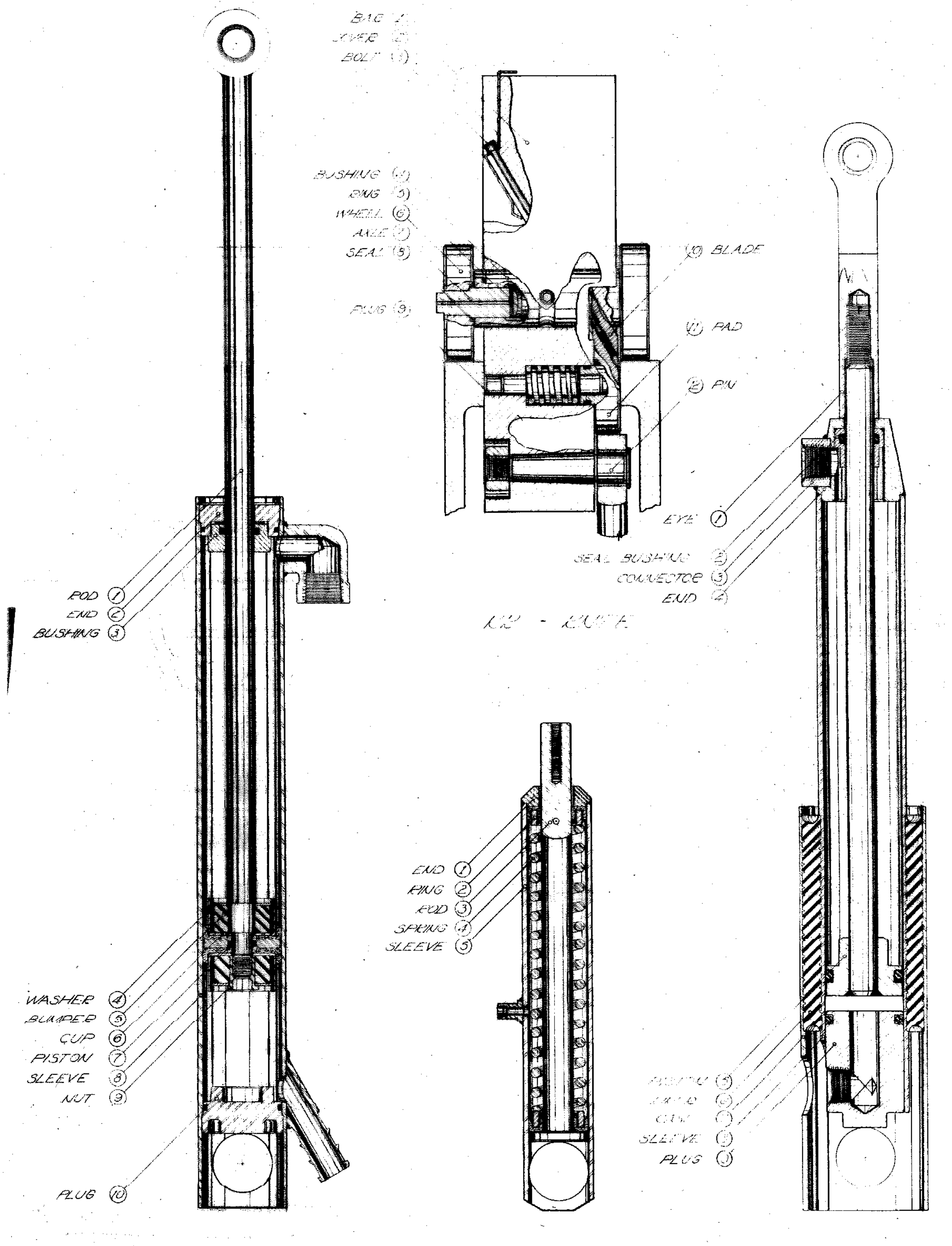


Fig. 11. Knife Subassemblies - KK, KH, KCA, KCB

pivoting these on different supports, a single pin was used. Identical holes bored in the cylinder walls provide bearing surface and serve as pivots. At assembly, the hold cylinder end is positioned inside of that of the knife cylinder. The main pin then supports and pivots both.

The hold cylinder was designed by J. Ito.

Knife Cylinder - KCA - Figure 11

The cylinder designed at this time was similar to that of KCA of Figure 11. It differed from KCA in that the BUSHING (3) and PLUG (10) were longer by 3/8 in., the neoprene BUMPERS (5) and the WASHERS (4) were absent, and the NUT (9) consisted of a simple locknut. In principle both are similar, but neoprene bumpers were added to make KCA when tests of the original resulted in excessive shock.

Because space was limited the central pin which supported the cylinder also was made to secure the PLUG (10). CUPS (6) served as throttle type cushions and as backups for the packings. The cushion length was limited to the overshoot allowed at the bottom of the ramp. Throttle type cushioning action results at the ends of the stroke. As one of the cups slides over one of the tapered cylinders near the end of stroke, the size of the outlet port is gradually reduced to zero.

A long bushing was used as the throttle type seal around the rod, as leakage would be to the reservoir. The leather washer between the bushing and the end cleans the rod.

<u>Knife Bar Assembly - K K - Figure 11</u>

Holes for the eccentric BUSHINGS (4) of the AXLES (7) are bored normal to the cutting plane; the center distance is equal to the distance between the ramps. For equal angular settings of the eccentrics the bar remains parallel to the table.

Steel wheel assemblies are hardened and ground. The WHEELS (6) are press fitted to the axles. Oil is fed to cavities through holes in the ends of the BAR (1) and is carried to the axles by slinger RINGS (5). To exclude dust and contain the oil, plastic dynamic SEALS (8) are used.

The BLADE (10) is diamond shaped and provides two cutting edges. It is installed at a slight angle to give back-clearance. The two faces to be ground are concaved to reduce metal removal on re-grinding. No special fixture is needed for sharpening, as all faces to be ground are parallel, and grinders with magnetic tables are readily available.

A sheet metal COVER (2) acts as a retainer for the mounting bolts and as a fixture for blade removal. The BOLTS (3) are long and small in diameter. Their spring rates are low enabling the bolts, once tightened, to hold even if cutting forces drive the blade further into the groove.

Bronze bushings were used in the ends of the hold cable and the knife cylinder rod. These bushed ends are attached by means of removable hardened steel PINS (12) to the corresponding ends of the knife bar. To prevent slack at the connection, the pins are tapered to fit similar holes in the bar. On the other side, nuts draw the taper for a zero clearance fit. Should the bronze bushings wear, hardened steel ones will be substituted. Motion is slight at these pins.

Two porous bronze PLUGS (9) carry oil to the gibs. One is set in the ground face of the knife bar that directly rides the gib. The other is fixed in the spring-loaded gib PAD (11). Due to this pad the gibs do not have to be spaced precisely. Side loads on the bar are reduced by back-clearance of the knife blade and its parallel cutting motion. Preload of the gib pads exceeds the expected side loads.

To guarantee that the wheels do not leave the ramps, the cutting action was calculated for positive ramp reactions. A reduction in wear results from the fact that these reactions are proportional to the cutting force. With the force closure used the alternative would require permanent positive wheel reactions when no paper was cut, in excess of the maximum negative reaction components for all other cases.

If all four wheels remain on the ramp during descent, their axles should remain normal to the plane of cut. This is subject of course to machining tolerances, but the ramps were made in a single set-up. Since the clearances between the axles and bushings, and bushings and knife bar are slight, the knife bar plane should not tend to "roll" or rotate from the plane of descent. In addition, since the knife pull forces are in the plane of cut they contribute negligible "roll" moment to the bar. Side forces are assumed to act near the edge of the blade. Because of back-clearance and parallel or single-shear cutting their magnitude is assumed to be small. Finally, to determine the cutting plane, the gibs contact guides along a line extended from the cutting edge. Not only does this eliminate moments for the side forces assumed, but it also guarantees that roll does not appreciably displace the line of cut.

For convenience, removal of the blade is from the operator position.

The shape of the knife-bar cross section is determined by rigidity, function, the knife cross-section, the use of long blade bolts, and the simplicity of a one-piece pattern. Metal can be removed if desired by drilling or coring from each end. Strength and flame-hardening properties were provided by casting the bar of meehanite.

4.6 Clamp Assembly - C C - Figure 12

A force of 10,000 lbs was used to design the clamp and cylinder. The cross-section of the CLAMP BAR (5) is a compromise between a closed member for rigidity, and an open member with slots for entry of the backstop. Due to the narrow slots used, they were machined instead of cast. Lateral rigidity was assumed to reduce "draw". A flexible clamp shoe could later be added, so some vertical bow was allowed in the bar. Thus stress due to point loads of foreign objects controlled the vertical characteristic of the bar. Holes in the top of the bar were provided to house the row of lights for the clamp pressure indicating feature. This feature, dependent on sequence valve setting is also referred to as the "Sequence Set".

The hardened and ground PULL BARS (8) bear only on the top of the clamp. Screws and mounting slots act only to preserve the geometry. One side of the bars slides on finished pads on the base, and the opposite slides against the GUIDE PLATES (2, 7). Another side slides on the GIBS (1, 6) to control end motion.

The guide plates are bolted to brackets cast on the base. They determine the plane of descent of the clamp bar, and prevent rotation from that plane. Shims under these can be changed in case of wear.

Dowels in the loose plate gibs are retained in oversized holes of the guide plates. The hardened gibs are ground at assembly to prevent end play of the clamp bar. Other tolerances are relaxed in this way.

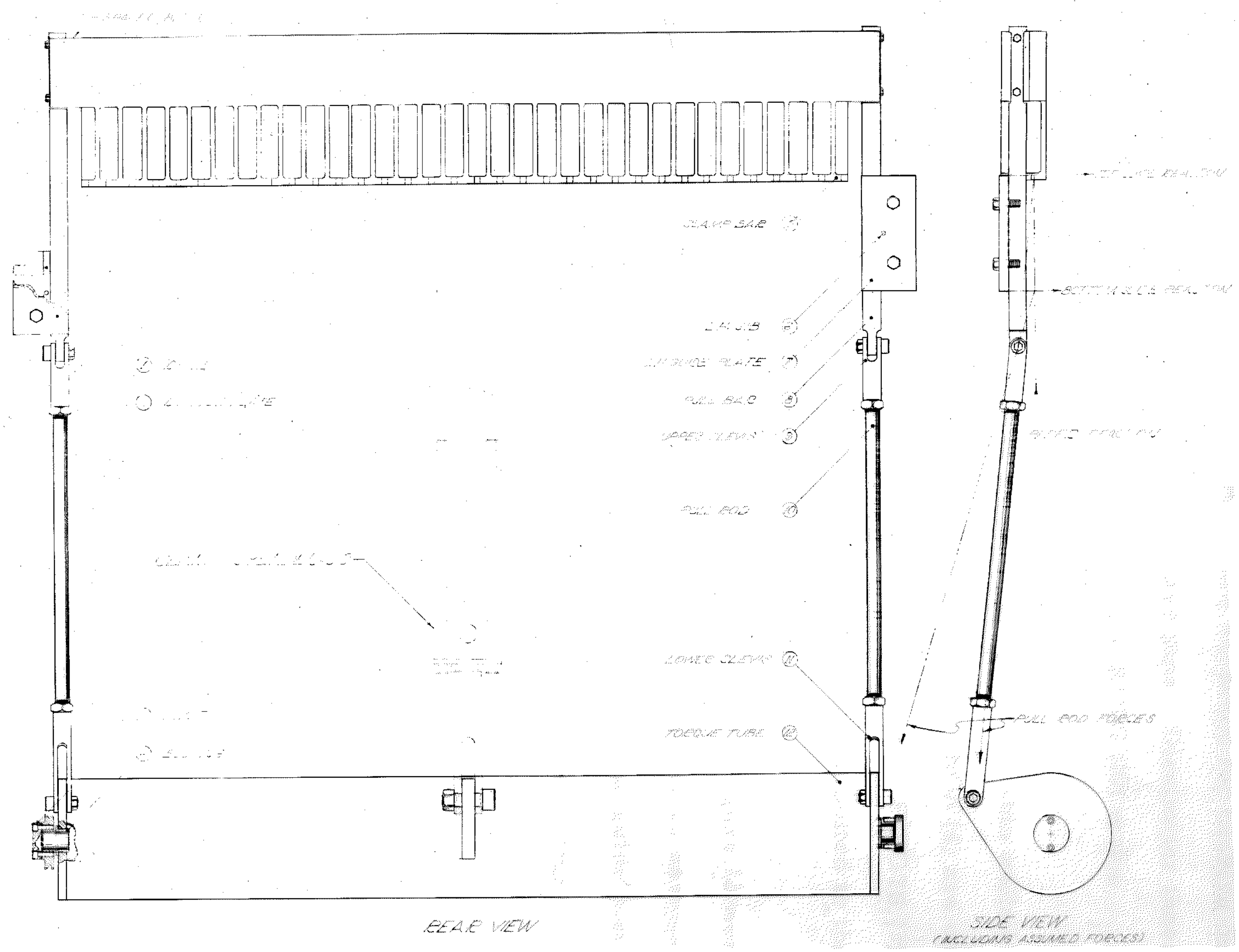


Fig. 12. Clamp Assembly

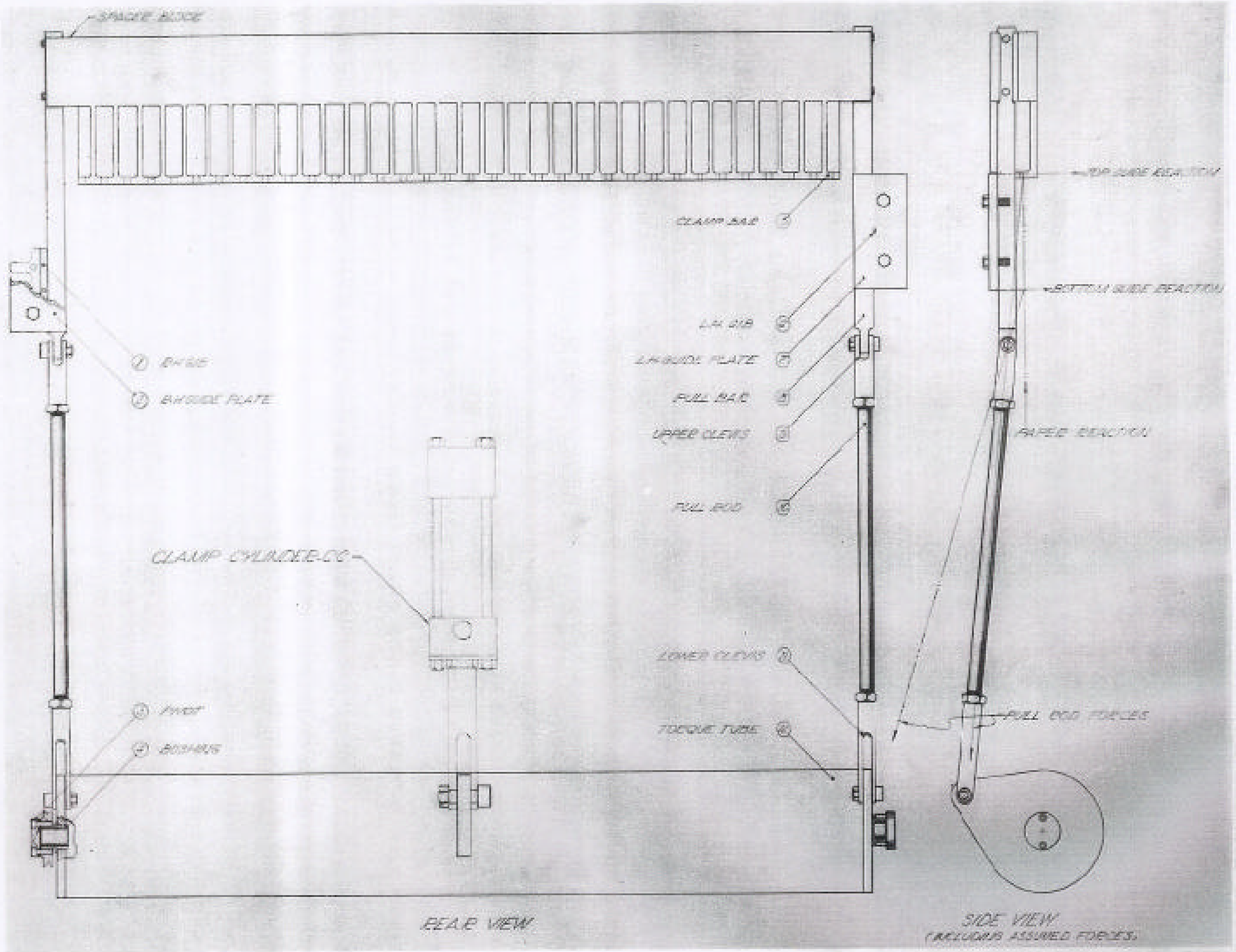


Fig. 12. Clamp Assembly

Upper and lower CLEVISES (9, 11) are hardened for bearing on the pivots. Shoulder bolts supply cheap precise pins. Dissimilar threads on the opposing clevises and ends of the PULL RODS (10) make possible fine adjustment of the clamp height.

The TORQUE TUBE (12) is welded from plate and thin walled tube and holes are bored after fabrication. Bronze BUSHINGS (4) are used, to be replaced by steel if wear proves to be a problem. PIVOTS (3) for the torque tube are hardened and ground. They are bolted to the walls of the base casting and can be removed easily.

To prescribe the side view, forces as shown on Figure 12 were assumed. If the paper reaction is located approximately as assumed, the forces to guide the clamp descent are supplied by the guide plates. Thus no moment is produced and the guide bars do not tend to cock within the guides.

Deflection of the torque tube was determined for various assumptions. Since the critical point loads could occur at any position on the clamp bar, the clamp cylinder was located at the mid-point of the tube.

4.7 Clamp Cylinder - CC - Figure 13

The location of the clamp cylinder had been established as shown on the clamp assembly of Figure 12. The stroke had been determined during the selection of the geometry for that assembly. The upward reaction of the clamp force was located close to the plane of the downward reaction of the paper force. The bore of the SLEEVE (13) had been established by the maximum pressure of the system and the assumed 10,000 lb clamp force.

Commercial cylinders were examined. These were expensive, bulky, and for attachment would require changes to the head end and to the rod. The availability of suitable commercial cylinders was not allowed to dictate the functions or forms of the machine.

A possible custom cylinder was considered. A sketch indicated that a "short-stroke" feature could be included. This was a means whereby the operator could set the upward limit of the clamp travel just above the lift to be clamped, or, if the clamp were used as a punch-press, to the height of the die-set. Since the flow required for the cycle would be reduced in both cases, increased cycle speed would result.

Another advantage of the custom cylinder was that it would be possible to include in the assembly a needle valve for the adjustment of stroke. This would reduce external plumbing connections. It was based on the idea of mounting the clamp cylinder to the main PIN (3) along with the knife and hold cylinders.

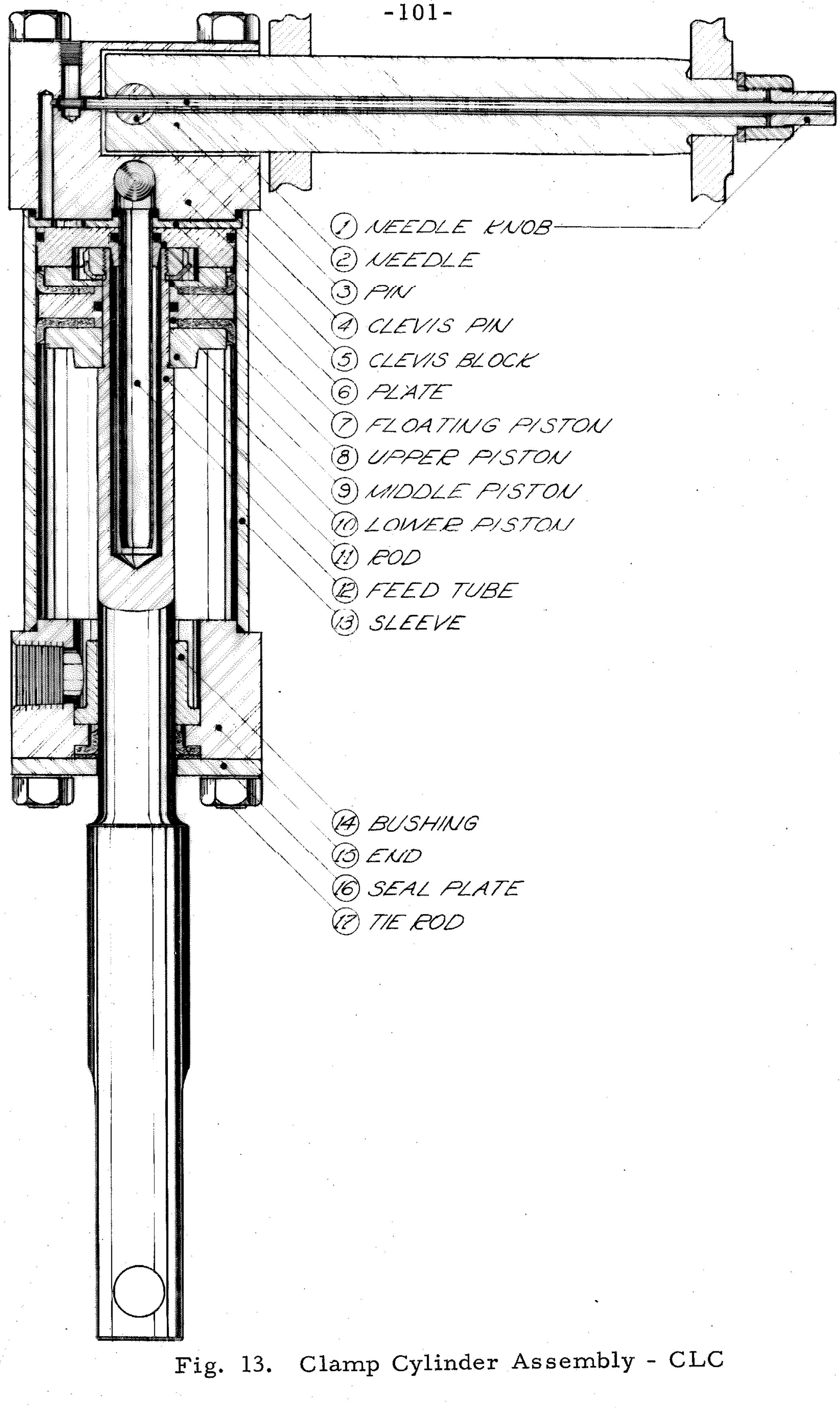


Fig. 13. Clamp Cylinder Assembly - CLC

Another possibility was to include the "clamp-drop" valve in this assembly also. This was the feature that allowed the operator to drop the clamp under gravity prior to starting the cycle. The feature is commonly used to gage a line to be cut along the edge of the clamp or to apply slight pressure at zero speed to lifts that would be disturbed if struck by the moving clamp. Due to the lack of a successful detail for the space available, however, this valve was relegated to its final position on the outside of the sequence set assembly.

The clamp cylinder assembly that resulted is shown in Figure 13. The short-stroke feature consists of the FEED-TUBE (12), the FLOATING PISTON (7) and the PLATE (6) and a hole in the ROD (11). In operation, bias pressure oil, available at the needle valve, is fed to the cavity above the floating piston. Then the valve is closed. This lowers the floating piston, then locks it against upward thrust. The new position serves as the upper limit for the main piston consisting of UPPER, MIDDLE and LOWER (8, 9, 10) portions. This causes the limited upstroke or short-stroke of the clamp.

To raise the upper limit, the needle valve must be opened. Bias pressure is once again applied to the top of the floating piston. At the bottom of the descent stroke of the cycle high pressure oil is applied between the main piston and the floating piston. This tends to force the floating piston up. A number of machine cycles with the needle valve open are needed to raise the setting. The hydraulic circuit of Figure 8 shows how this is done. To retain any setting attained, the valve is closed.

A standard needle valve includes at least a valve stem with the needle on the end, threads to vary the setting of the orifice, and a seat for the needle. By including the valve as an integral feature, these same parts were made to serve other functions. The extension of the stem or NEEDLE (2) is used to retain the CLEVIS PIN (4) and by means of the NEEDLE KNOB (1) serves as an operation control at the front of the machine; the adjustment threads are used also to retain the main pin. The length of the needle and the small rotation of the cylinder are such that no articulation is needed for the connection. Instead, the needle is designed to flex.

4.8 Backstop Assembly - B P - Figure 14

The moving backstop is used to advance the lift along the table and position it prior to a cut. The stationary backstop is used to supply a reference distance and plane so that the operator can position a lift prior to a cut. The backstop operates within the corner of a cube formed by the plane of cut, the fence, and the plane table beneath.

It was to be designed so that its front plane would remain normal to the plane of the table and at right angles to the edge. Assuming that the table was accurately positioned, the backstop plane would be parallel to the plane of cut.

The backstop was to advance all lift sizes that could be cut by the machine, including a single sheet. In addition it would position them to make a minimum final cut of one inch. To accomplish the latter fingers on the backgage are included in a standard fashion. These are such that they advance into slots in the foot of the clamp bar.

A number of former decisions had influenced the design of the backstop assembly. The drive and squaring functions would be under the table. The drive would be a hydraulic cylinder and to reduce the overall table length, it would be a doubler type of cylinder. The necessary squaring length and connections to the two ends of the backstop bar dictated a tee shape for the drive. A laminar connection was to be used at each side to connect the drive below to the backstop above the table. Ways under the center of the table provided two full length surfaces parallel to the table surface, and two parallel to the

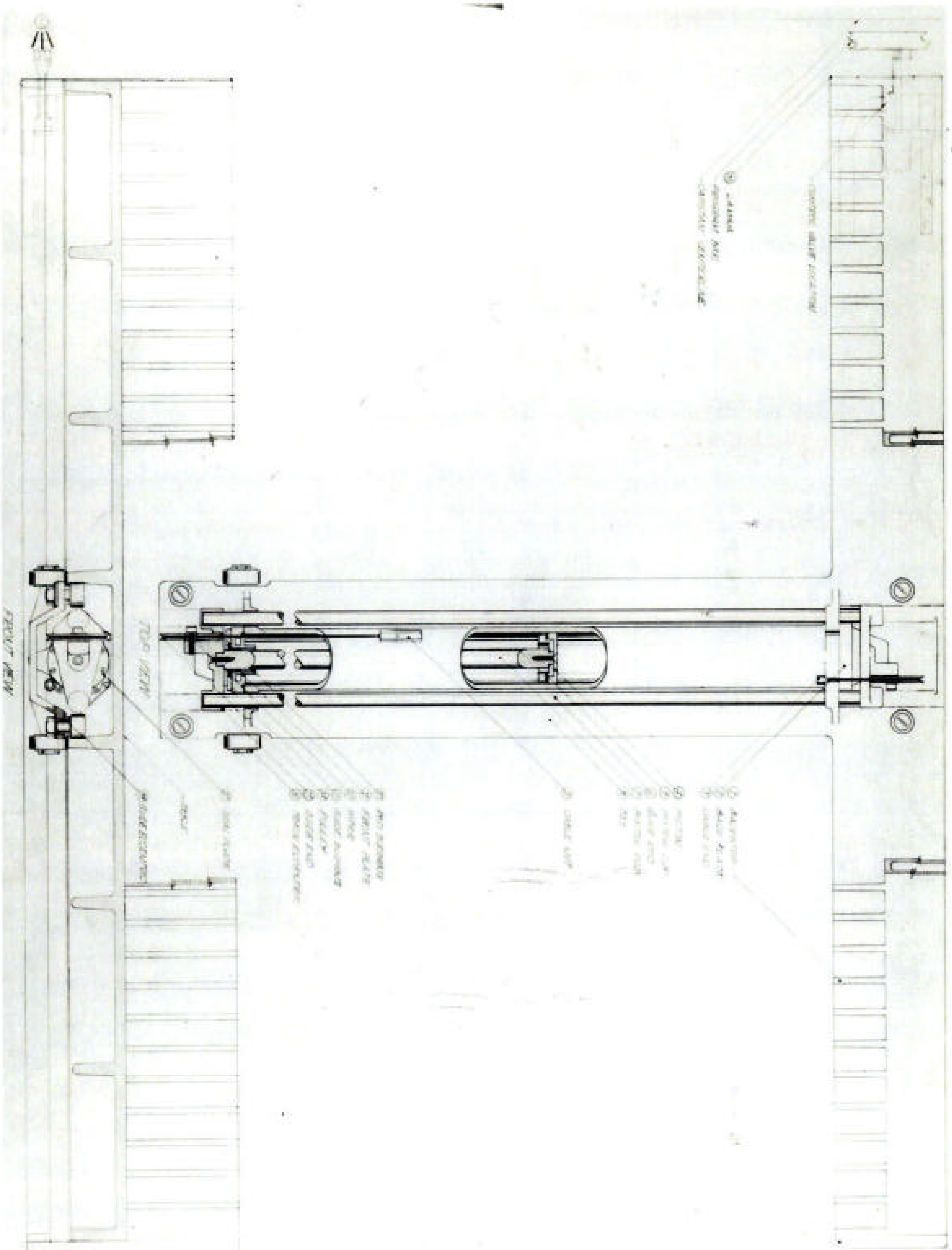


Fig. 14. Backstop Assembly - BP

edges of the table. Cam rollers could ride these for squaring in both of the required directions. The leg of the tee-shaped drive could be used as the cylinder and thus would be a rigid member. The leg was to protrude toward the front of the machine, thus using the "reach" space available in front of the knife line. Others put the squaring leg behind the backstop, thus adding appreciable table length. Plumbing components that could not be included in the tee would be attached to it under the center rear portion. Examination of the details of Figure 14 will reveal how this was done.

The doubler feature is accomplished by means of two PULLEYS (14) and cables. The pulleys are attached to a frame made up of two GUIDE RODS (6) and two identical cast GUIDE ENDS (15). The cables are rigged so that the resultant moment force-closes the two cam rolls of the TRACK ECCENTRICS (16) on the horizontal track. The bushing track eccentrics are used to adjust the cam rolls and thus determine the angle between the plane of the backstop and the plane of the table. The moment produced by the offset cable tensions also tends to rotate the assembly clockwise. This loads only those rolls of the two GUIDE ECCENTRICS (18) which normally are loaded due to advancing lifts situated near the fence. These two rolls suffice for adjusting the backstop parallel with the line of cut. The two others are for accidental loads. CABLE NUTS (9) are used to tighten the cables to screws located in clearance holes at the ends of the table.

The castings are aluminum, to minimize the weight.

4.9 Control Assembly - C - Figure 15

The valve shown mounted on the left end of the backstop tee controls the speed of the backstop. It is shown in detail in Figure 17. For purposes of the control assembly, it is sufficient to note that the PULLEY PLATE (4) is pivoted on a shoulder screw that passes through the tee and fastens to the body of the valve. The CONTROL PIN (3) passes through a slot in the tee and closes the valve as the plate is rotated clockwise. This reduces the speed of the backstop. Counter clockwise rotation of this plate allows an imbalance within the valve to cause it to open.

Functions of the assembly that were used to determine the details shown are:

1. Provide a means by which the operator at the front of the machine can obtain an accurate reading of the position of the backstop.

The READOUT TAPE (11) is an accurately graduated steel tape. The TAPE ECCENTRIC (8) NUT, and WASHER allow a point on the tape to be fastened to the backstop and adjusted without distorting the tape. The tape is wound around a TAPE PULLEY (7) for storage when not fully extended. Since the tape will sag, and cause an error in the reading, it is stretched so that at no position is the error greater than .0015 in. The necessary varying force is applied by a TAPE SPRING (20). One end of this is attached to the PIN (18) of the pulley, and the other end is held stationary by the HOLD SCREW (19). The hold screw is removable to permit assembly of the device. A HOUSING (14) is supplied to guarantee that the point of observation is

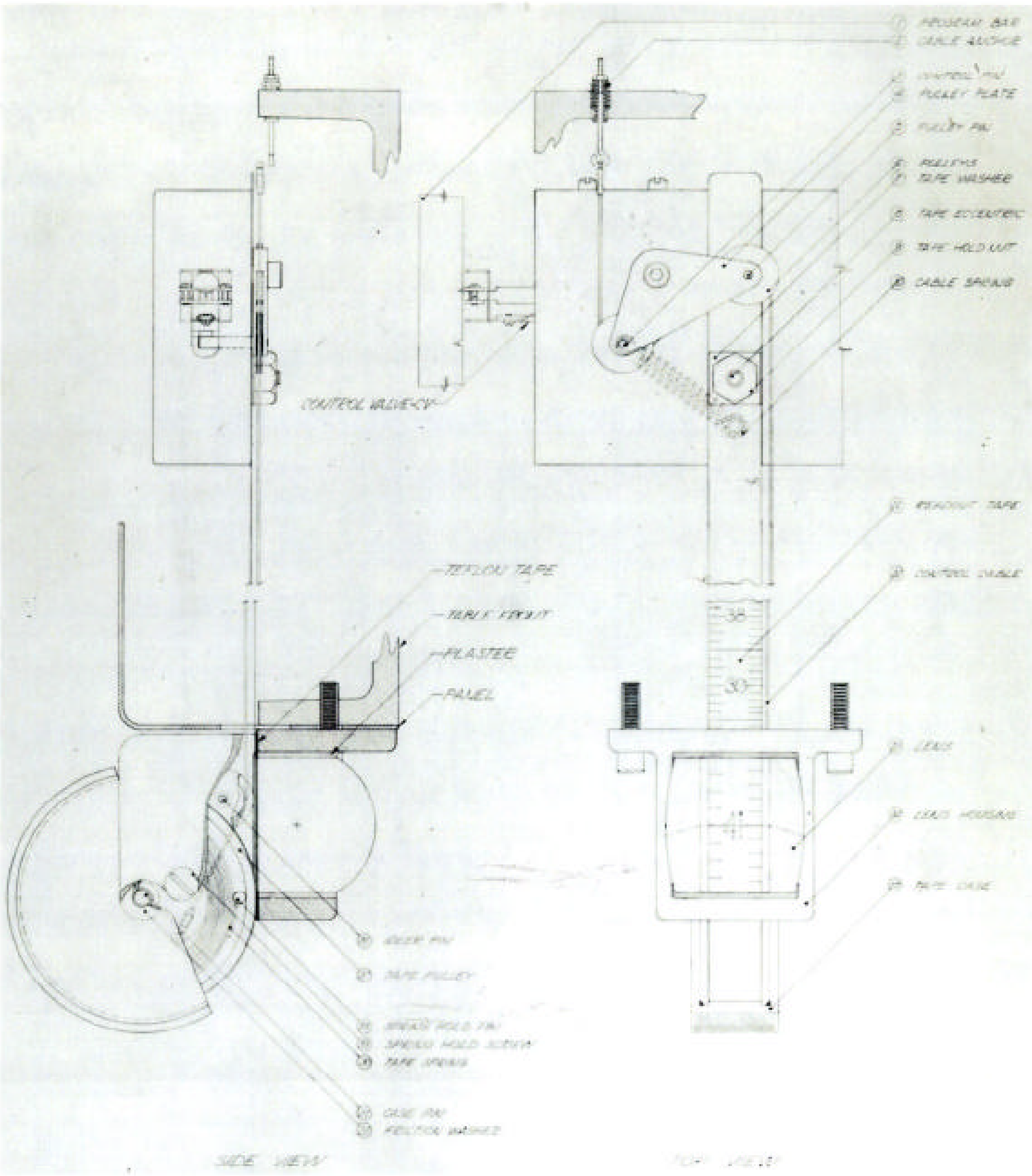


Fig. 15. Control Assembly - C

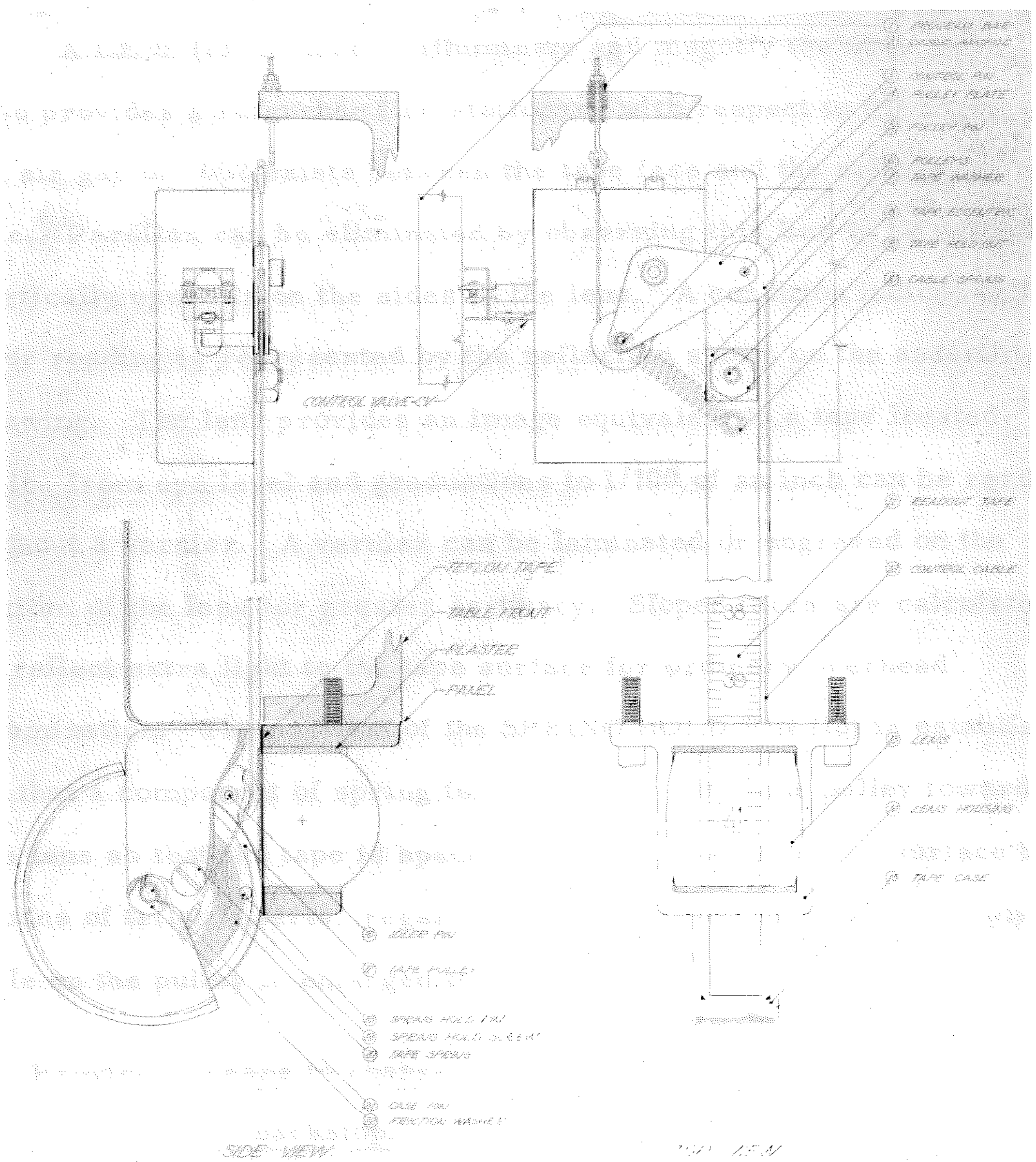


Fig. 15. Control Assembly - C

fixed with respect to the table, and to support and pivot the CASE PIN (21). The CASE (15) protects the spring and tape from damage.

A LENS (13) is used to illuminate and magnify the area read. It also provides a reference line stationary with respect to the table. An air gap of .002 exists between the tape face and the reference line. Parallax can be eliminated by observing this line to be reflected vertically upwards on the sides of the lens. A condition indicating a poor reading is represented by the reflection shown on the assembly drawing. The lens provides an image equivalent to a tape located 10 in. from eye level and graduations to 1/100 of an inch can be read without a vernier. A vernier can be laminated or engraved on the bottom of the lens for greater accuracy. Sloped faces are calculated to reflect extra light to the tape surface for ordinary overhead illumination. The position of the SPRING HOLD PIN (18) is established so that a component of spring force tends to raise the pulley towards the lens so that the tape is spaced .002 ins. from the lens surface by means of teflon inserts, regardless of the length unwound. The pivot hole on the pulley is enlarged to allow this motion.

2. Provide a means to change position of the control valve attached to the moving backstop.

A sensing lever or arm on the control valve is hinged to swing toward the front of the machine. Internal imbalance in the valve forces this open, increasing the backstop speed, until the arm strikes a mechanical stop. Such a stop is provided by CONTROL PIN (3), attached to the PULLEY PLATE (4). The pulley plate is pivoted so that clockwise rotation decreases backstop speed. Counterclockwise rotation allows a speed increase, but does not guarantee it. Because the control pin is not attached to the lever on the valve, it only acts positively to decrease speed. Thus a given setting of the pin corresponding to reduced speed does not prevent the function of the automatic control valve, it only limits the maximum speed attained, thus allowing the maximum speed of the backstop to be adjusted easily.

Manual control of the backstop speed is by means of the rotation of the pulley plate. Two PULLEYS (6) pivoted on PIN (5) on the plate take up slack in a CONTROL CABLE (12). Cable tension is due to a counterclockwise torque applied by CABLE SPRING (10). As the cable slack is changed, by the motion of a lever consisting of the tape case to which the cable is attached, the pulley plate rotates. FRICTION WASHERS (22) allow the tape case and thus the valve to be set in any given position, or continuously varied throughout the stroke. An IDLER PIN (16) is included to support another pulley for directing the cable to the radius of arc needed for full valve travel. A CABLE ANCHOR (2), limits the operator force applied to the components of the control. It includes a spring preloaded with a greater force than the cable tension required to close the valve. After the case is moved sufficiently to close the valve, some extra motion and force is absorbed at that cable anchor thus allowing lighter components than would be necessary otherwise. Further motion of the case is limited by the shape of the housing.

4.10 Fence Assembly - F - Figure 16

The main functions of the fence assembly and the reasons for the details shown are:

1. To square the edge of the lift with the knife line and the plane of the table.

   The maximum size of lift to be cut established the basic outline of the FENCE (3). The main vertical stiffness for squaring with respect to the table plane is the vertical rib to the rear of the machine and the boss near the front. Longitudinal ribs offer lateral strength and stiffness, for squaring with respect to the knife line.

2. To eliminate the wedge-shaped opening below the left of the knife bar.

   The paper deflector consisting of the PLATE (16) and ARM (2) accomplishes this. A spring supplies an upward counter-balancing force to hold the plate against a flat portion of the blade as the blade moves. Geometry was established so that the large friction coefficient of cast iron could be used to slide with no binding. The arm preserves geometry only, and is adjustable to relax length tolerances. The arm is as long as possible and the plate pivots on a point on the knife blade in order to reduce angular change and thus the area of the opening. With this design, no more than a 1/16 in. gap resulted between the sloping surface of the moving knife bar and the sloping edge of the plate that moves with it.

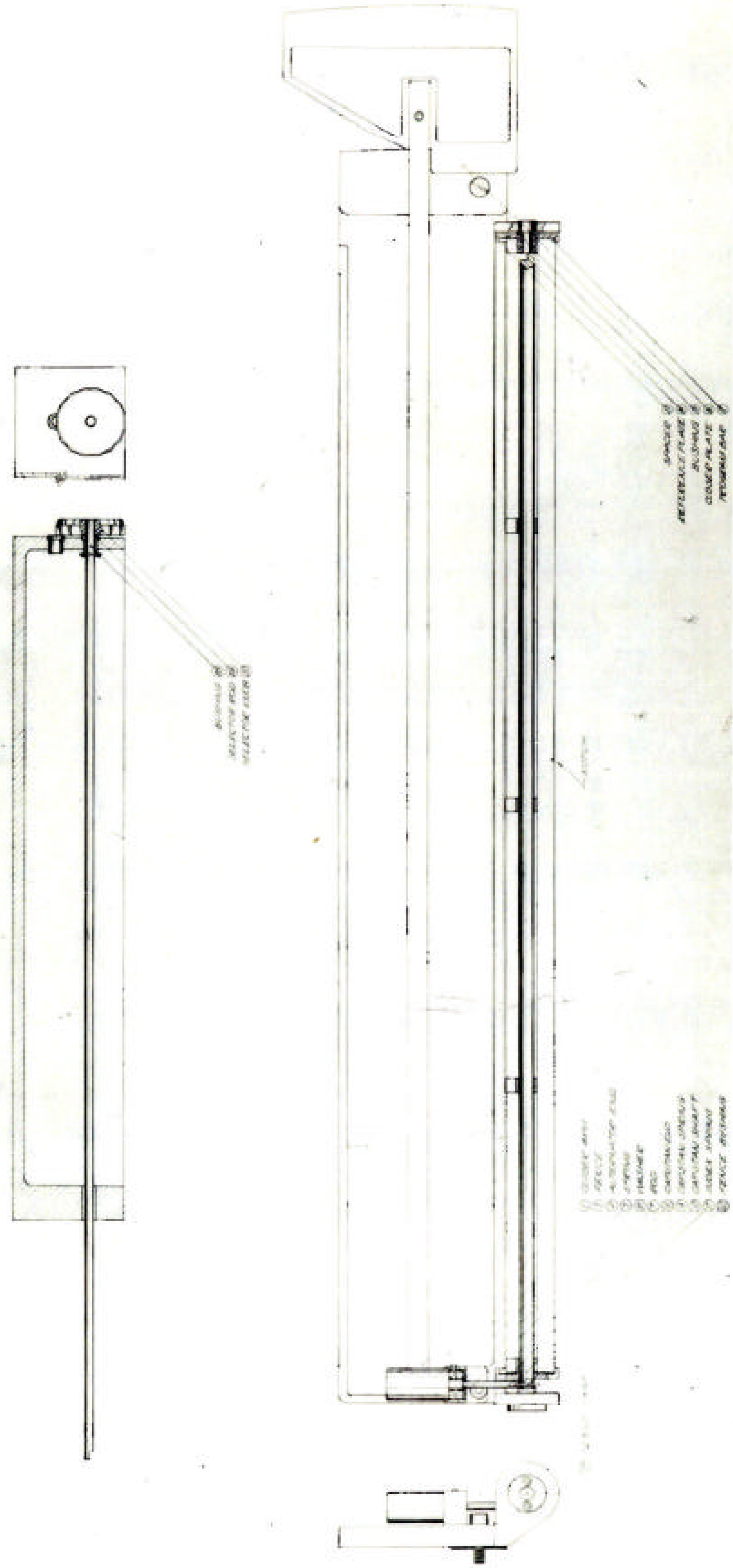


Fig. 16. Fence Assembly - F

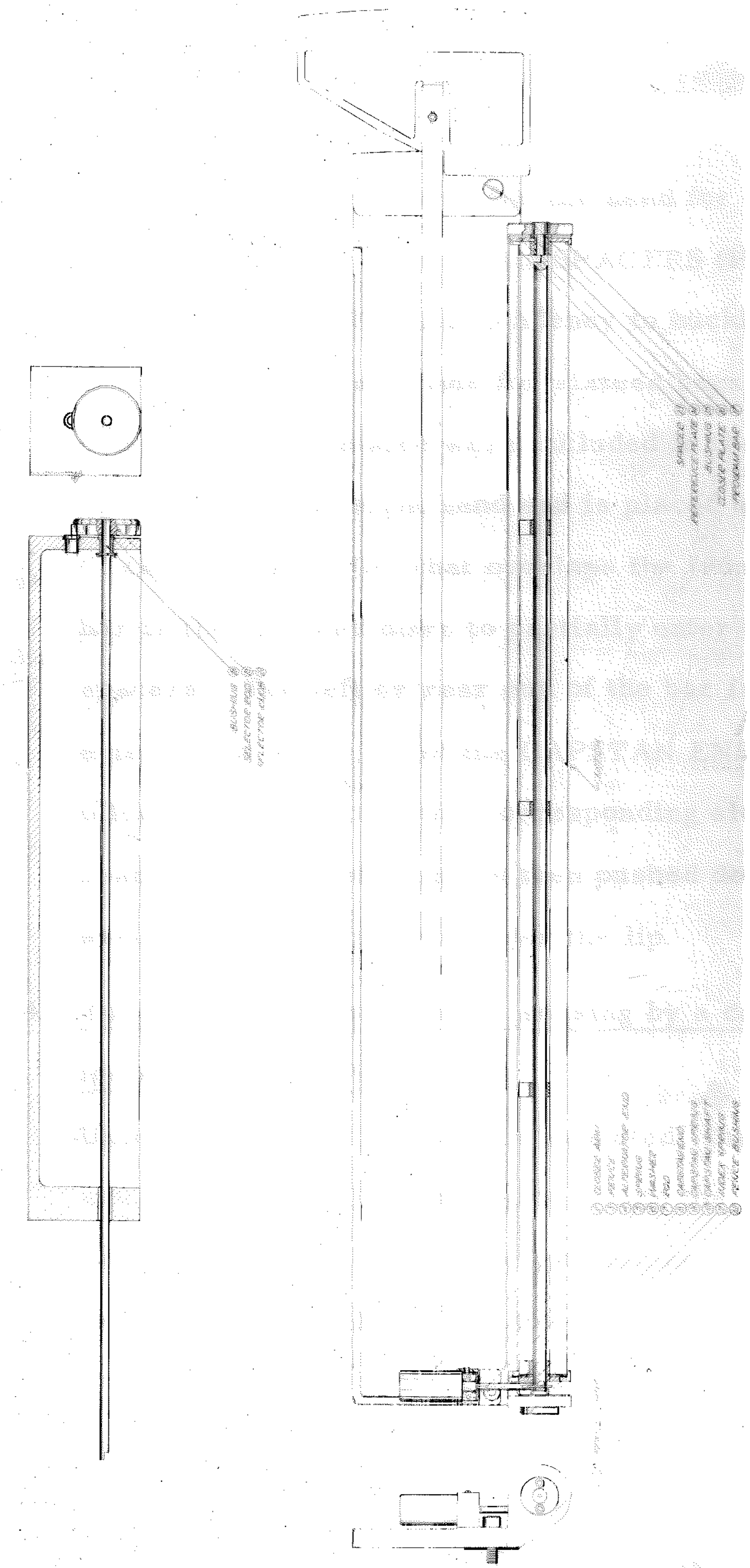


Fig. 16. Fence Assembly - F

3. <u>To provide removable program bars, radially spaced on a drum.</u>

Calculated buckling loads and the need for lateral support determined the five aluminum SPACERS (13) shrunk on a central SHAFT (10). The tendency to buckle is caused by valve sensor loads. Slots for sixteen bars wide enough for programs on each side were included in the spacers. To insert a bar, the right hand end is placed under the lip of the CLOSER PLATE (16) that overlaps the front spacer. The bar is then pivoted down to partially enter the three next spacers. The left or rear end of the bar is then twisted to enter a slot in the lip of the CAPSTAN END (8) which is offset with respect to the corresponding slot in the last spacer. The twisted end is then pushed downwards until it springs back and is held under the lip.

4. <u>To locate program bars for sensing by a control valve, and for manual selection.</u>

Unless accidental loads were expected, the valve and sensor could be small. This helped to locate the valve body inside the tee, with only the sensor protruding. This allowed locating the capstan so that the sensor riding on a program bar was shielded by the capstan from accidental loads. Manual selection is done by means of a rotatable KNOB (20) near operator position. To prevent errors in indexing following removal of the capstan, the SELECTOR ROD (19) can be inserted to the capstan only one way.

5. <u>To provide for indexing of programs and automatic alternating indexing of any two adjacent programs.</u>

   Indexing is provided by means of steel balls forced by the INDEX SPRING (11) to detent in the sixteen existing slots of the CAPSTAN END (8). A solenoid returned by a SPRING (5) can be used to alternately locate one point on the INDEX SPRING (11) to two locations. This corresponds to alternate indexing of two adjacent programs. The circuit that controls this is described in "Control Circuits - Electrical" and provides for two solenoids instead of one solenoid and a spring, as the spring solenoid system shown required too large a solenoid.

6. <u>To accurately locate programs with respect to the table and the knife line.</u>

   This is done by using the front end of the bars as a reference. The INDEX SPRING (11) forces the capstan against the fence which is dowelled to the table thus establishing the reference plate at a fixed distance from the knife line. The combined effect of the Belleville type CAPSTAN SPRING (9) and the sensor force is to push the reference end of the bars against this REFERENCE PLATE (14), thus completing the location of the bars.

7. <u>To provide for removal of the capstan as a unit.</u>

Loosening of two screws in the FENCE BUSHING (12) and a slight forward motion of the SELECTOR ROD (19) permits this removal.

4. 11 Backstop Control and Control Valve Assembly - C V

The control of the backstop depends on the functions of a number of different assemblies, and to describe each individually would require frequent repetition. The following is intended to relate associated assemblies with the operation of the backstop. Any numbered reference to specific details refers to the Control Valve Assembly of Figure 17, unless stated otherwise.

Most of the essentials of the problem have been stated in Section III C.

A. Hydraulic Backstop Circuit - Meter-Out.

The hydraulic backstop circuit used is shown as the Effective Backstop Circuit in Figure 8. This performs the same function as the portion of the Complete Existing Circuit shown mounted on the backstop. The increased complexity of the latter is due to substitution of available components for a satisfactory 3-way valve which was not available. To describe the control of the backstop it suffices to refer to the effective circuit.

The small orifice connecting the two ends of the cylinder represents a calculated leak. Its effect on the circuit is ignored, except for cases where it is comparable to the leak in the control orifice.

Direction.

The position of the 3-way valve determines whether the backstop is in the advance mode or the reverse or runback mode. Each mode represents the direction in which the backstop tends to move.

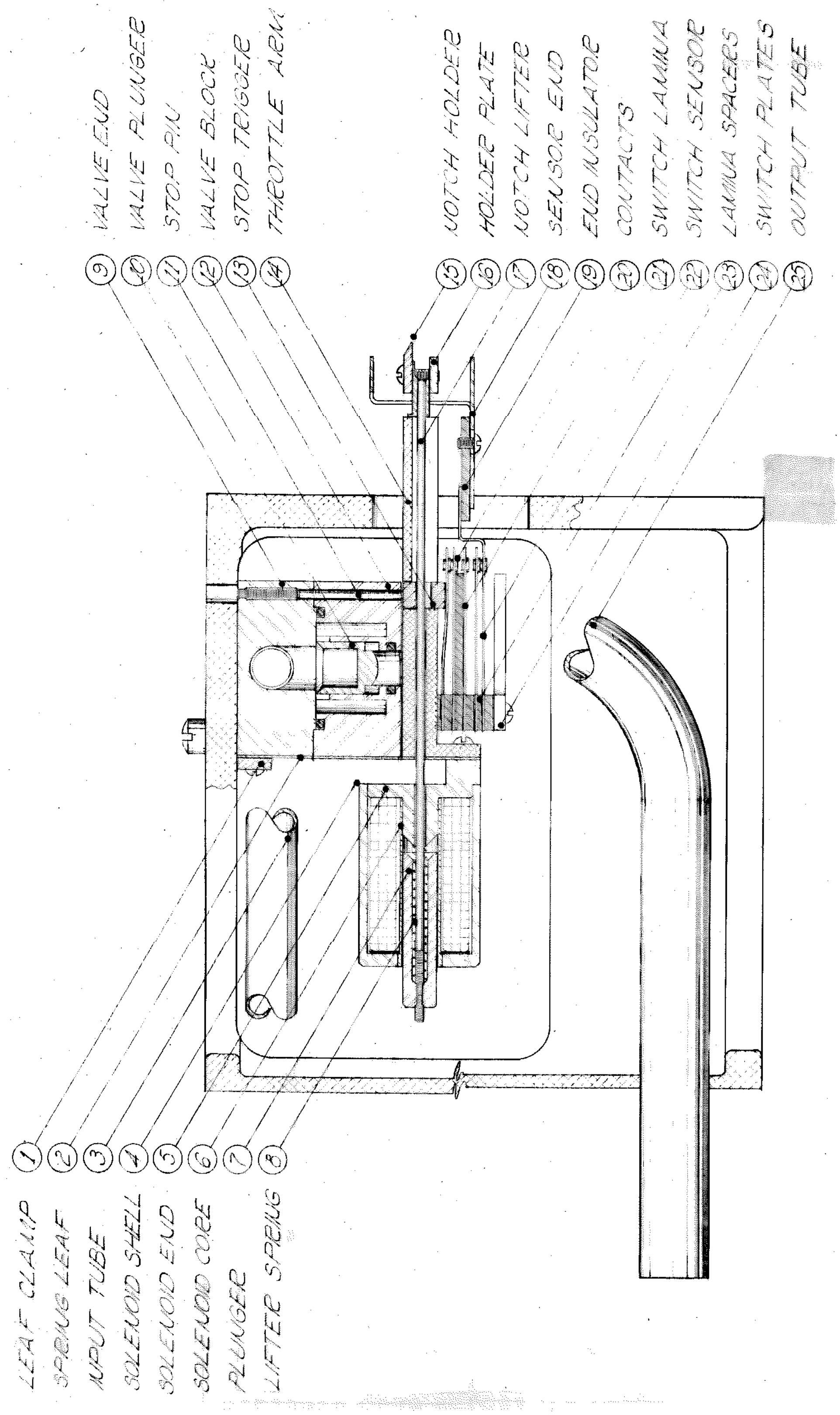


Fig. 17. Control Valve Assembly - CV

Other factors such as a closed control orifice, may prevent motion.

Reverse.

For the reverse or runback mode, the 3-way solenoid valve is powered. It is necessary only to limit the runback speed to some maximum value. To obtain high speed where only low force is necessary, the circuit (commonly termed a "differential circuit") as shown is used. For no line losses the force of the cylinder is equal to the bias pressure times the area of the piston rod, and the volume of oil needed for a full stroke is only this area rather than the full piston area times the length of piston stroke. Because both bias pressure and backstop load on runback are essentially constant, the maximum runback speed is inherently limited by the line losses which increase with speed.

Advance - (Speed Controlled).

In the advance mode as shown in the circuit diagram, solenoid B, the 3-way solenoid valve is unpowered. Bias pressure is then applied to the rod side of the piston, and the flow from the face side is directed to an adjustable pressure regulator. The output of this regulator provides a constant pressure input to a variable orifice. The output from the variable orifice is directed to exhaust. It is assumed that the losses in the lines between the outlet of the regulator and atmospheric pressure of exhaust are negligible. (In practice the effect of these losees is reduced by using large lines and ports for the portion concerned, and by increasing the setting of the pressure regulator.) The pressure drop across the orifice then remains constant, and independent of load. For each orifice size there is a definite speed of advance. In this circuit then, for any controlled variation of the orifice, the setting of the pressure regulator determines the maximum and minimum speeds, and the acceleration characteristics.

Speed Control.

In this machine, the variable orifice is included in the control valve assembly as a plunger type, knife-edged VALVE (10). The displacement of the valve and thus the size of the orifice is determined by the position of a THROTTLE ARM (14) which positions the plunger. The full open position of the arm corresponds to full speed of advance. A hydraulic imbalance tends to hold the valve and arm open.

Advance to a Program Notch.

Notches on one edge of program bars mounted on the capstan shown in the Fence Assembly of Figure 16 provide mechanical signals for the valve, and thereby for speed control.

When the backstop moves forward, the throttle arm is open. It continues, either accelerating or at maximum speed, until a blade or NOTCH HOLDER (15) on the end of the arm engages a notch in the program bar. Further advance speed is reduced by the closing of the valve, until the backstop stops. The deceleration during this length of travel is such that the lift of paper pushed does not slide forward by more than a tolerable amount after the backstop stops. When the throttle arm is closed, the valve is closed and is at a distance fixed with respect to the notch. Thus the backstop stops at a location fixed with respect to any programmed position.

Fixed Backstop.

Once the backstop stops, the paper is located and could be cut. In some cases, however, it is desirable that the backstop remain stationary for a number of cuts at a fixed location. The knife-edged valve or control orifice has a small leak. This tends to cause a small net flow from the outlet or piston face side of the backstop cylinder which corresponds to a slow forward motion of the backstop. This had to be prevented, otherwise the notches, bars and control system would have to be considerably stronger. The device used to do this is the small fixed orifice shown on Figure 8 joining the ends of the cylinder. Once the backstop has stopped and the paper has overshot, there is relatively little net force across the piston. Thus the pressure on the rod side exceeds that of the face or outlet side. The fixed orifice is calculated such that the flow from the rod to the face side equals or exceeds that through the leak in the control valve. The result is that either the backstop remains stopped, or reverses slightly. If it reverses, the valve opens and an imperceptible oscillation can occur.

Notch Sensing and Escape.

Once again the properties of the stock to be cut were introduced into the control design. Trimout cuts are often made between impressions left by the printers. They can produce a ribbon of waste as narrow as .015, but these are never adjacent on automatic programs. If this .015 in. spacing were used as the limit for the size of the notches, the depth would not be sufficient for a positive signal. For this reason, notches are on two levels at the location of a trimout cut. The deepest, the standard size, is first sensed. In order to guarantee this, the throttle arm must move laterally with considerable speed relative to its forward speed. This speed requirement was one main criterion applied to the determination of the spring constant of the SPRING LEAF (2) of the control valve.

The problem after a cut has been made according to a notch signal is for the notch holder to get from the standard notch either up half its depth to find a trimout notch (spaced, say .015 - .050 ahead) or to climb fully out of the notch in case it was not followed by a trimout notch. In neither case can the notches be assumed to be a precise lateral distance from the valve as this would require expensive manufacture and maintenance of the bars, notches and capstan, nor can the notch holder blade be allowed to miss. The NOTCH LIFTER (17) escapement device accomplishes this. This device achieves all functions while using only one solenoid, N, the notch solenoid, and thus only a single logic element in the relay circuit. The laminar end of the notch lifter is the critical portion of the device. It slides against the notch holder and its thickness is less than the spacing of the closest trimout cuts. When the solenoid N is deenergized, the distance between the end of this lamina and the end of the notch holder is slightly greater than the depth of a standard or deepest notch. When the solenoid N is energized, the motion of the lamina is limited (by the small protrusion to the left of the notch holder as shown), such that the lamina protrudes a small distance a beyond the edge of the holder. Adjustments are included so these features are guaranteed.

The relay circuit is designed so that the notch solenoid N cannot be powered unless the STOP signal is present. The STOP signal is supplied by the displacement of the STOP TRIGGER (13) of the control valve when the valve is closed. This STOP signal is also used as an additional safety interlock in the clamp-cut cycle, guaranteeing that unless the backstop is stopped, the knife cannot be fired.

The throttle arm can be moved to close the valve, and thereby supply the STOP signal and allow the notch solenoid N to be powered, by one of two signals; force of the pulley plate pin caused by the operator, force of the notch holder caused by a program notch. If the solenoid N is powered in the first case, friction between the pin and the arm prevents any motion of the throttle arm. If the solenoid N is powered when the valve is closed due to a programmed notch, three cases are considered:

1. <u>Blade in a standard notch.</u>

   The notch lifter slides a distance a beyond the end of the notch holder. The lifter is on the edge of the bar beyond the notch, and the holder is a from the bar. The valve starts to open due to hydraulic imbalance. Thus the STOP signal is removed, and the holder drops down against the bar. Once this happens the lifter leaves the bar and advance proceeds.

2. Blade in a standard notch with stepped trimout notch.

   The notch lifter slides a beyond the end of the notch holder. The holder is on the edge of the trimout step beyond the notch, and the holder is a above the step. The valve starts to open due to hydraulic imbalance. Thus the STOP signal is removed, and the holder drops down against the step. Once this happens, the lifter leaves both the step and the bar, and the holder detects the trimout notch.

3. Blade in a half depth (or trimout) notch.

   Action is as in case 1 but the holder is raised and lowered a plus the trimout notch depth from the bar. The extra distance and therefore time of descent does not interfere with sensing the next notch, because the next notch is never a trimout notch.

B. Electrical Backstop Circuit

For purposes of discussion of the electrical portion of the backstop control, references are made to the control valve assembly of Figure 17, the hydraulic backstop circuit discussions of section 4.11A and the relay circuit of Figure 20.

Four electrical functions are included in the control valve assembly. Three are limit switches and the other is the notch solenoid.

N (Solenoid). This is the notch solenoid. When powered it tends to raise the holder out of a notch and allows the valve to open. Cases where the lifter does not actually lift the holder out of a notch do not affect any function of the control.

STOP (Limit Switch - NO). This is closed by the STOP TRIGGER (13). By means of the STOP PIN (11) and the set screw it is set to correspond when closed to a "valve closed-backstop stopped" situation.

REAR (Limit Switch - NC). The SENSOR END (18) activates this limit switch. When open it is displaced forward toward the sensor end. In operation as designed, this represents 'backstop is reversing - rear limit stop is within 1/4 in. of notch holder".

FRONT (Limit Switch - NO). The SENSOR END (18) activates this limit switch.

When closed it is displaced backward toward the sensor end. In operation as designed, this represents "backstop is advancing-front limit stop is within 1/4 in. of the notch holder".

The limit stops described are small mechanical stops set on the fence to limit forward and reverse travel of the backstop to any prescribed region.

Operations.

A review of the functions of various elements of the relay circuit, as presented on page 136 allows the interrelated function of the electrical, mechanical and hydraulic control of the backstop to be described.

Two portions of the relay circuit are considered:

1. Solenoid N. When powered, this raises the notch holder out of a notch in a program bar. Hydraulic imbalance within the valve then opens it until either the next notch is struck or the valve is fully opened. In both cases, the backstop advances to the next location to close the valve.

   Conditions for powering Solenoid N:

   a) Valve is closed (STOP) and ADVANCE button is depressed.

   b) Valve is closed (STOP) and AUTO switch is on and relay K is not powered (thus knife is ascending) and wheels are on RAMP limit switch, located to indicate that the knife is near the top of the stroke.

2. Solenoid B. When powered, this Solenoid B built into the backstop 3-way valve makes the backstop reverse or runback. Solenoid B is powered if relay B is powered, thus conditions for powering either are identical.

   Conditions for powering Solenoid B:

   a) FRONT limit switch is depressed.

   b) REAR limit switch is not depressed and REVERSE button is depressed.

   c) REAR limit switch is not depressed and ADVANCE button is not depressed and Solenoid B is powered.

A large number of combinations are possible. All are safe and many are useful.

It should be noted that according to la), as soon as the notch holder lifts from a notch, a slight opening of the valve removes the STOP signal. This guarantees that the notch holder is lowered to the sensing position at the same time as the backstop tends to resume advance. Thus the holder is always riding the capstan bar during backstop motion.

The main modes are MANUAL and AUTOMATIC. In both modes:

1(a) The notch holder can be removed from a notch by pressing the ADVANCE button.

2(a) When the front settable limit stop is struck, the backstop reverses due to the FRONT limit switch.

2(b) When the REVERSE button is depressed the backstop is reversing, except when the REAR limit switch is depressed.

2(c) Once reversed, the backstop continues to do so until either the ADVANCE button is pressed, or the REAR limit switch is struck.

The only difference between AUTO and MANUAL modes is given by the fact that in AUTO:

1(b) The notch holder can be removed from a notch by powering the cycle. Removal occurs when the ascending wheels pass over the RAMP limit switch located near the top of the ramp.

The AUTO feature allows the operator to run a sequence of cuts without removing his hands from the cycle buttons. Alternatively, he may run only one or several in this way, and then release the cycle buttons in order to remove the stock cut. Following this he can resume the sequence until completed. After the backstop pushes the last cut portion of the lift towards the front table, it then can advance slightly to strike the front set limit stop which then causes it to reverse and strike the rear set limit stop. It then advances and comes to rest in a "rest notch". The "rest notch" is the first notch encountered on the program after the backstop has reversed at the rear limit stop. For convenience, an equivalent "rest notch" can be built into the forward portion of the rear set limit stop.

C. Genesis of a Meter-Out Control Valve - Figure 18

As is revealed by the succession of layouts shown in Figure 18, the control system which led to the Control Valve Assembly was not the result of a first attempt. These layouts can be used to demonstrate the evolution of some of the features incorporated in the final design. This design process for this valve is representative of the method used for all portions of the machine.

Layout A

An early proposal, this included some disadvantages. Both edges of a program bar were sensed by the two small vertical pins shown. This solved the problem of sensing closely spaced trimout cuts by reading two channels. It introduced the problem of extra logic to read the two channels and to move the head out of the way for program change, as well as requiring that two sides of a program bar be used for a single program.

The pulley mounted cable for manual control was located such

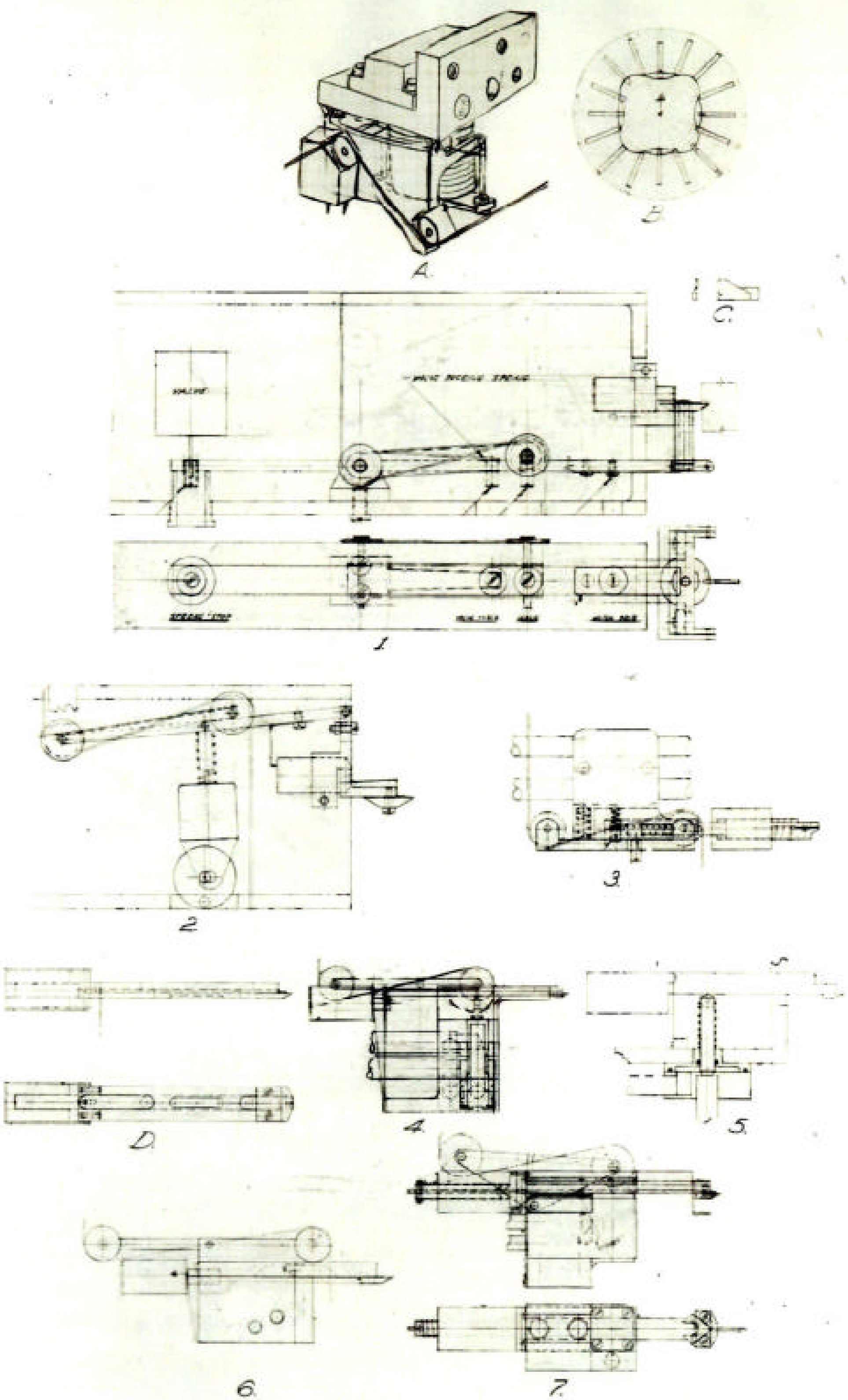


Fig. 18. Genesis of a Meter-out Control Valve

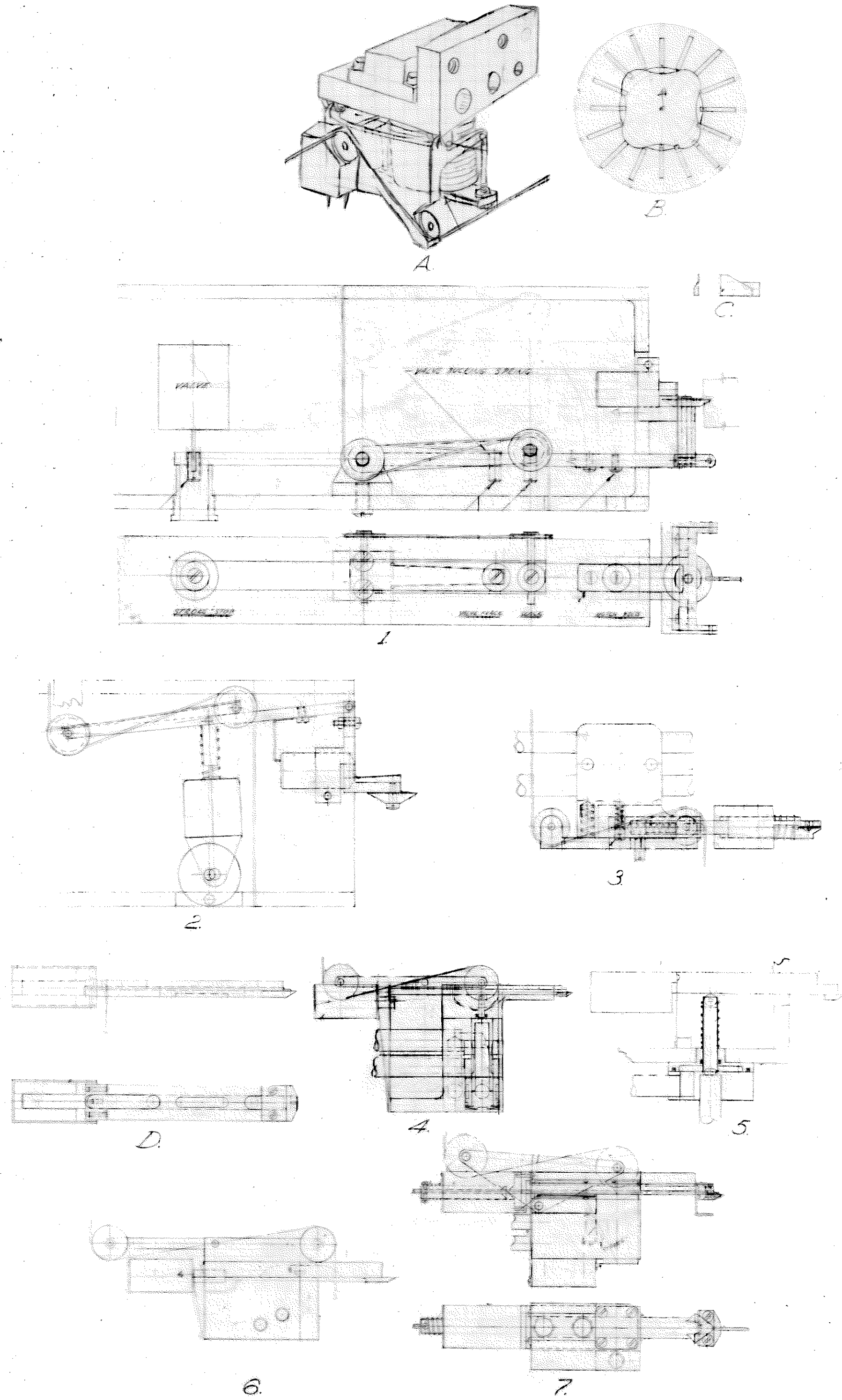


Fig. 18. Genesis of a Meter-out Control Valve

that unless idlers were added, it would be controlled by a lever at the front of the machine to the left of the table. This would restrict later use of that region. It was more logical for the manual control to be located on the main table.

The center line of the capstan would be lower than the table edge at the front of the machine. This would require extra brackets and an offset drive for convenient manual rotation of the capstan.

The valve and sensor assembly was bolted to the end of the backstop. This required that it be rugged or else it would be prone to damage. The assembly would have to protrude beyond the rear table and would be the only feature establishing more than the minimum theoretical length of the table.

There was little doubt that the assembly could be made to work, but if the criticisms are reviewed a statement of a more ideal solution can be made:

1. Only one edge of a program bar should be used.
2. A single electrical signal should be used to escape from one notch and to prepare to sense the next (either a trimout or a regular cut).
3. Programs should be changed by simply rotating the capstan, requiring no extra logic.
4. There should be no offset in the manual drive to the capstan.
5. The manual control cable should be located so that the operating lever is towards the left front corner of the main table.

6. The body and sensor for the valve should not be exposed to damage.
7. The associated features should be within the minimum table length.

By means of the designs that followed layout A, these criteria were met in the final control valve assembly.

Layout B - (Meets Criterion 1)

This showed all cuts could be sensed with a quarter rotation of a thin lobed lamina adjacent to a circular knife edge. The lamina, thinner than the closest spacing of trimout cuts, would raise the knife edge out of a full depth notch, then lower it into a trimout notch or on to the bar. While it did this, it would hold the location of a trimout or other notch until it was replaced by the knife edge which recorded the signal. This was regarded as an innovation of great importance to the whole control system.

Layout C - (Meets Criterion 1)

This is the analog to A, except that a single oscillation over a 45° portion of the lobed solution is substituted for a 90° rotation.

Layout 1 (Meets all Criteria)

This layout demonstrated that a complete solution could be attained. The decision to incorporate all features in a single smaller assembly lead to the other layouts which explore different variations, adding some features, while rejecting others. Reasons for these features can be understood by referring to the description of the features of the Control Valve Assembly, Figure 17.

D. Genesis of a Meter-In Control Valve - Figure 19.

It has been pointed out above that both the meter-out and meter-in types of backstop control were considered before the meter-out system was finally chosen. Since the meter-in system still has some advantages and several persons participated in preliminary designs, some of the resultant layouts are reproduced in Figure 19 and described herein. In this way a reference is supplied for others who may wish to design such a control. The fact that all of these layouts represent solutions that satisfy the initial criteria of the meter-in valve later described, should be noted. The question of which one should be selected can then be answered only by a decision based on secondary criteria.

As will be seen by a consideration of the description below, these secondary criteria might be:

1. For cheap dynamic sealing of low pressure that provides an integral spring, a bellows should be used.

2. For rapid large area access to exhaust by application of a short motion at a fixed location, a coaxial knife edge valve is the best of the solutions shown.

3. Most of the effort should be applied to the location of ports such that internally machined annuluses are avoided or their depth is reduced.

4. Spring loading of the knife edge valve, due to its short-stroke, can be accomplished by a Belleville washer.

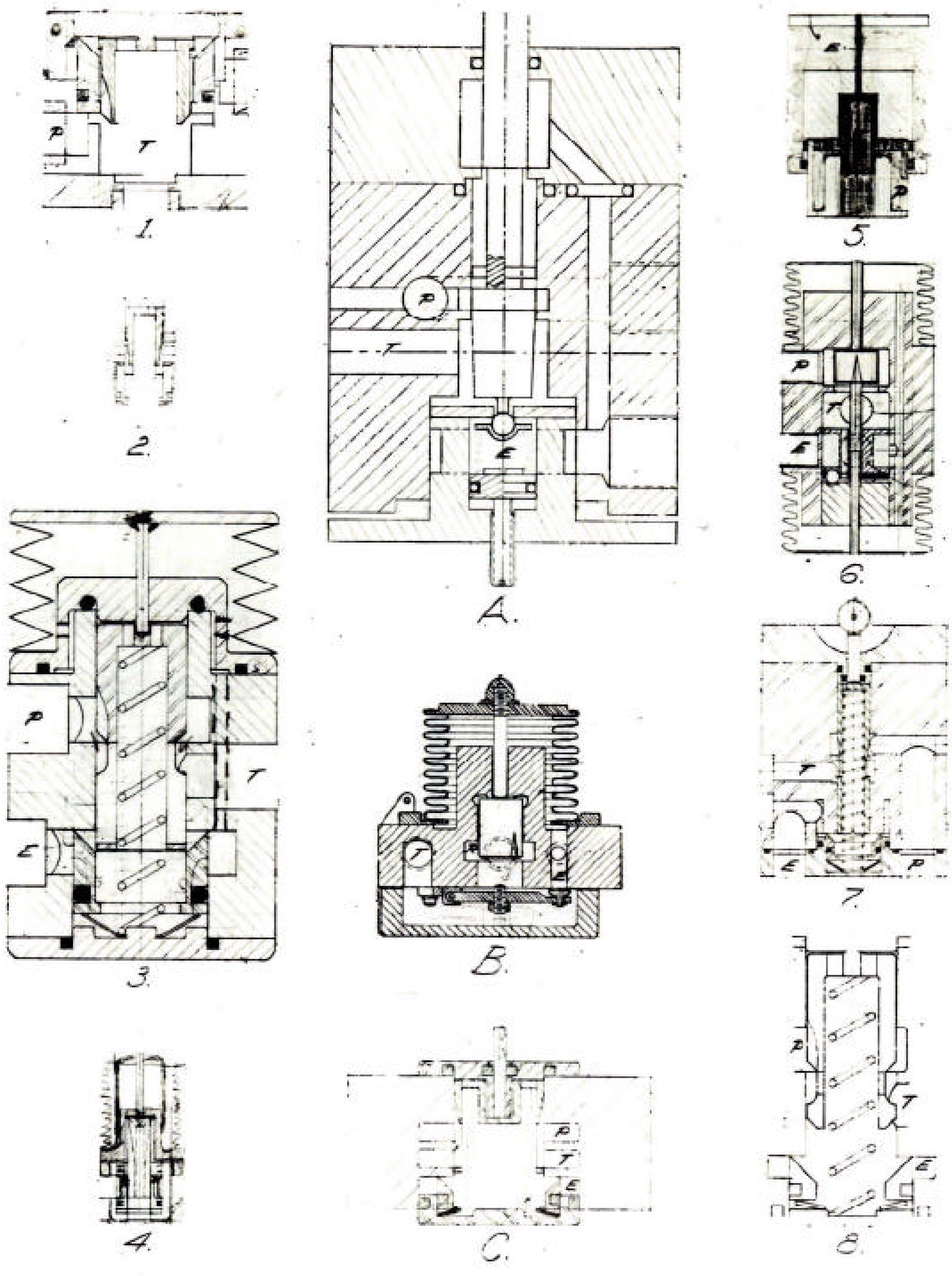


Fig. 19. Genesis of a Meter-in Control Valve

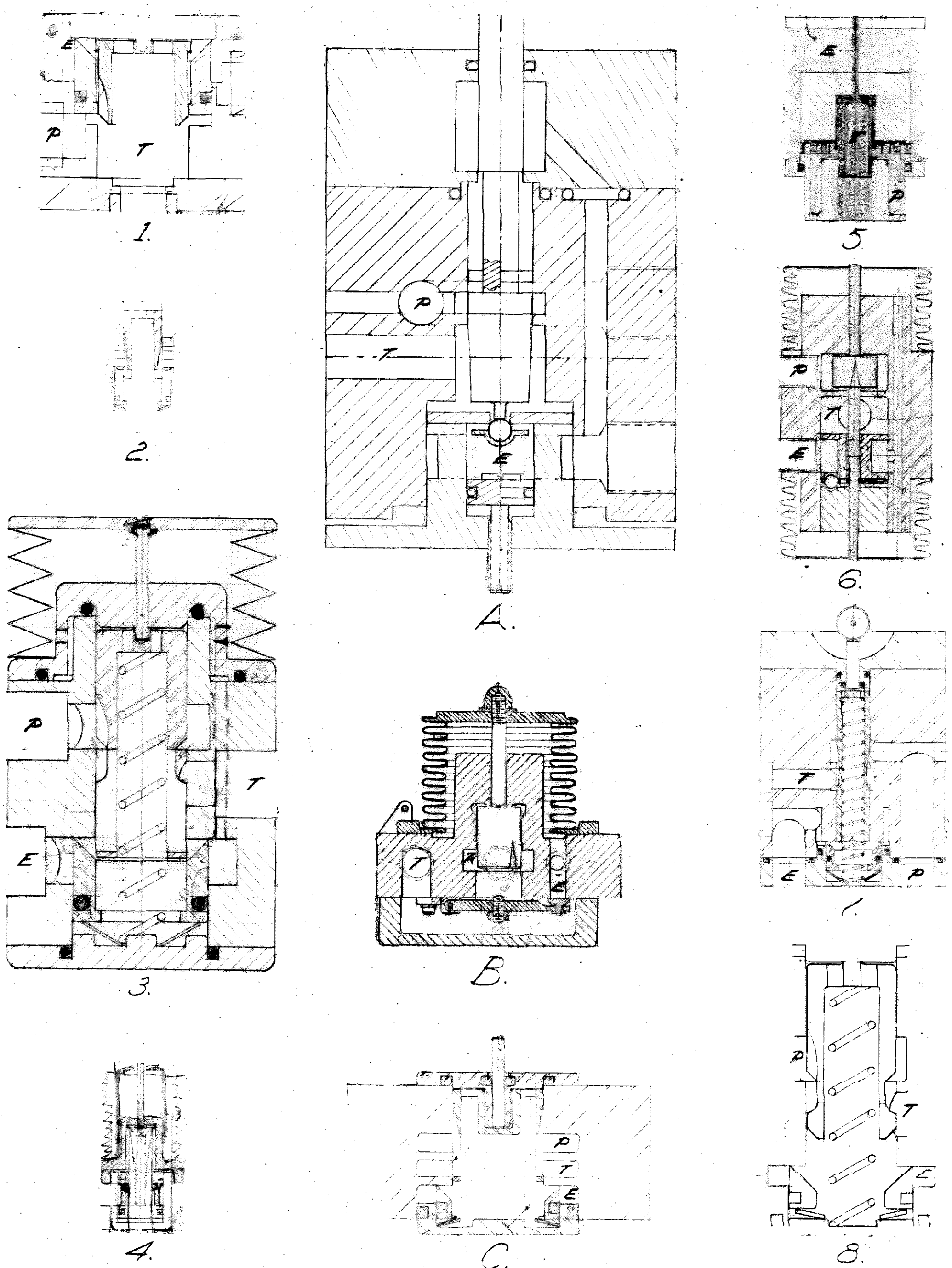


Fig. 19. Genesis of a Meter-in Control Valve

These criteria, although elementary, should prove useful in the synthesis of a new layout of a meter-in valve. It is noted that the advantages determined in exploratory layouts of this form are not included in the initial "problem statement". Instead they are an intermediate step used to generate subsequent more precise and useful problem statements.

Ports lettered P, T, E correspond to PRESSURE IN, THROTTLED OUT, and EXHAUST, with respect to the valve of Figure 19.

Early consideration of backstop control established preliminary criteria for a valve that would decelerate and stop the motion of the backstop at a fixed location with respect to the valve body. If possible, the valve might also include adjustment of this location.

The suggestion was made that motion of the plunger could control flow by varying the size of an orifice. When the output became almost zero it would be suddenly directed to exhaust. This would occur at a fixed location with respect to the valve and would amount to removal of the force of advance at a speed such that variation in the distance required for the lift to decelerate to zero speed would be of the order of .002 in.

The main problem occurred when the throttled flow was to be exhausted suddenly. It was necessary to provide, during a stroke of about .002, an opening to exhaust which was relatively large with respect to the clearances of the valve. The location at which this exhaust occurs must be constant.

It was concluded that layout C best satisfies this requirement and features of this layout would have been used if a meter-in valve had been chosen. The large circumference knife edge valve allows precise location and short stroke for a large port. Layouts 1 through 8 are variations of the geometry and porting involved.

Layout A includes a small circumference of the exhaust port. It offsets this by using a detent action to provide increased stroke of the exhaust valve with respect to a stationary mechanical stop, without extra travel of the valve body. This layout was made by H. Sturtevant, a former design student.

Layout B also uses a small circumference port. It increases exhaust port area for the small stroke by means of a lever multiplier. This allows convenient placement of the ports. Layout B was made by R. Peters, a former design student.

4.12 The Electrical Circuit - Figure 20

Prior to the first test of the machine, only that portion of the electrical circuit essential to the firing of the knife and clamp was established. It was only at a late stage of the design that the mechanical functions were known sufficiently well to complete the circuit. The circuit is best described as one unit, considered in sections.

Power Supply

Three phase, 220 volt power runs to a standard motor-start box attached to the motor. Two phase wires from this are connected to the DC power supply attached to the starter box. This supplies 28 volt DC between terminals a-b and b-c. All further connections are to 28 volt DC.

Pressure Control

The pressure control portion of the circuit operates the solenoids and lights of the sequence-set feature. One additional light corresponds to a basic low pressure setting and indicates that DC power is on. By means of the decade switch shown, the operator controls the number of solenoids powered and thus the pressure applied to the clamp. The length of the row of glowing lights corresponds to the length of lift for which the pressure is sufficient.

Relay Circuit

Functions of Components

To understand the functions offered by the relay circuit, symbols are included on the diagram. In addition, it is necessary to note the

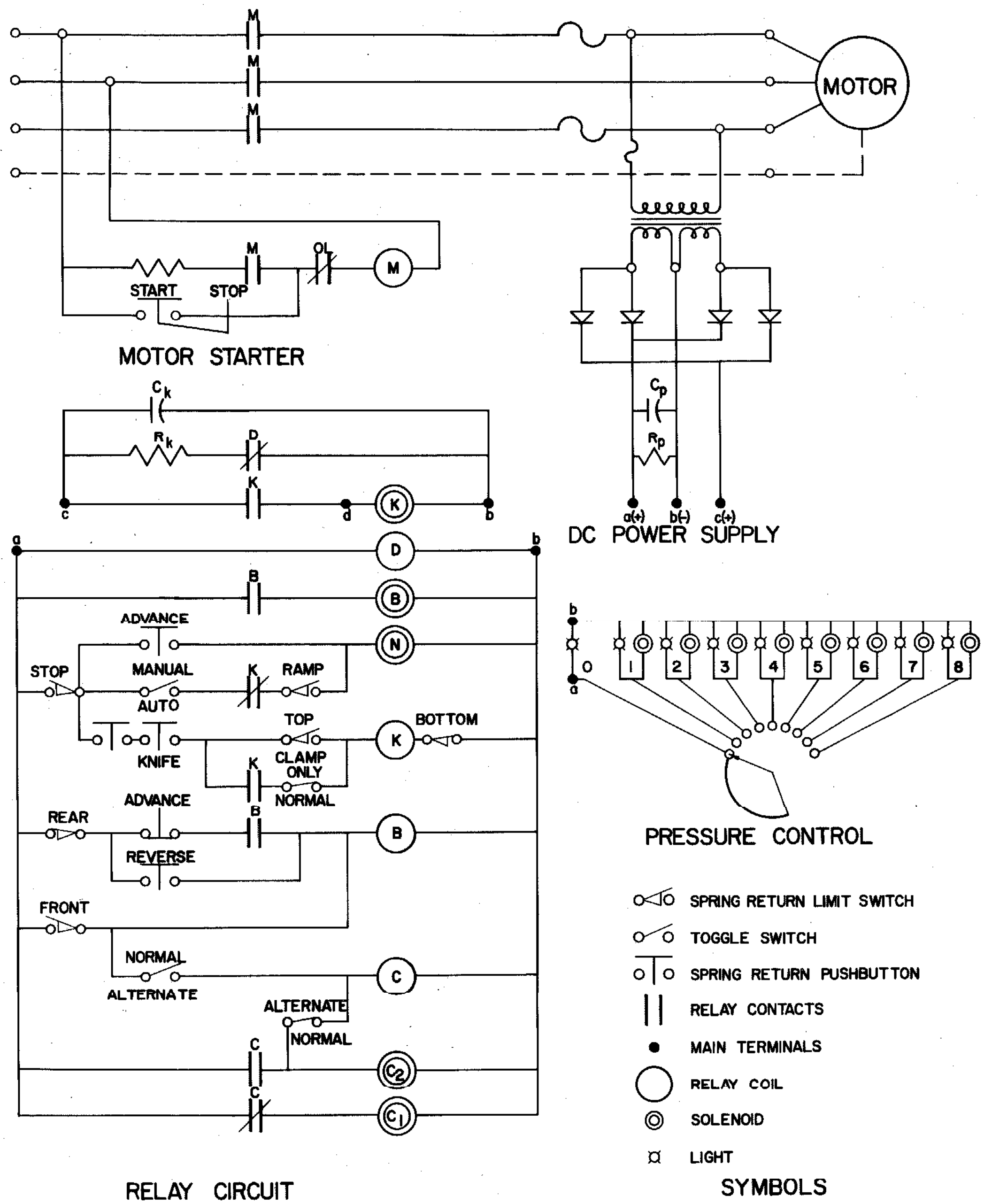


Fig. 20. Control Circuits - Electrical

functions which result from powering the solenoids shown, and the functions required for actuating the limit switches. Other switches are operator controlled. Switches described with one word are spring return; the others are two-position.

<u>Solenoid and function if powered:</u>

K - knife solenoid - clamp closes, then the knife descends;

B - backstop solenoid - backstop goes back;

N - notch solenoid - force to raise notcher, then hydraulic force to open control valve;

C_1 - capstan solenoid (1) - pulls capstan program (1) to read position;

C_2 - capstan solenoid (2) - pulls capstan program (2) to read position.

<u>Limit switches and operations to change their position:</u>

STOP - Control valve closed - (backstop is stopped);

FRONT - Force applied to front of valve seeker (such as due to a mechanical stop);

REAR - force applied to rear of valve seeker (such as due to a mechanical stop);

RAMP - knife bar wheels on a definite point near top of ramp;

TOP - knife bar wheels at top of ramp;

BOTTOM - knife bar wheels at bottom of ramp.

An extra relay, the discharge relay "D", is used only to discharge C_K when the power supply is off. Since C_K must be charged to power the knife descent this eliminates an accidental firing when the power supply fails.

Many variations were tried to reduce the number of relays as well as the number of limit switches. The only limit switch considered to be optional is the RAMP. Its function is really that of a time delay. It guarantees that when the automatic mode is used, the knife will be almost at the top of the stroke before the notch seeker is powered to start the backstop looking for another notch.

Alternator Circuit

One component is not clear from the diagram. For relay C a stepping relay was modified. It is enough to note that successive pulses applied to the coil alternately change two sets of terminals from closed to open. The normal unpowered condition is shown, and alternate terminals are included in the connections to capstan solenoids C_1 and C_2. Before setting up for ALTERNATE, the switch is put on NORMAL to guarantee that the number 1 capstan bar is the first program to be used.

Clamp-Knife Cycle

As an example of the use of the application of the stated component functions to the interpretation of the relay circuit, consider the solenoid K that shifts the four-way valve to start the cutting cycle.

It is assumed that the power supply is on, thus relay D is powered and capacitor C_K is charged. Then it is convenient to list conditions under which solenoid K is powered, i. e. , the clamp-then-knife tends to descend. A single condition is that relay K is powered. Thus relay K can be replaced by solenoid K in our equation. NOT is used to indicate no change or in the case of two position switches, the opposite position.

SOLENOID K = RELAY K = [STOP and NOT BOTTOM and KNIFE and TOP] or [STOP and RELAY K and NOT CLAMP ONLY]

By referring to the lists of functions included, this can be written as: "Clamp then knife descend" if ["control valve is closed - (backstop is stopped)" and "knife bar wheels not at bottom of the ramp" and "operator depresses KNIFE buttons" and ("knife bar wheels at top of ramp")] OR ["control valve is closed - (backstop is stopped)" - and "clamp then knife descend" and "operator has not set CLAMP ONLY]. This sentence includes all features of operations of the knife. Once the wheels leave the top of the ramp, then the OR condition must be satisfied for the cycle to continue. If "clamp then knife descend" then the cycle will not continue if the CLAMP ONLY switch is on. This reveals the operation of that feature. The sequence valve causes the clamp to descend first. Then the knife begins to descend, i.e., "clamp then knife descend" is signalled. But once it leaves the top of the ramp, it cannot continue to descend since the CLAMP ONLY switch is on. Thus it rises back to the top position. Time delays are such that oscillation does not occur.

The operator is required to depress both knife buttons throughout descent, but ascent results as soon as the bottom limit is struck and continues even if the buttons are held. This is due to the fact that the knife must be on the top limit before descent can be begun.

Backstop Control

The other components of the relay circuit are used for control of the backstop. Their functions are described in Electrical Backstop Circuit, page 123.

Control Panel - Figure 29

Manually operated electrical switches are mounted on the removable panel shown at the front of the table in Figure 29. This sheet metal panel can be made with a brake and chassis punches. Production or replacement of units to suit individual customer requirements would be economical.

Relay Box - Figure 21

Components of the relay circuit are centralized in the removable relay box located under the left side cover box as shown in Figure 30. From there cables lead to switches at locations throughout the machine. TOP, RAMP, and BOTTOM wheel limit switches are attached to the relay box. Figure 21 shows the location of these and details of the wiring. Terminals on the left side of elements correspond to the same terminals shown in the circuit diagram. If maintenance of relay logic is necessary, or a different circuit is substituted, the cables are disconnected and the relay box replaced as a unit. This will reduce down-time and requirements for skilled maintenance personnel.

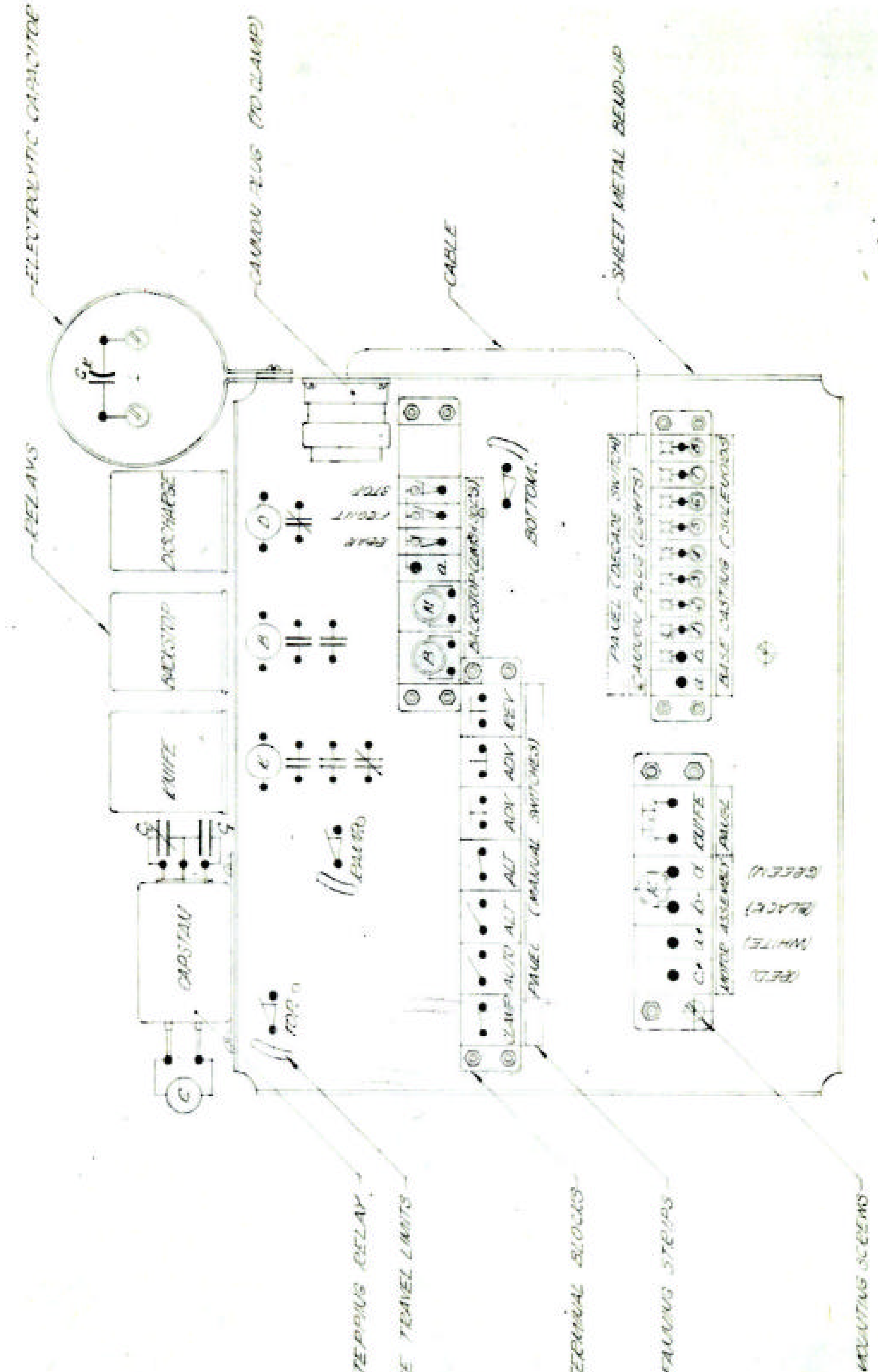


Fig. 21. Relay Box

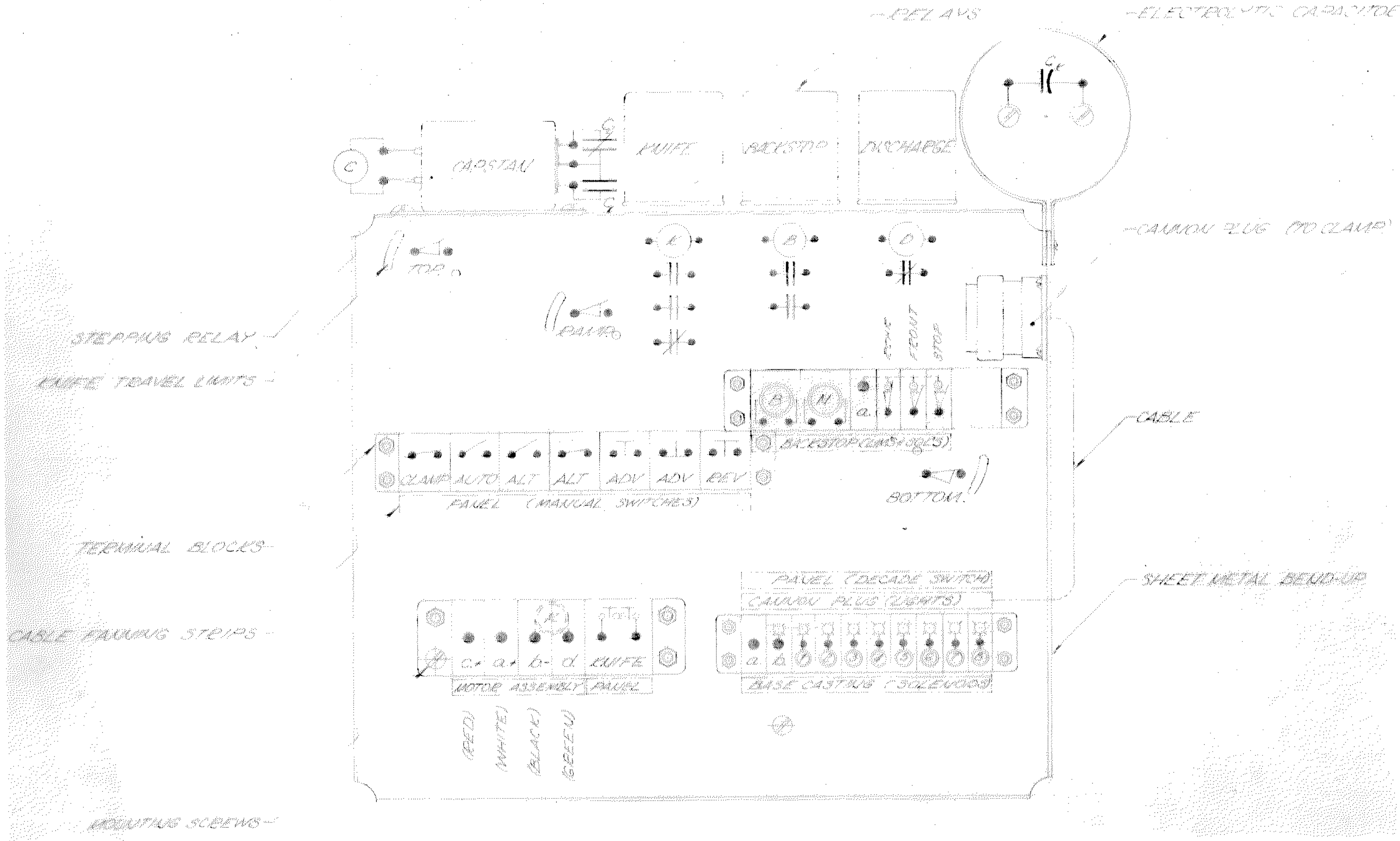


Fig. 21. Relay Box

4.13 Calculated Cutting Performance

1. Synthesis Estimates

During the design of the knife drive, values for the cutting force and cutting angle had been assumed. Based on these, it was determined that for the descent of the knife bar on the straight portion of the ramp, the wheels would not leave the ramp. This was done by first calculating conditions for which wheel reactions would be zero and then establishing the geometry such that the reactions would be positive.

2. Analysis

Following construction the knife drive assembly as shown on Figure 9 was analyzed in greater detail. This was done because of the possibility that the values introduced were incorrect and also because of the need to provide a means whereby actual performance could be used as a preliminary check on the assumptions.

In the event that changes would be required, it was recognized that changes of the values for knife bar weight and hold cylinder forces would be the easiest to make. The ramp angle could be changed without modifying the nature of the fundamental design, but not without some modification of the base casting.

Useful results are included in the form of five plotted curves. They are based primarily on the assumptions used for the design:

$F_c/in = 75$ lbs/in = cutting force per inch of cut

$\theta_c = 21^o$ = angle of cutting force with plane of table

and on the values determined by the design:

w = 300 lbs = knife bar weight

$\theta = 21^{\circ}$ = ramp angle

$r = F_K/F_H$ = knife cylinder force/hold cylinder force

The ramp was divided into positions 1 through 5, equally spaced from bottom to top. The table was divided into positions a to m, equally spaced from left to right. Position g corresponds to the center of a full-width lift, i. e., the location of maximum force if only paper and not some foreign object is cut. "Normal operation" assumes only single lifts of paper are cut, and that one edge of these is against the fence. On the curves shown, calculated forces for positions a to g are based on assumed normal operation and a cutting force of 75 lbs/in. When the term "wheel zeros" is used, it means that the wheel reaction force becomes zero and thus the wheels tend to leave the ramp.

Cutting Forces vs. Angle of Cut - Figure 22

For each of the locations of forces for normal operation, a curve is drawn. This curve shows the maximum cutting force prior to wheel zero for various assumed cutting angles. These are noted as "equilibrium curves".

Superimposed on these is the single curve of calculated cutting force values. For the assumed unit cutting force of 75 lbs/in, required total cutting forces were established and located on the equilibrium curves for the various locations. The single curve that joins these then indicates the maximum cutting angle that can occur without causing wheel jump. The critical angle is 37°.

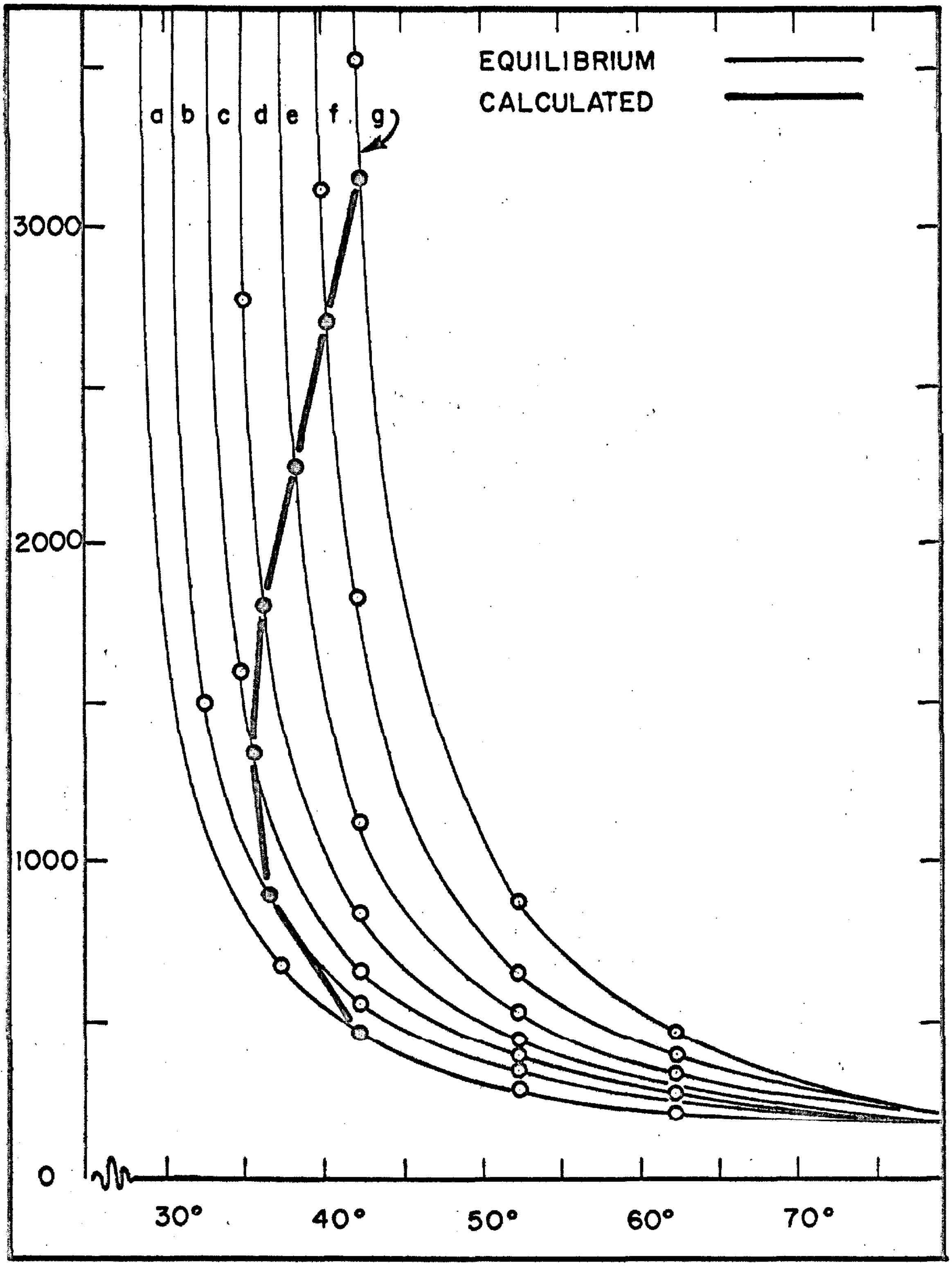


Fig. 22. Cutting Forces vs. Angle of Cut

Conclusion:

For the machine as built, if the cutting force is 75 lbs/in, during normal operation, no wheel zero will occur during descent on the straight portions of the ramps until the cutting force angle exceeds 37^{o}.

Extra Hold Force vs. Location of Cutting Force - Figure 23

It had been shown on Figure 22 that a cutting force angle of 37^{o} could be tolerated with no change to the machine. The curves of Figure 23 show the necessary extra hold force needed if wheel zero at a cutting angle of 45^{o} were to be countered by hold force alone. The angle selected seemed high enough to provide for any possible error in the assumed 21^{o} angle of the design. It can be seen that position 5, corresponding to the upper position of the knife bar, is critical.

Conclusion:

For a cutting force of 75 lbs/in located at an angle of 45^{o} to the plane of the table, wheel zero can be avoided in at least two ways:

(a) A constant extra hold force of 260 lbs can be added.

(b) The ratio of knife cylinder force to the hold cylinder force can be raised to 2:1. 46 from the existing ratio of 2:1.

Angle of Cut vs. Location of Cutting Force- Figure 24

The weight of the knife bar may be reduced at some time in order to reduce shock loads or increase speed of operation, or as a result of experimental values for cutting and side forces. In this case it would be important to know the contribution of the knife bar weight to the equilibrium of the drive.

As a step in this direction, the curves of Figure 24 represent

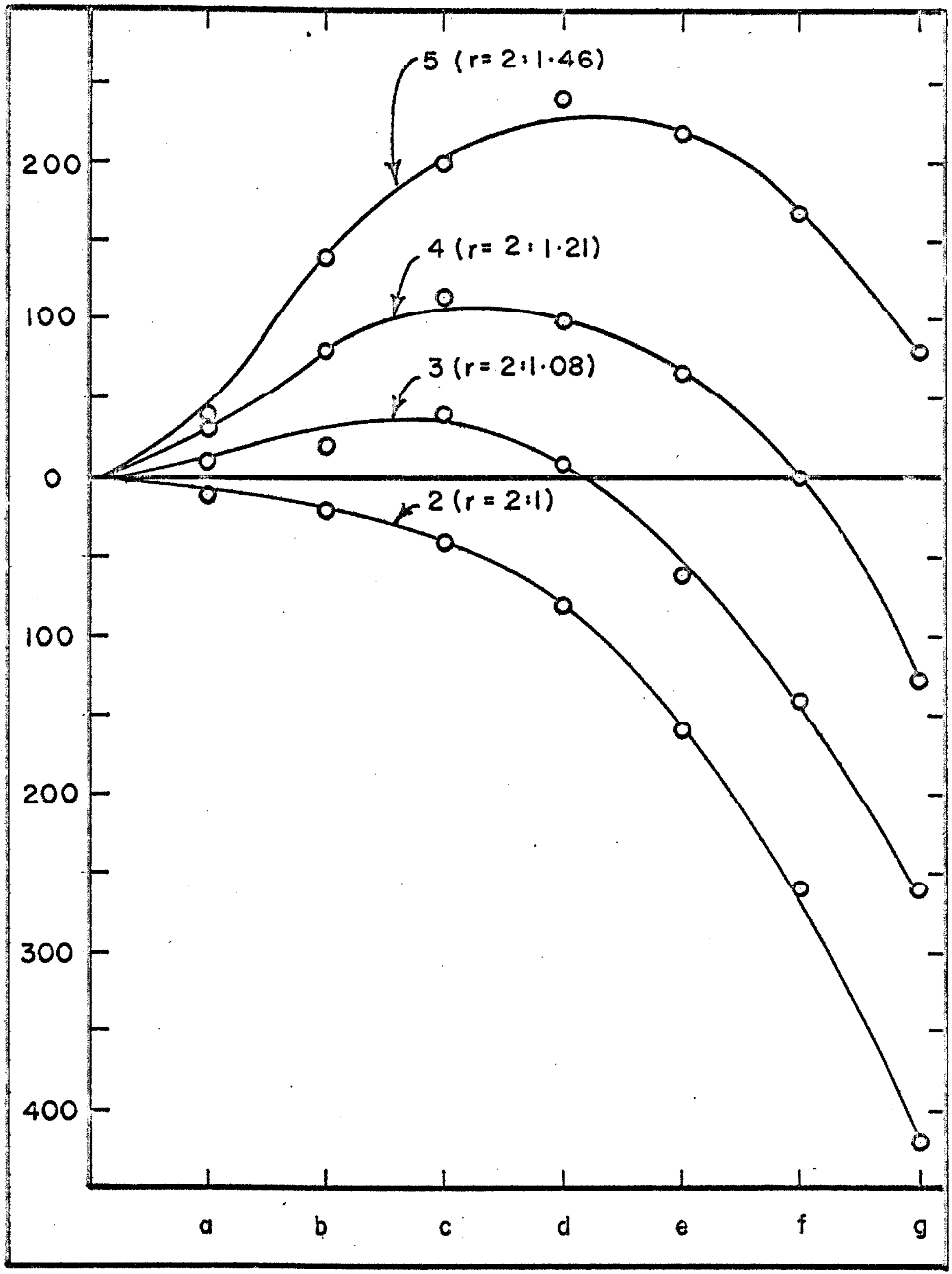


Fig. 23. Extra Hold Force vs. Location of Cutting Force

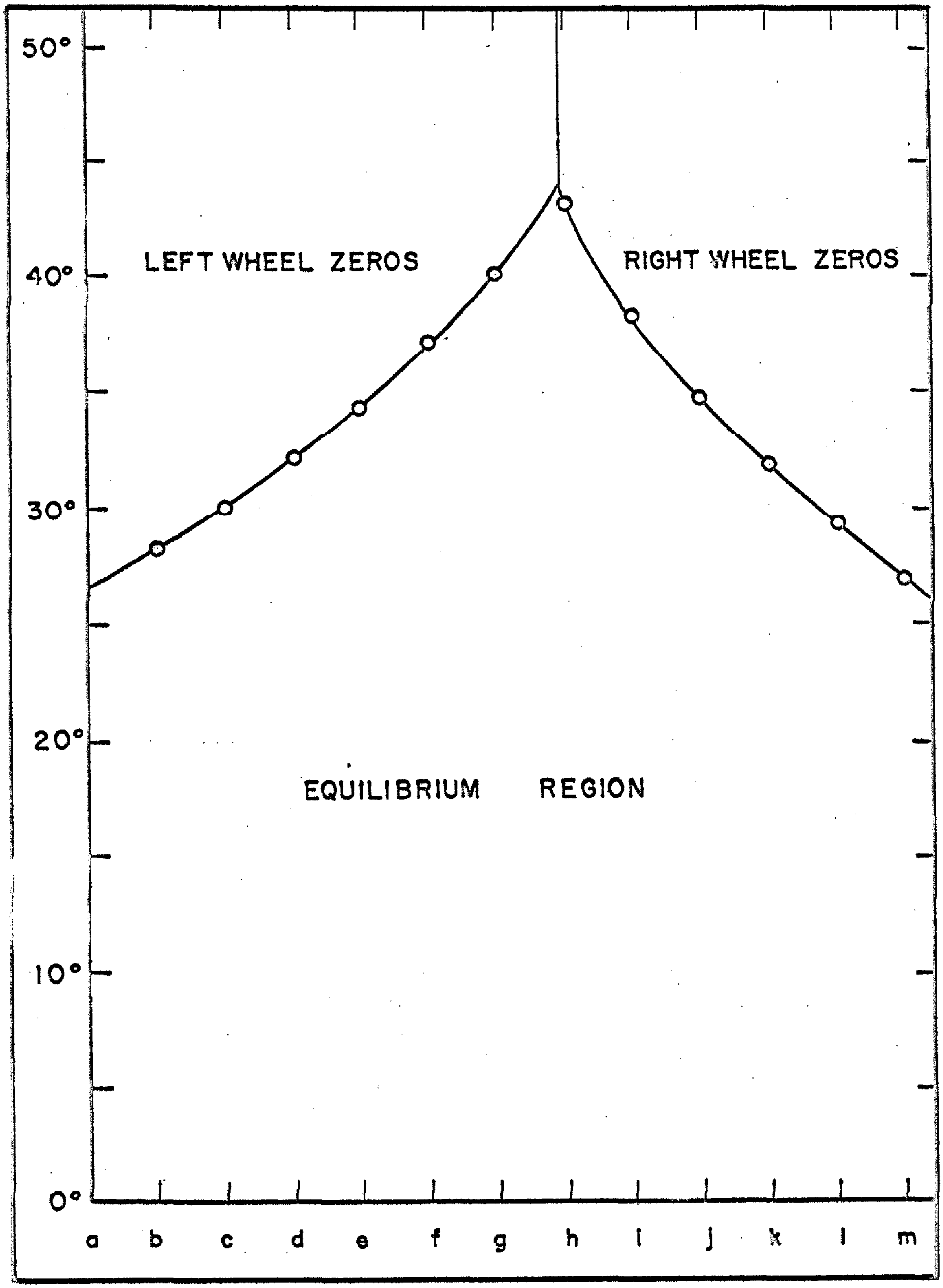


Fig. 24. Angle of Cut vs. Location of Cutting Force (w=0)

values obtained for a knife bar weight of zero. It is noted that for normal operating conditions, i. e., positions a to g, only the left wheels tend to jump.

The critical angle is 27°. This can be compared with the value of 37° obtained for the 300 lb knife bar. It is seen that even the relatively small static wheel loads due to knife bar weight are significant. The natural suggestion that this could be true only for low values of the hold force is supported by the curve of Figure 24. It is seen how an increase of the critical angle (by increasing the hold force) only in the range of low cutting forces, and therefore of low hold forces, (locations a, b, c, d) could raise the overall (locations a to g) critical angle to 37°.

Conclusion:

For the machine as built, and a unit cutting force of 75 lbs/in the wheels of a weightless knife bar would not jump for cutting angles less than 27°.

Wheel Reaction Force vs. Wheel Position and Knife Cylinder Force vs. Wheel Position - Figures 25 and 26

These results were obtained during the analysis of the other effects and show further properties of the drive. From these figures it is seen that the wheel loads are approximately one third of the knife cylinder force, and that the knife cylinder load is approximately proportional to length of lift cut.

The calculated knife cylinder force is approximately equal to th estimated design value.

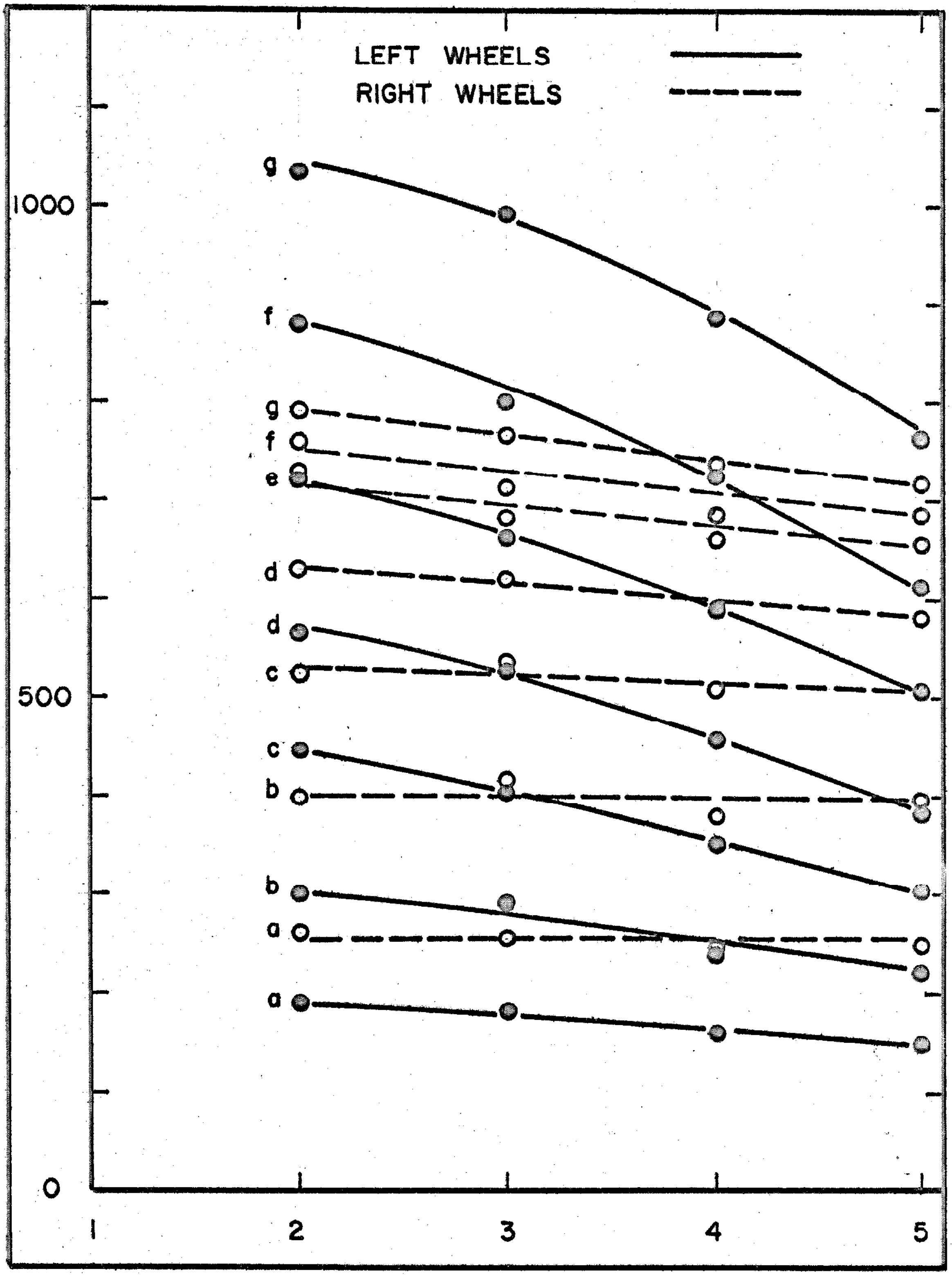


Fig. 25. Wheel Reaction Force vs. Wheel Position

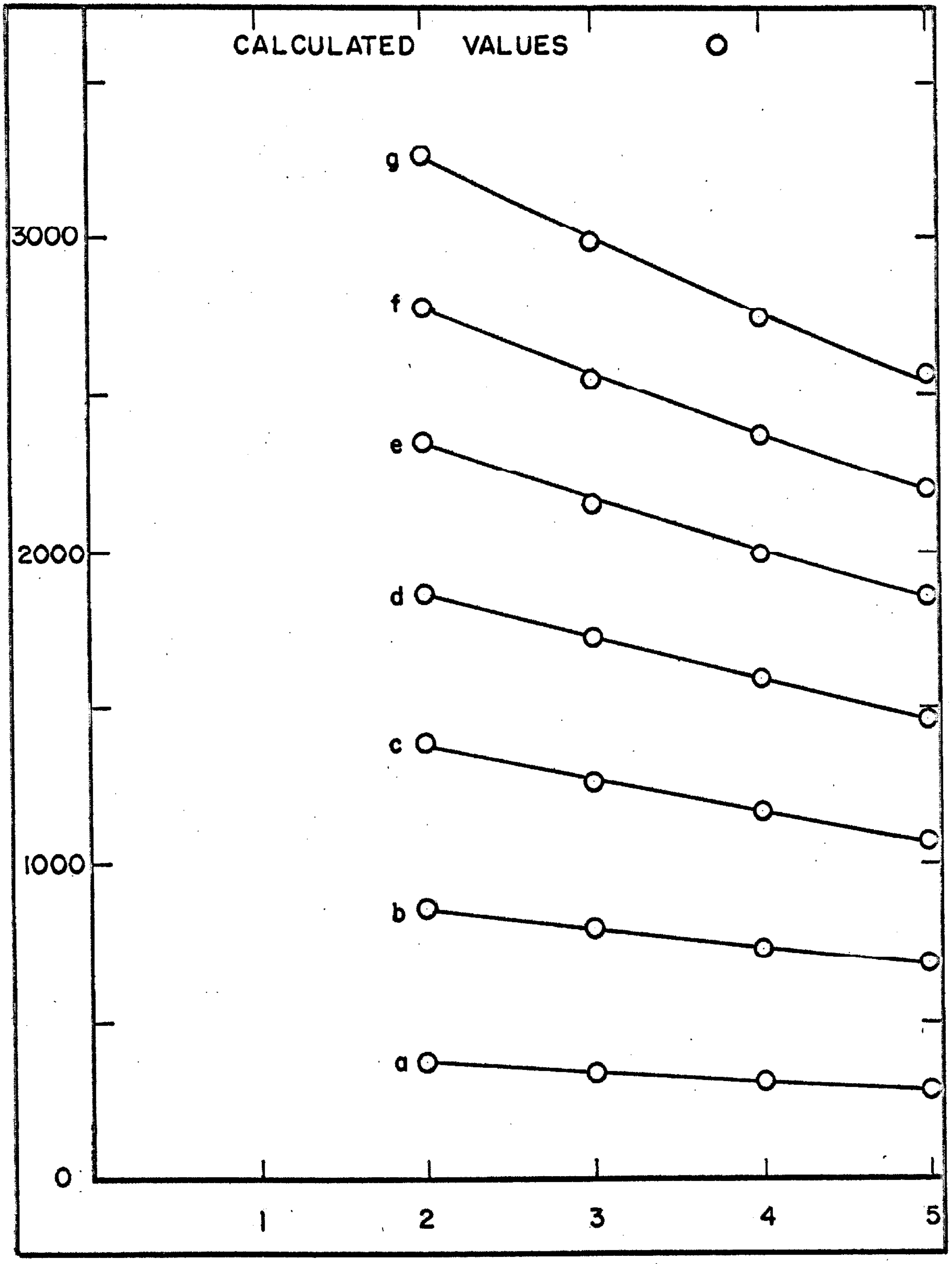


Fig. 26. Knife Cylinder Force vs. Wheel Position

4.14 Hydraulic Tests of Knife, Clamp, and Backstop Drives

A. Knife and Clamp Drives

Only the components associated with the drive of the clamp and knife bars were tested at this time, as the backstop was not yet complete. It was not considered safe to include the blade itself in these initial tests.

The tests brought out the following problems: knife bar shock, component noise, slow clamp stroke adjustment, excessive bias pressure, and clamp pressure in excess of set pressure. By applying pressure taps at key points of the circuit to study the behavior of the hydraulic system, reasons and solutions for these problems were found.

Knife Bar Shock

Problem:

On the first several cycles a large shock occurred at the end of knife bar ascent. Remaining ascents were cushioned. At the limit of the descent a large shock occurred. By adjusting the 4-way valve so that it reversed when the wheels were a couple of inches from the bottom of the ramp, the shock of descent could be eliminated.

Reason:

For the knife bar, ascent and descent had about the same kinetic energy and the hydraulic cushions at each end of the path were identical. Air in the cylinder had accounted for the first several shocks on ascent. The remainder of the ascents were quiet. The difference between the two conditions that caused shock only on descent was explained in terms of energy input. When the lower cushion was dissipating kinetic energy of the bar, energy was being added by high pressure oil available to maintain the maximum velocity of the bar. When the upper cushion was acting, only negligible energy was being added due to small bias pressure.

Solution:

The problem was solved when the valve was set to reverse so that high-pressure oil was not available during cushioning at the lower end. To determine if this might be a reasonable permanent solution, curves of knife bar travel during the "clamp-only" mode were obtained. Although not too accurate a measure, the curves indicated that the bar traveled approximately two inches on the ramp during valve reversal. The order of this estimate was substantiated from the manufacturer's value for reversal time. Since all of the overshoot at the bottom of the ramp was needed for the length of the cushion, the valve would have to be set to begin reversal more than an inch up the ramp. It was calculated that a full lift of paper would stop the motion of the full-speed bar in a ramp distance of less than 3/8 inches. This would mean that for a permanent solution by means of a set limit switch, the setting would be critical and depend on such factors as a constant firing time of the valve. If the switch failed the large shocks would return. It was decided that the switch alone would not be acceptable.

Neoprene bumpers would have to be added to reduce shocks at the top due to air and at the bottom due to a failure of the limit switch. These would be designed to absorb the full kinetic energy of the knife bar.

The rod length would be changed so that cushioning would begin only when the wheels entered the overshoot distance beyond the horizontal portion of the ramps. This could not be exact due to the adjustable eccentric bushings on the wheels.

Component Noise

Problem:

Certain loud noises from components were heard during operation. Mainly these were from the areas of the pump and of the low pressure relief valve.

Reason:

The noises were attributed to two causes:

a) High speed and large output operation of the pump.

b) Excessive vibration of the low-pressure relief valve plunger. The valve used was of the positive seating variety as opposed to the spool type used for the high pressure relief valve, which for the same flow made little noise.

Solution:

a) Neoprene and cork gaskets would be used to partially isolate the pump and motor noise from the "drumhead" of the base. Reduction of the speed was suggested, but would require a new motor. Reduction of the pump output by means of a replacement kit was suggested also and was eventually adopted.

b) The low pressure relief valve would be replaced by one of the quiet type used for the high pressure.

Slow Clamp Stroke Adjustment

Problem:

The clamp was easily lowered and its stroke set by the "stroke-set" feature. To raise it a short distance, however, a large number of machine cycles were required.

Reason:

To raise the setting of the "short-stroke" feature, pressure on the lower side of the floating piston had to exceed that on the upper. This occurred only during the short knife descent portion of the cycle. Due to the small outlet ports from the cavity above that piston, pressure drops were large, and a large number of cycles were needed to raise the piston. This was the result of an oversight in the hydraulic circuit.

Solution:

No minor change was offered to improve the condition.

Excessive Bias Pressure

Problem:

The bias pressure setting required was higher than expected and the excess was regarded as a direct loss.

Reason:

The pressure taps inserted at various points in the circuit indicated that there was a large pressure drop across the run-around check valve included in the sequence valve.

Solution:

It was suggested that either another check valve be added in parallel with the one included in the sequence valve block, or that the built-in check be enlarged. The latter proved satisfactory.

Clamp Pressure in Excess of Set Pressure

Problem:

It was discovered that the clamp had been applying full pressure in spite of low pressures set for the test.

Reason:

This was due to a small casting interference that prevented the sequence valve plunger from opening fully. Thus a large pressure drop through the valve built up to full pressure on the clamp cylinder side.

Solution:

The casting interference was removed.

It was concluded from these tests that all of the problems could have been foreseen. However, all but the ones pertaining to knife bar shock were minor and even a solution to the shock problem was expected without difficulty. No inherent disadvantages in the circuit were detected, and it was considered to be safe and satisfactory.

B. Backstop Drive

When the backstop was ready for preliminary tests, for convenience the backstop tee was secured to a bench away from the main machine. All necessary plumbing was attached. To duplicate final conditions, two hoses later to be used in the final assembly were used to connect this to the hydraulic supply of the base. Battery power was used to activate the 3-way solenoid valve.

Factors that were tested were:

1. Distance for reversal of direction
2. Advance speed and force
3. Reverse speed and force
4. Speed control valve

There were a number of problems, which, along with their reasons and solutions, were as follows:

1. Distance for Reversal of Direction

This gave the most difficulty. For some time it was impossible to reverse direction with the circuit as built. Reasons are best explained by reviewing the complete existing circuit diagram of Figure 8 which includes several changes made at this time. "Pilot pressure" connections are shown as dotted lines within the blocks representing valves. As shown the backstop is advancing.

Advance to Reverse

Problem:

To change from advance to reverse, the solenoid valve was energized. This applied pilot pressure to the diaphragm of the modified pressure regulator to close it and to the modified pilot regulator to open. The pressure regulator did not close completely, thus some of the bias pressure oil leaked through to exhaust by passing through the speed control valve. This dropped the pressure on the piston face below that of the rod side. Since the rod area was only a small fraction of the total area, this drop considerably reduced the resultant net force across the piston and reversal did not occur. When the speed control valve was closed, reversal would occur.

Reason:

The incomplete closure of the pressure regulator was due to the internal orifice and pilot line leading from the cavity above the diaphragm to the output side of the regulator. Because of flow through this orifice, pressure drops occurred upstream which reduced the closure pressure applied at the diaphragm.

Solution:

The orifice involved was the feedback orifice to operate the regulator. A check-valve was first considered which would block flow when the regulator was to close, but it was soon recognized that feedback had to be in both directions when operation as a regulator was required. Thus a check valve was not added. To reduce the flow as much as possible, the orifice was reduced in size.

Reverse to Advance

Problem:

Change of direction was sluggish or did not occur. The modified pilot regulator was closing slowly or not at all.

Reason:

When in the advance mode exhaust flow was considerable. Due to pressure drops downstream, this raised exhaust pressure near the pilot regulator. As can be seen on the circuit, exhaust pressure is present on the side of the pilot regulator that opposes its opening by spring force. The spring was not strong enough to close against exhaust pressure.

Solution:

A satisfactory spring was installed.

Reversal of direction from both modes then became satisfactory. At maximum speed it was within the 1/4 in. slack designed into the reversing switches installed on the speed control valve. For slow speeds of advance the reversal distance was negligible.

2. Advance Speed and Force

This was no problem, the machine functioned as designed.

3. Reverse Speed and Force

Problem:

The force and speed were unacceptably low. It was found that the hydraulic lines were too small, causing excessive pressure drops. The piston rod area was too small and was the only net area to which pressure was applied. In addition, some flow went through the pressure regulator.

Reason:

Force and speed were coupled. To obtain any reasonable force, speed had to be decreased to decrease pressure losses, but there was a critical speed below which this did not work. This critical speed depended on the setting of the control valve and was due again to some flow through the pressure regulator. At zero speed for example, if the control valve was wide open, the net force on the piston could actually tend to advance the backstop while in the reverse mode.

Solution:

Pressure drops for the portion included were reduced by enlarging the tubing. The reduction of the orifice in the pressure regulator increased force and speed.

4. Speed Control Valve

Problem:

There was a tendency to slowly advance beyond a stop location. The knife-edge valve did not seal perfectly.

Reason:

When the cylinder is stopped in the advance mode, there is pressure on the rod side of the piston which tends to advance it. For equilibrium with no external forces, the pressure on the face side of the piston is lower than this value. Any leak out of the low pressure side allows advance.

Solution:

A small leakage path was provided between the high and low pressure sides of the piston. This is represented by the orifice inside and across the ports of the modified pilot regulator. A small slit was made on the valve seat of the regulator. In operation, to guarantee no overshoot, it is necessary only that the flow through this slit due to the pressure drop across the piston equals the flow through the small leak in the control valve. A larger flow causes a slight tendency to reverse. This motion is limited by a resultant small opening of the valve which tends to cause advance.

Two general conclusions were reached.

1. The circuit would suffice to demonstrate the functions of the machine. It would be necessary to add an extra force for the run-back or reverse mode. Temporarily, this could be done by means of a spring loaded reel fastened under the drive tee and connected to the rear edge of the table.

2. Any change of components of the machine should include reconsideration of the meter-in system shown in Figure 8, and also should consider a control valve such as the throttle-to-a-fixed-position-then-quick-exhaust type, as described in Meter-in-Control, pages 77 and 131.

4.15 The Final Assembly - PC 64

All of the components were assembled for demonstration of the machine in operation. The photographs of PC 64 shown in Figures 27 and 28 can be compared to Figures 6 and 7 which show the machine of 1960. Figures 29, 30 and 31 identify the locations of the assemblies on PC 64.

It remained to demonstrate that the machine could cut paper.

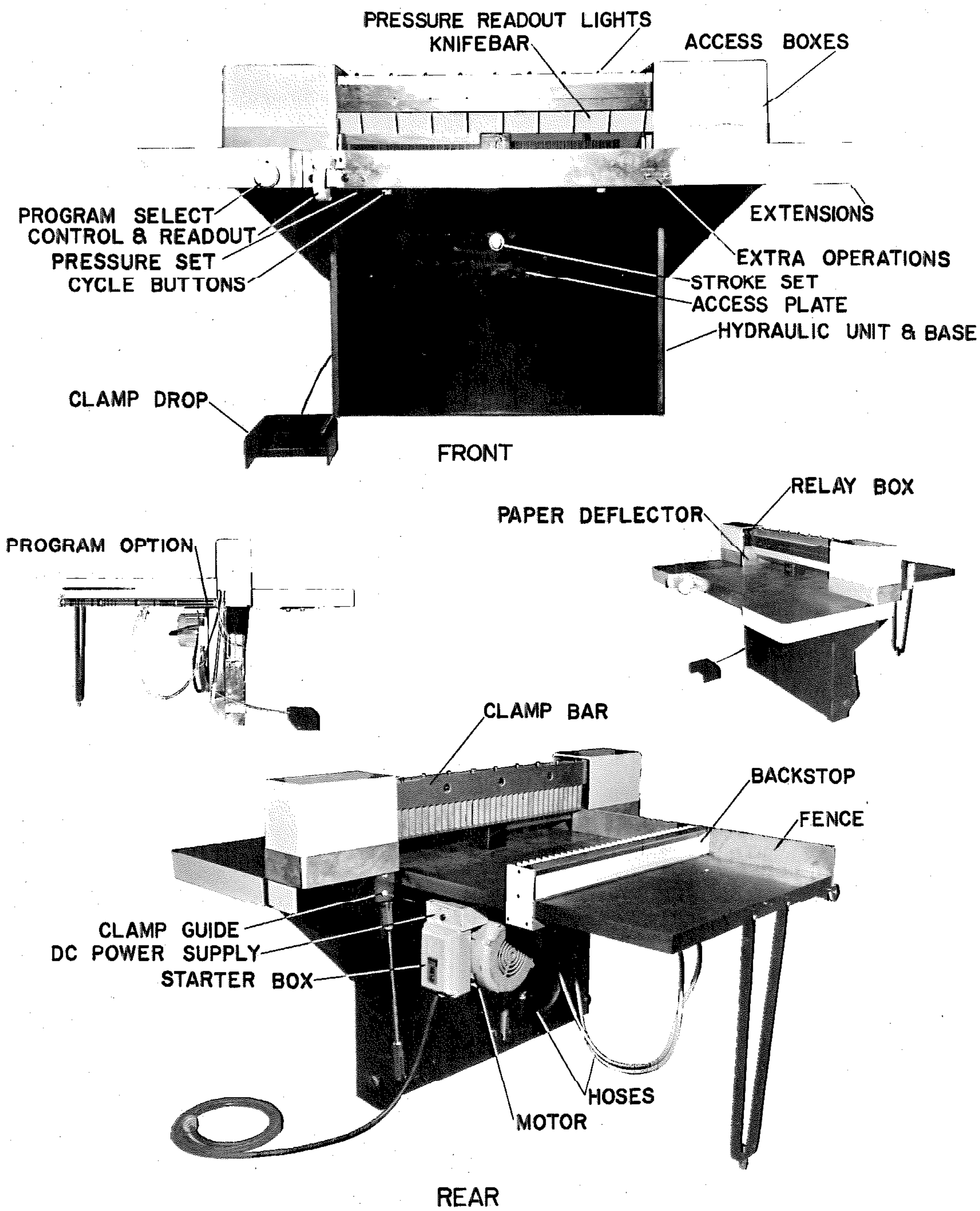
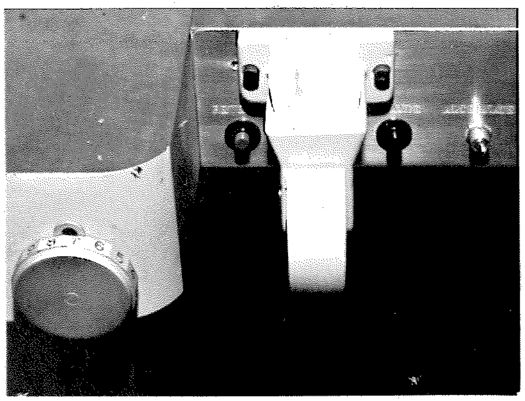
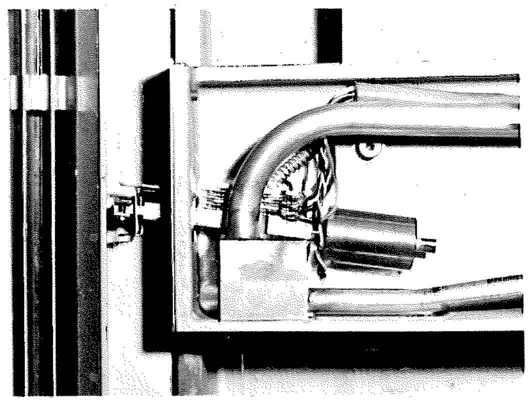


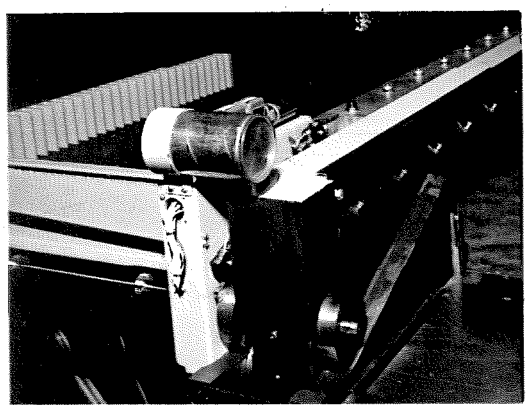
Fig. 27. PC 64 Hydraulic Paper Cutter



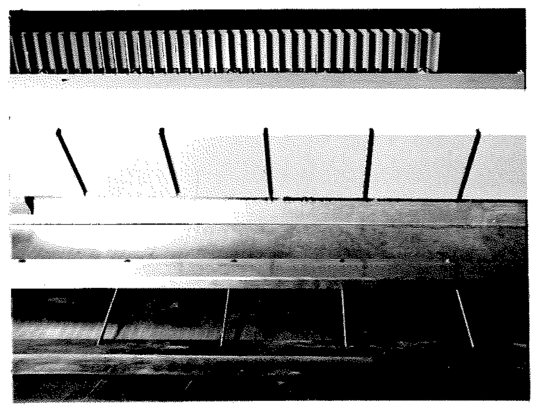
POSITION CONTROLS



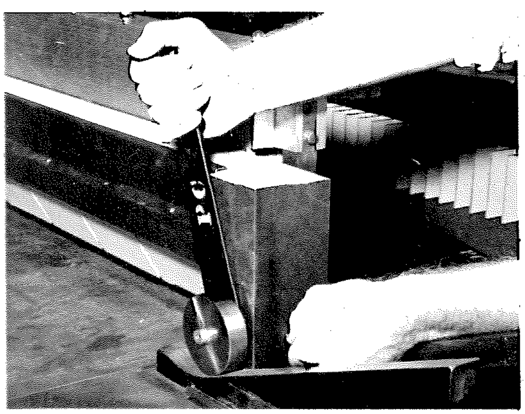
POSITION CONTROL UNIT



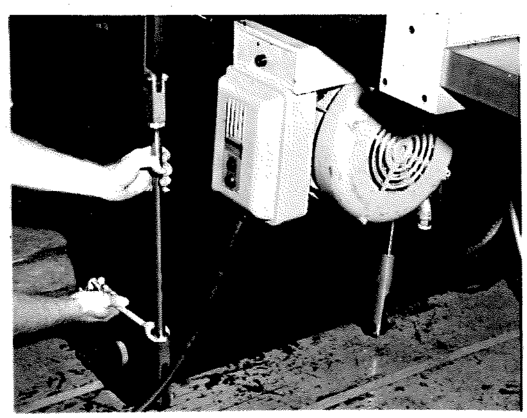
RELAY BOX



KNIFE CHANGE



KNIFE ADJUSTMENT



CLAMP ADJUSTMENT

Fig. 28. Features of PC 64

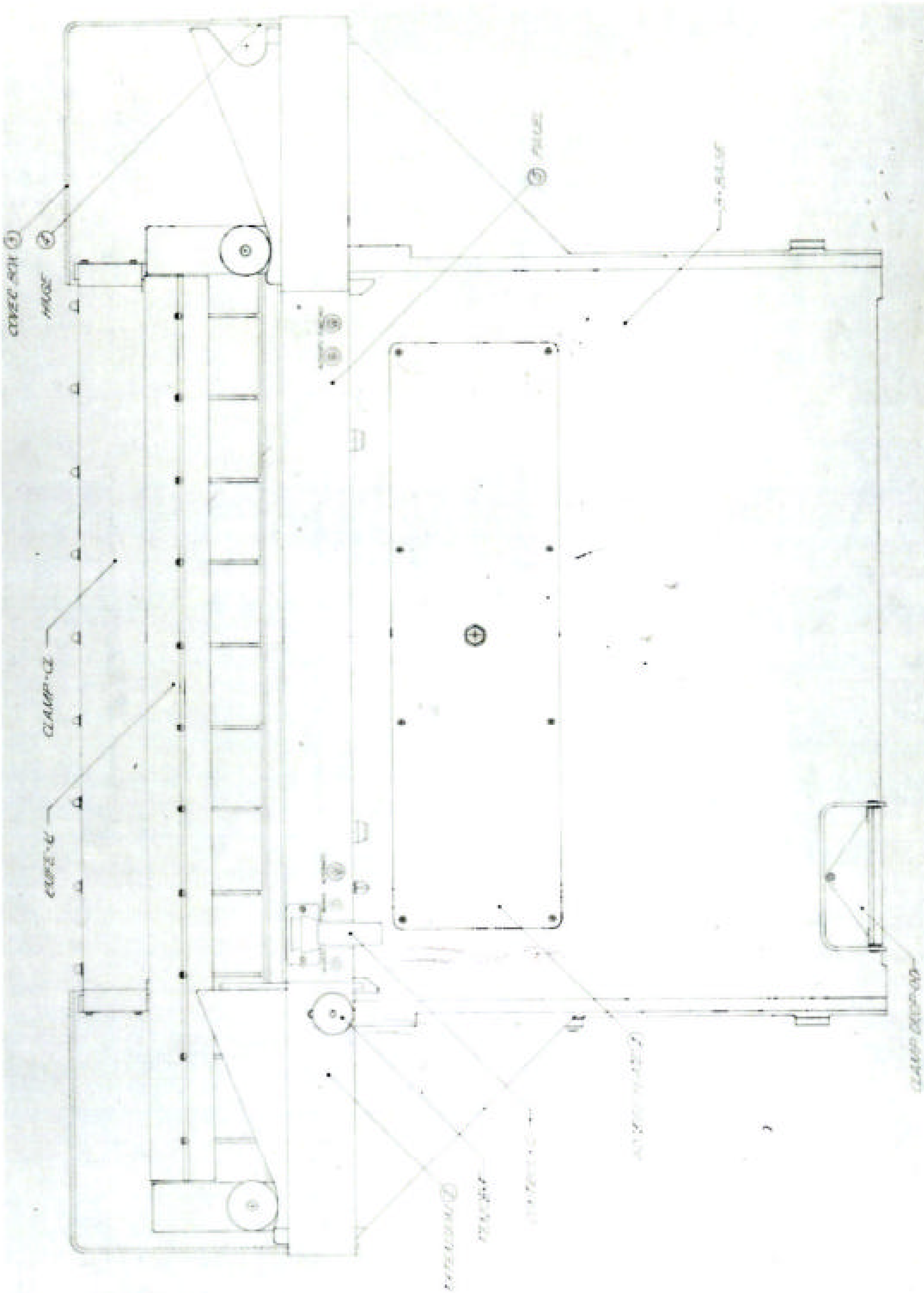


Fig. 29. Front Overall Assembly - 0

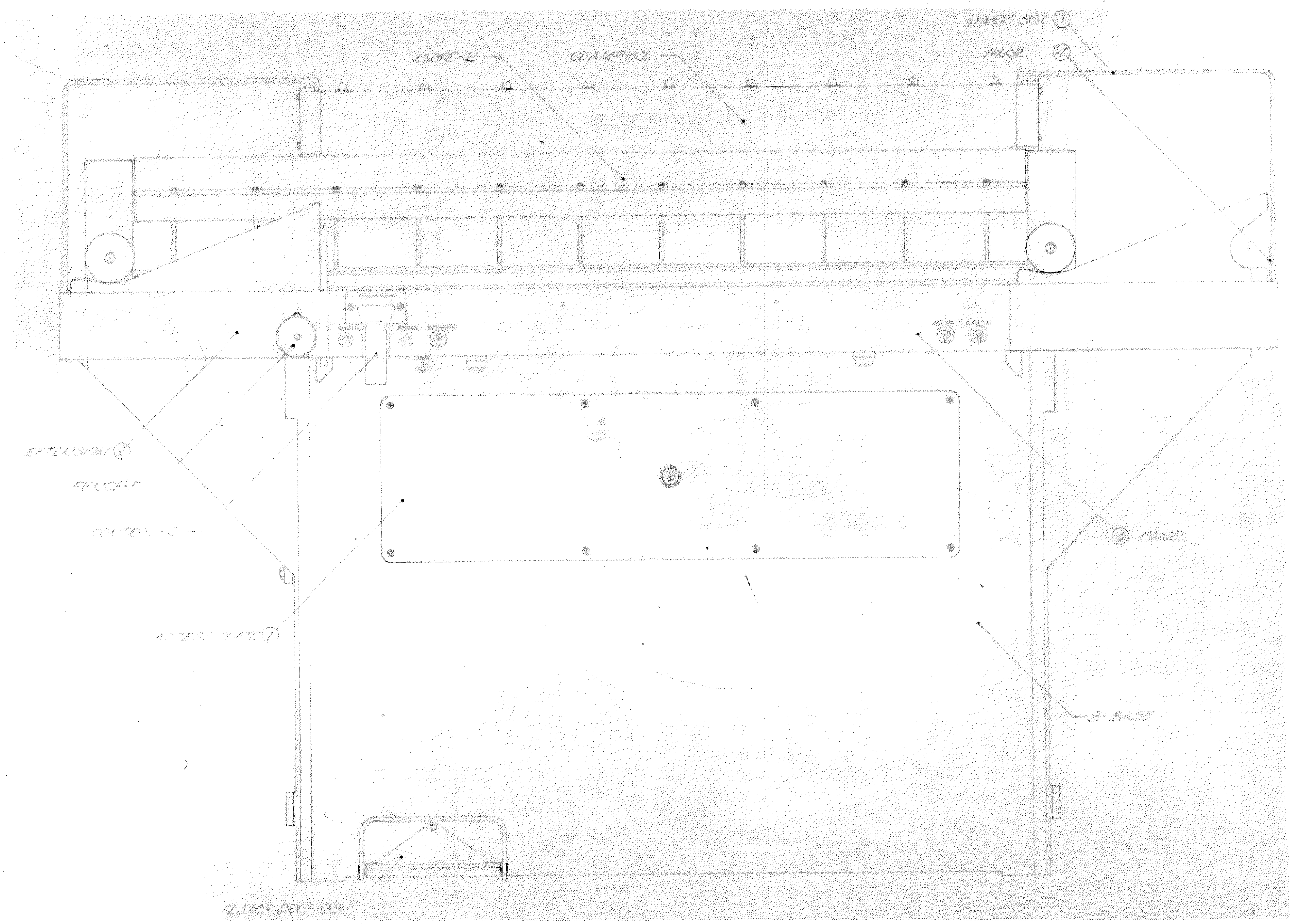


Fig. 29. Front Overall Assembly - 0

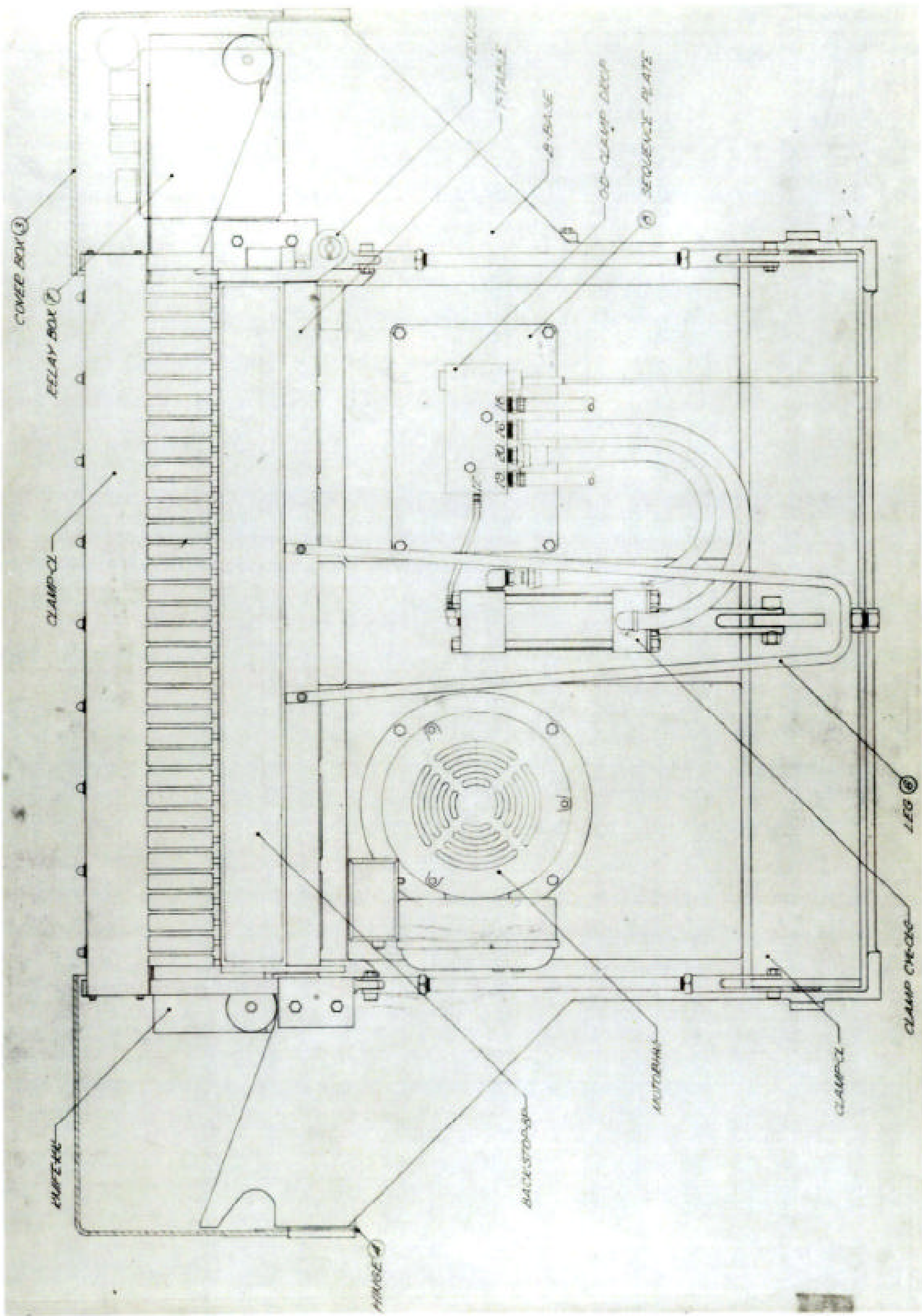


Fig. 30. Rear Overall Assembly - O

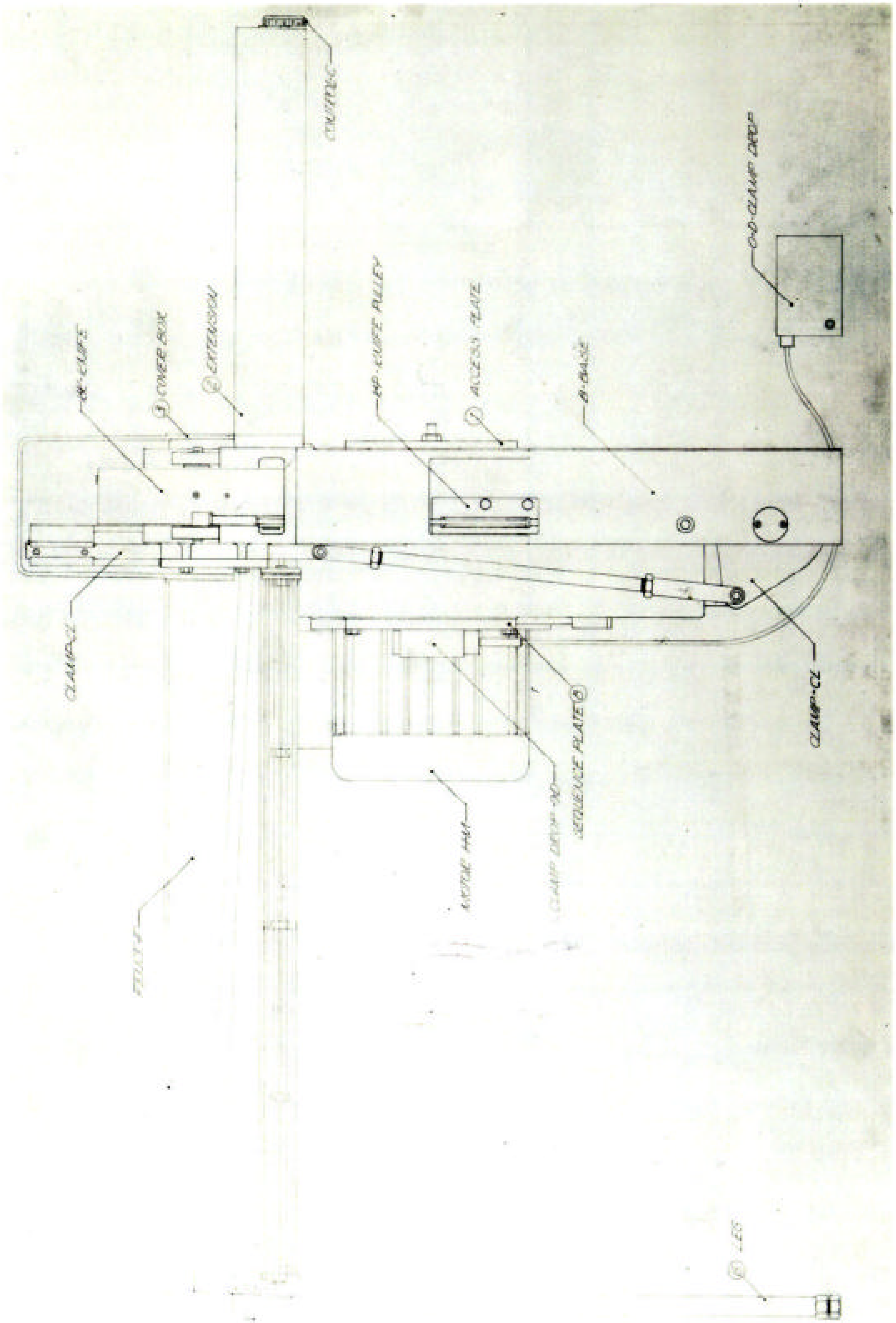


Fig. 31. Side Overall Assembly - O

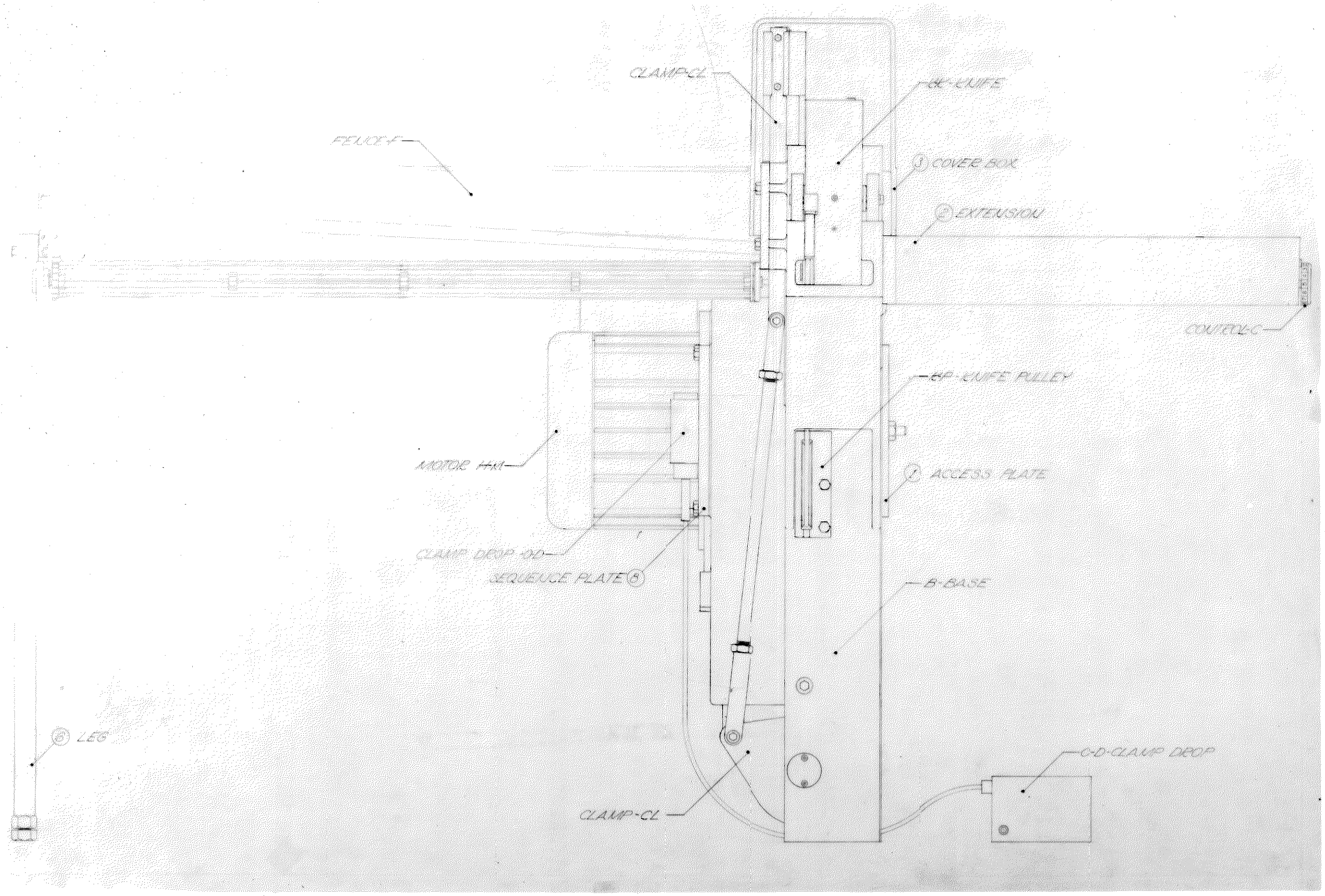


Fig. 31. Side Overall Assembly - O

V. DEMONSTRATIONS

5.1 The First Demonstration

Preliminaries

In previous bench and operating tests, some components had been found to be defective. The changes made or suggested are given below.

Knife Drive.

Shock had characterized the first tests of the knife drive. To correct this knife cylinder KCA of Figure 11 was obtained by slightly modifying the existing original cylinder following the first test of knife and clamp. The rod was shortened and the neoprene bumpers were added, involving a shortening of the hydraulic cushions. Two extra features offered by this design were:

1. The neoprene bumpers, while reducing the shock forces, would return the kinetic energy of the descending bar to the ascending bar. This would speed reversal when a cut was completed. Neoprene washers were obtained which absorbed the necessary energy during static tests.

2. Steel WASHERS (4) used to prevent extrusion of the neoprene BUMPERS (5), offered a source of additional energy dissipation for the control of normal deceleration. By maintaining a calculated fit between washer and CUP (6), a cheap means for variation of the hydraulic cushion action was available.

Backstop Drive.

It had been found that the force in the reverse mode, and the lateral squaring adjustment were insufficient. Cable rigging tension of the backstop increased the friction drag. The tension should be reduced, increased force should be provided for runback, or the moment designed into the assembly should be removed. This could be done if more space were allowed for the backstop. Temporarily, two spring loaded reels were applied to increase the backstop runback force.

Eccentricity of bushings on the backstop cam rolls was insufficient as drawn. New ones were to be substituted to provide sufficient adjustment.

Capstan Alternator.

An alternator employing a single solenoid as shown on the fence of Figure 16 was not adequate due to the size of solenoid necessary. It was intended that the solenoid zero-air-gap force would hold the capstan in one position, and a spring force would hold it in the alternate position. High preload of the spring to supply sufficient return and hold force would dictate too large a solenoid. A two-solenoid system would be a solution. The relay circuit of Figure 20 includes the circuitry for this two solenoid solution, shown as C_1 and C_2.

Readout.

Three main defects of the readout system were: the attachment of the readout tape to the pulley was not strong enough; motion of the tape on spring return was erratic; motion of the tape tended to move the tape case, which in turn controlled the backstop speed.

Epoxy had been used to join a small length of steel tape to the pulley of the readout reel. The graduated tape was slotted to attach to this. Epoxy also was used to attach the INSULATOR (19) of the control valve. An adequate bond was not obtained with the epoxy in either case; steel pins were substituted.

Spring returned action had been erratic due mainly to asymmetries inside the pulley. The pin that had attached the outer end of the coiled spring to the pulley was removed; it had provided two stable spring positions on either side. The end of the spring was then flattened and riveted flush with the inner surface of the outer rim of the pulley. In addition, a loose bushing was inserted over the pulley hub to enlarge that hub and further promote symmetry.

As shown in Figure 15, the clearance between the edge of the tape and the tape case is small. Because the tape was made to slide on the lens, the pulley has no flanges. Although countered by accurate mounting, there is a tendency for the moving tape to bind or move the tape case. The latter effect changes the speed of the backstop. Temporarily, the clearance can be enlarged by machining of the case. A permanent solution should include extra clearance or a guide for the tape. The guide could be included as part of the proposed vernier plate to be added to the bottom of the lens.

Speed Control Valve.

The switch plates of the speed control valve were not sufficiently stiff. Manual closing force applied to the valve could have the same effect as the front limit stop, i. e., cause the backstop to reverse.

In order to stiffen the switch plates mounted on the speed control valve, they were both made of steel, and their thicknesses were increased. This changed the geometry built into the cable control and required that:

(i) More change of cable length be provided by a shift of the anchor pin on the movable tape case.

(ii) Center distances on the pulley plate of the control assembly be increased. The opportunity was taken to replace the aluminum plate initially designed with a more rigid steel bend-up.

End Cover Boxes.

To resist jog forces on the faces of the end cover boxes, two sheet metal clips were added where the extension tables attach to the base. These could be replaced by bosses on the base casting. The jog forces tended to distort the hinges.

Results of the First Demonstration.

The first demonstration was not an unqualified success. When the motor was started the backstop lunged forward, and the knife rose to the top limit causing a large inertia shock. Cycle buttons were then depressed but no action followed inasmuch as the power supply was off as indicated by the fact that the green light was out. It was found that a fuse had blown.

Indicated Changes.

Air in the backstop circuit had caused the backstop to lunge. The backstop had to be checked for leaks.

Air in the knife cylinder had eliminated the effect of the upper hydraulic cushion although the neoprene bumper should have reduced the shock. A solution to the shock problem had to be found.

The fuse had blown prior to knife descent, thus the relay circuit was not necessarily at fault. To find the trouble, the electrical system had to be checked for a short circuit.

5.2 The Second Demonstration

Preliminaries.

In the first demonstration problems in the electrical system, backstop hydraulic circuit, and knife shock absorbers had been observed. These were eliminated or reduced. Some additional small changes were made and a satisfactory test resulted.

Electrical.

It was found that a short circuit in the relay box had ended the first demonstration. To avoid further failures, the whole circuit was checked and rewired.

Backstop.

Leaks in the backstop circuit were eliminated. Any softness or air in the system tends to cause overshoot beyond the programmed stop by preventing the speed control valve from functioning. Furthermore, the arm of the control valve is not designed to accept the resultant loads.

Knife Drive Shock.

The main problem with the machine was the shock at the bottom of the knife bar stroke. Knife cylinder KCA was still a possible solution and numerous neoprene washers were tested. It was concluded that a bonded neoprene sandwich might be adequate, but the prospect of further expense to modify this cylinder with no guarantee of a solution led to the design of cylinder KCB.

Knife Cylinder KCB.

Cylinder KCB is shown in Figure 11. Essentially, it is a common cylinder with a hydraulic cushion to reduce the velocity at the top of the stroke. It powers the knife until the last sheet of paper is cut, and then the relief valve fires. The 4-way valve then reverses to apply bias pressure to raise the knife.

An additional feature is that this common cylinder is attached to a spring-case of neoprene. Thus, instead of a solid shock occurring when the cylinder bottoms, the knife bar is bounced to reverse direction In addition, as the bar strikes a lift, not only is contact followed by a slow pressure build up of the oil supply, but also the spring mount attenuates the forces.

Two disadvantages were seen. One was the possibility of a shock noise when the cylinder closes prior to spring action, i. e., a noise due to the inertia of the cylinder itself. A preliminary test showed that the noise due to the velocity of this impact was not a problem. The other disadvantage was that fatigue might be a problem. No values were found to estimate the fatigue life, but consideration of the life of well-known applications of neoprene in shear indicated that the life would be considerable, as only stresses of 75 psi were allowed. In any case, even a temporary solution to the shock problem was necessary to demonstrate the other features of the machine.

Test of Knife Cylinders KCA and KCB.

Cylinder KCA was equipped with statically tested neoprene bumpers, installed, and tried but the shock was still present.

Cylinder KCB was then tried. The upstroke cushion was quiet. A shock noise was still present at the bottom. It was found that after crossing the bottom of the valley and starting up the other side, the deceleration force made the wheels jump. This region had not been analyzed in former work and this source of shock was unexpected. This led to the conclusion that the direction as well as the magnitude of the deceleration forces should be considered.

A Solution to the Shock Problem.

It was chiefly a change of direction of the deceleration force that eliminated the shock problem. An additional contribution was a package change to the pump which reduced the output. The energy of the knife bar was thereby reduced by half, as it varied as the square of pump flow.

To absorb the energy at the ramp bottom, and to change the direction of the deceleration force, a one inch thick neoprene pad was used. It was attached to the right edge of the table so that the projecting ear of the knife bar contacted it slightly before the wheels reached the horizontal portions of the ramp. The deceleration force thus was almost horizontal, and acted along a line a short distance from the bar center of gravity. In this way, the shock was eliminated. This took the deceleration function from the bottom of the cylinder. A similar solution for the top position of the knife would remove all cushioning functions from the cylinder, and lead to the cheapest possible cylinder.

Additional Preparation.

An improved adjustment for the TOP and BOTTOM limit switches was provided.

The connections at the end of the clamp-drop cable were made more rigid. This eliminated binding during spring return of the clamp-drop 3-way valve.

A piece of O-ring, found stuck in the backstop 3-way valve, was removed to allow the valve to work.

Notch seeker blades were replaced. The originals had not been hardened sufficiently.

The exhaust line to the backstop circuit was restricted to increase back pressure. This reduced the erratic behavior of that circuit, but was regarded as only a temporary solution since the circuit as built should soon be replaced by the effective circuit.

Leaks were eliminated to prevent entry of air into the system.

Programs and adjustable stops for the backstop control system were prepared.

A small length of material was machined and ground to the initial knife blade dimensions. A slight change of dimension resulted from trial of this sample in the installed bar. The final blade was ground to this new set of dimensions and was installed so that it would just cut a single sheet as the wheels passed the low point on the stroke.

Due to the relative areas of the low sides of the clamp and the knife cylinders, and the small weight of the clamp, the clamp tended to rise before the knife. In addition, when the clamp-drop feature was used, only a small force was applied by the clamp weight. Therefore the weight of the clamp was increased by adding lead shot. In a future machine the clamp could be a heavier casting, or low hydraulic pressure could be used for the clamp drop force. A small extra hydraulic circuit was applied to the low port of the clamp cylinder. A sequence valve then guaranteed the knife would rise before the clamp. Both of these devices were aimed at eliminating disturbance of the lift.

Results.

Lifts of about one inch were cut under power, and then lifts were advanced automatically for sequential cutting.

All functions except the short-stroke feature operated acceptably. Failure of this feature was due to the addition of the sequence valve to the low port of the clamp cylinder. Its effect on the overall circuit had not first been determined.

The machine was accepted as operative.

VI. EVALUATION OF THE MACHINE

6.1 Indicated Changes

Prior to extended use of the machine some changes would be advisable.

The Backstop Circuit.

1. A three way valve with large port sizes should be attached to the base. Three hoses then should connect this to the backstop. The "effective circuit" of Figure 8 will result, and the modified pilot regulator will not be needed.
2. At present a slot in the valve seat of the pilot regulator serves as the fixed orifice connecting both ends of the backstop cylinder. When the control valve is closed, a small leak does not cause forward "creep" of the backstop. An adjustable orifice should replace the fixed one, to allow for finer adjustment of the stop location.
3. More force should be applied for runback of the backstop.

Backstop Control.

1. Occasionally the blade of the notch seeker jumps from a notch. To eliminate this, the reaction of the leaf spring on the program bar might be increased.
2. Damaged notches result if the backstop reverses while the seeker blade is in a notch. As a counter measure,

the angle of the notch could be increased.

3. When the backstop is manually controlled, it will reverse if the control case is rotated too far. This is caused by deflection of the switch plates after the valve is closed. The assembly could be stiffened but a new switch design is recommended.

The Clamp.

Enlarged hoses, mounted outside, and a larger valve for the short-stroke feature could be used to increase the speed of operation.

Fence Assembly.

A different system should be used to counterbalance the paper deflector which also could be lightened with ease. In addition, the spring of the present system interferes with maintenance and adjustments under the left side cover box, and should be remounted or replaced.

Capstan Adjustment.

Corrections to the automatic backstop location with respect to the machine are now made on the valve located under the table. This is difficult and inconvenient. A threaded bushing substituted at the front of the capstan would be simpler and more convenient. At the same time the method for capstan removal should be improved.

The Cutting Forces.

Prior to any further changes aimed at reducing shock, or any modifications of the knife drive geometry, a detailed analysis of forces and geometry involved at the bottom of the ramp should be made, as this region has not yet been fully analyzed.

6.2 Other Solutions of 1964

During the later stages of the effort, it was decided to compare the new machine with similar machines on the market. Information was obtained from the Thomas Register, Foreign Consulates, and domestic and foreign manufacturers.

Representative brochures were obtained from seventeen manufacturers in six countries. These countries were: U. S., U. K., Japan, West Germany, Sweden and Italy.

One brochure, of the self-asserted world's largest manufacturer of such paper cutters, gave some indication of the market. It stated that the firm had manufactured 10,000 machines between 1949 and 1964. This would represent sales of more than $80,000,000 at the current market price.

Although some features shown in the brochures were not incorporated in PC 64 they could be added without difficulty. They are essentially independent of the basic design and could be optional at a competitive price. Examples of such features are: three-piece backstop, air table, and electronic spacers. The last of these could be added at less cost, due to the use of hydraulic backstop control in PC 64. Only an electronic sensor would be necessary, instead of both the electronic sensor and speed control of the competitors.

These observations were based mainly on an examination of the photographs reproduced in Figures 32 through 35. In addition, specification data were compiled for twelve machines and are included as Figure 36.

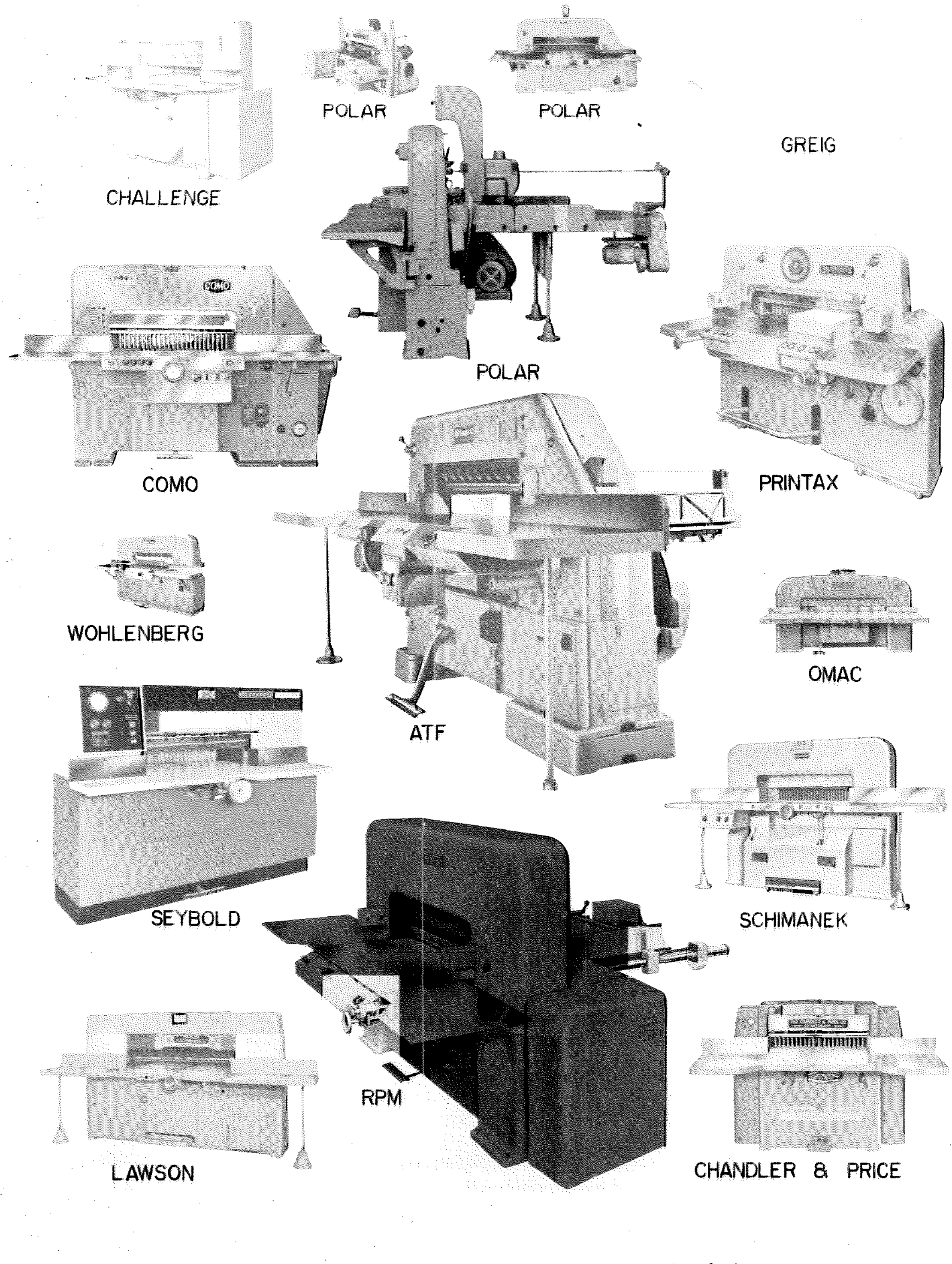


Fig. 32. Competitive Machines (1964)

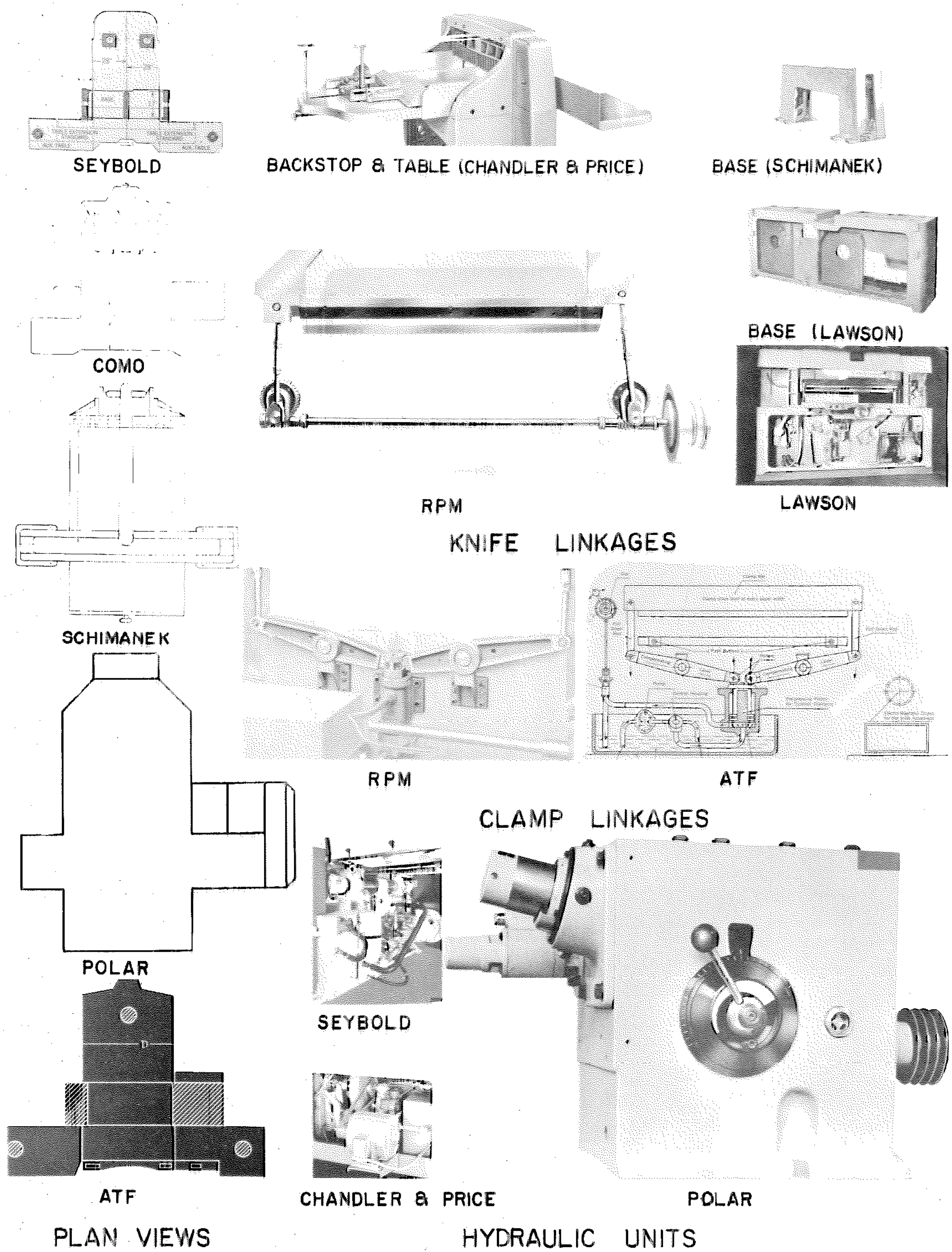


Fig. 33. Features of Competitive Machines (1964)

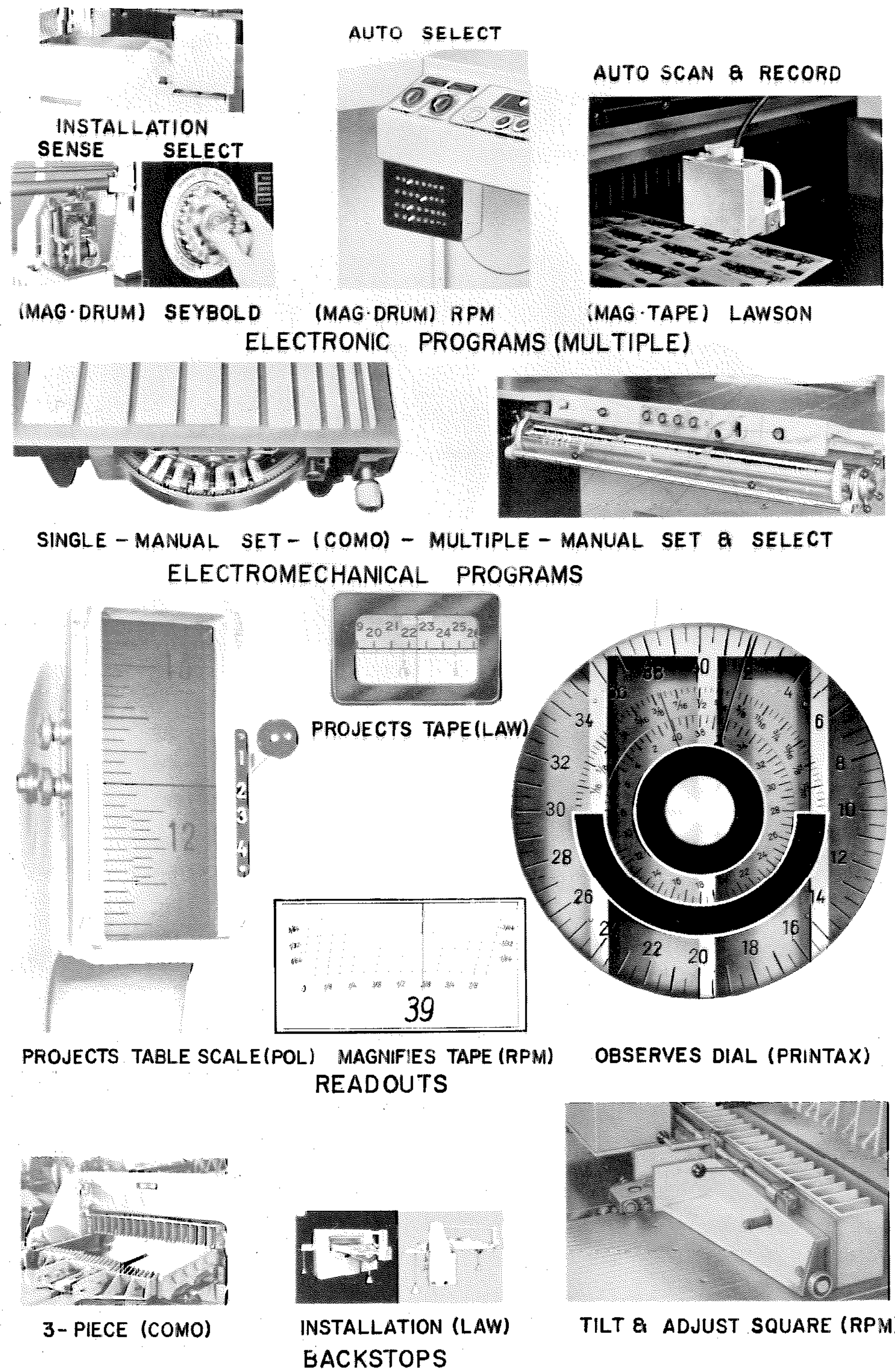


Fig. 34. Backstop Associated Features of Competitive Machines (1964)

Fig. 35. Details of Competitive Machines (1964)

APPROXIMATE DATA FROM BROCHURES

		RPM	Wohlenberg	Schimanek	Greig	Lawson	Seybold	Polar	Printax	ATF	Como	Chandler & Price	Challenge
Stock (in.)	width	42	42	42	42.5	42	42	42	42	42	43.75	37	37
	length	42		42.25	44.5		42	42	42	41.75	43.25	37.25	38
	height	5.5	5	5.75		6.5	5.5	5.25	4.75	4.75	5.12	4	4.5
Weight (lbs)	net	6170	6200		8000		5200	6285	4400	5400	5500	3400	
	gross						6400	6500	5200	6150	6200	3775	
Drive (hp)	motor	5.5		6.25	3		3	5	4	4	5	3	
Clamp Force (lbs)	min	660	660						1650	650			
	max	8800	6600		8500		6000		6600	8200			
Last Cut (in.)	plate	3 15/16			3								
no	plate	13/16		3/4	3/4		3/4			1			
Speeds (in/sec)	gauge	4 3/8			5.8		9						
(stroke/min)	knife	35	42		40	50	35	42	30	42	40	40	40
Overall (in.)	width	86		79	82.5		87.5	84	70	83.25	81	85	73
	height	61.75			68.5		65	60	55				
	depth	92.5		87	94.5		94	96	77.5	86.6	86	77.5	76.75
(extended)	width	109		120			122				93.5		
(auto)	depth	100											
Table (in.)	height	31.5		38.25	38.25		38			36.9	34.5	36	
	front	25.8		21.8			22.5	27	21.8	24.5	24.5		
Cut (± in.)	accurate	.004				.001				.002	.004		
(°)	shear		40	53			53						
	trim			1/16									
	sequence			1/16									
	stick						3/4 x 3/4					1/2 x 1/2	
Knife (in.)	dimension						53 x 5 x ½					44 x 4 x 7/16	
	bevel			23			26						
	grind			1 1/16			1 7/8						
Clamp (in.)	width			4		4	3						
Tape (in.)	graduate			1/16		1/64	1/16					1/16 & 1/12	
Base (in.)	depth				24 3/4					20	19 3/4		
Oil (gal.)	supplied						20						
Door (in.)	width											30	29
Price (U.S. $)	standard	7110			4718					7825		5825	
	auto	{8585 9360				10,375				10,285			6425
Port		N.Y.			U.K.	L.A.				L.A.			L.A.

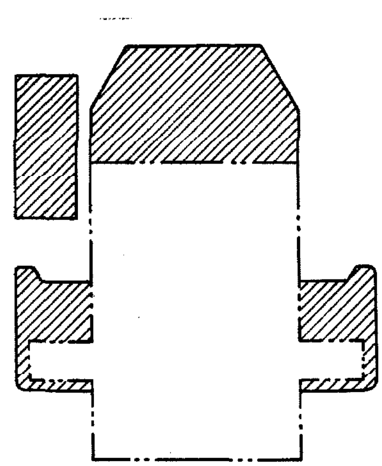
Fig. 36. Specifications for Competitive Machines (1964)

The photographs included were selected from a larger number available and represent any competitive features considered to be noteworthy or advantageous. These features should be reviewed prior to commercial use of the design of PC 64. All are explained in the brochures of the machines shown in Figure 32.

Comparison With the Lawson Machine of 1960

Weight of the 46 in. Lawson Machine with a powered backstop was given as 12,100 lbs. For an equivalent PC 64 the weight is approximately 3,000 lbs. The automatic spacer option adds 1,000 lbs. to the former, 10 lbs. to the latter.

In the sketch below, the floor plan of a 46 in. PC 64 is superimposed in phantom on that of the Lawson. Optionally sized extension tables are omitted in both cases.



The shaded areas represent the extra floor area needed with the Lawson machine over the needs of PC 64. Further visual comparison of the two machines can be made by referring to Figure 6 and Figure

7 for the Lawson, and Figure 27 and Figure 28 for PC 64.

The only feature of the Lawson apparently not considered in PC 64 is the manually powered backstop. Reasons for this action were given during description of the preliminary design decisions.

Provisions for punch-press and short clamp stroke operation are included in PC 64, but not in the Lawson machine.

Although the cost of manufacture of either machine has not been determined, if number, weight, complexity and cost of manufacture of components are indications, PC 64 is considerably cheaper.

6. 3 The Next Step

The machine as built is not ready to be sold. In order to prepare for that possibility, the changes indicated in section 6. 1 should be made.

Operating tests should then be directed at three main areas:

Knife Drive

The magnitude and direction of all forces involved in cutting should be determined. Various blades, including those with diamond dust plated on the edges should be used.

Based on the results, some modifications to the knife drive may be necessary to eliminate the possibility of the wheels leaving the ramps during a cutting cycle with an acceptable knife edge.

In addition, some change of knife bar size and shape may be possible. These changes should be towards lighter construction.

Clamp Force

Optimum clamping forces for various types of papers should be determined. In addition, the nature of the clamp force buildup prior to and following knife descent should be determined.

It should be noted that with the sequence valve circuit used pressure builds up in the clamp cylinder to the set value, then oil flows to the knife cylinder. If the knife cylinder requires less pressure than that set for the clamp, the clamp force will remain constant while the knife descends. If the knife requires greater pressure than the sequence valve setting then the clamp force will increase accordingly. This shows that the knife cylinder should be slightly larger than necessary.

The tests should provide a means for determining a knife cylinder large enough to allow constant clamp pressure as set, but small enough that the speed of descent is not unnecessarily reduced.

Extra Operations

Requirements for the cutting of plastic and other sheet materials should be obtained. Punching and stamping operations commonly done in a print shop should be investigated also. As a result it may be found that further control of the knife and clamp bar speeds, and of the clamp forces will be required.

Also, this should furnish information for the design of attachments which can use the clamp unit of PC 64. Other print shop machines may be designed to use the hydraulic unit.

6.4 A Postulated Commercial Model

If the work outlined above is performed, the resultant machine will be similar to PC 64, and a large amount of cutting data will be available. Except for the knife bar and clamp, most changes will be contained inside the base casting. The machine should be dependable and suitable for extended use.

At the outset it was stated that PC 64 "should be suitable for immediate production of 100 cutters per year". Based on the improvements that are now apparent, this has not been achieved. As it stands the machine represents a large expenditure of time and effort and a certain amount of capital (estimated as less than $20,000). This capital is a small fraction of the amount needed to establish production on a profitable basis. No tooling has been acquired. If used as a production model all details would be scrutinized and some changes made. Since in most instances it is not desirable or possible to isolate portions of this machine and to "optimize" them separately, a totally new layout should be started with careful scrutiny of each part.

It is recommended that the design of this "postulated commercial model" be based on consideration of PC 64, data derived from tests, and the suggestions offered below. These suggestions may be regarded as opinions supported by some experience, or alternatively, as criteria for the next layout.

Accept but Improve:

(1) Basic hydraulic circuit
(2) Basic electrical circuit
(3) Extensive block manifolding
(4) Sequence Set principle
(5) Short Stroke principle
(6) Knife drive system
(7) Clamp assembly
(8) Control valve and notch read principle
(9) Capstan and bar concepts
(10) Paper deflector
(11) Manual control and readout assembly
(12) Panel for switches (removable)
(13) Panel for relay circuit (removable)
(14) Knife blade

Reject:

(1) Backstop hydraulic circuit
(2) Needle valve for short stroke
(3) Cushions in knife cylinder
(4) Fixed non-tiltable backstop bar
(5) Castings for end cover boxes
(6) Force closure to balance backstop drive
(7) Clamp cylinder outside reservoir
(8) Table as just a flat slab

VII. CONCLUSIONS

A. The Design Process

Although no general rules were used in making the final design decisions, a recognizable pattern emerged during the work. Because decisions generally were made by choosing among several feasible alternatives, it was necessary to have standards for comparison. The thesis objective dictated that the primary standard throughout the work be construction and operating costs. In order to apply an economic yardstick effectively to the design of the machine, when total costs could be estimated only roughly, it was necessary to introduce approximations to a pure dollar standard. Depending on the function of the machine component, a single measure such as simpler, lighter, more efficient, less wear, easier to maintain, easier to install, occupies less floor space, requires less production tooling, easier to sell, etc., often could be equated to "cheaper". Thus the clearest decisions were based on approximations that allowed measures of simplicity to be used as equivalent to the fundamental unit of dollars. In addition to these measures of optimality, whose applicability depended on the problem in question, there were analytical measures such as stress, strain, currents, flows, etc. which were dealt with according to rules and facts available from standard references or experimental data. When these analytical measures were applicable, they often could be equated to the primary economic measure, thereby serving as a standard for a design decision.

When more than one approximation of cost was applicable, conflicts sometimes arose. Clear decisions were impossible when, for example, one alternative was simpler, required less maintenance, but was less efficient than another alternative. In such a case a new alternative was sought without the latter disadvantage, the problem was rephrased so that the conflicting measures could be uncoupled, estimates were used to find the effect of the disadvantage, or as a tedious last resort, actual approximation of each of these factors in terms of dollars was attempted.

Sometimes an increase rather than a reduction of variables or criteria was made in order to clarify a decision. An example of this method can be noted if the decision to disregard the band-saw type of machine is recalled. The primary measures had seemed to be light overall weight of the machine coupled to poor efficiency of the cutting process itself. Nevertheless, a dollar analysis might have justified its use. However, addition of the criterion that a finished working prototype had to result in an estimated three years was used to rule out further consideration of the band-cutter system.

From the above it is recognized that decisions in the design of coupled components involving a large number of criteria require more effort than those for which only a small number of factors need be considered. This is clarified further if the problem of improving to the same level of sophistication the paper deflector device of Figure 16 is compared with that of the control valve c-v of Figure 17. In the former it is relatively easy to consider certain features individually. In the latter it is much more difficult due to the number of parts, features, properties, functions, and forms which are coupled not only within the assembly, but also to other aspects of the overall machine.

One particularly important feature of the design process is the complex interrelations of many machine components. This concept of coupling as indicated in the above comparison between the paper deflector and the control valve was the principal reason that it was impractical to employ a simple, deductive design procedure. Instead, a list of minimum functions for the whole machine was envisioned. To accomplish these functions, several alternative minimum means were available. For example, the cutter itself could have proceeded from a minimum band-cutter, minimum guillotine-type cutter as it did, or others which could be imagined. From the basis of the minimum components which were necessary to the guillotine cutter, progress was then made towards means sufficient to accomplish the minimum functions. The choices available for mechanisms sufficient to make a workable machine depend on the present state of technology, and the criteria for choosing among the possible alternatives must be equivalent to total cost in a project of this type.

B. The Problem and the Machine

The problem of precision cutting of rectangular lifts of paper has been examined, along with representative solutions in the form of guillotine-type production paper cutters. In addition, studies and tests aimed at determination of the parameters involved in the cutting action have been described. Based on the information acquired in this way, present technology has been applied to the design of a new solution, one that would be more economical for the users, mainly printshops and paper mills, to own. This has resulted in the design of PC 64, a new guillotine paper cutter of 42 in. by 4 in. capacity, which weighs less than 3, 000 lbs, has a low shear angle of 21°, and which occupies a minimum of floor space. It has been compared with a 1960 model, whose functions it was designed at least to duplicate. The 1960 model weighs approximately four times as much as PC 64 and occupies approximately twice the floor space. A prototype of PC 64 has been built, and has been shown to operate as designed. Based on the work described and the material included in this thesis it is the writer's opinion that a production model of PC 64 can be offered at less cost than the competitive models, including those of 1964 which have been seen. The market place affords the only final proof.

It is concluded from this work that the problem of cutting paper has sufficient scope and importance to merit considerable further study. However, based on the design of PC 64 it is concluded that further intensive study from the limited point of view of the guillotine cutter, other than to perfect a machine like PC 64, does not offer considerable further gains. A main reason for this is to be found in the basic nature of the guillotine cutting action and in the fact that the stock is moved along a table by the backstop. Most of the gains realized in PC 64 are in the form of reduced size, complexity, weight, and cost, and it is concluded that in a perfected version of PC 64 ready for production, the further potential of these sources of improvement would be slight. The remaining source of improvement would be in the speed of operation, and in this respect, the guillotine solution is severely limited, not withstanding the fact that reduced weight allows increased speed of operation of the knife and clamp.

In their usual order of importance, the speed of output is limited by: loading and unloading time; time to advance the stock between cuts; and time of cut or clamp-knife cycle time. Loading and unloading time is best attacked by the design of special fixtures adapted to the specific needs of high volume jobs where they exist. As long as printshops deal with a multiplicity of relatively low volume jobs no simple answer to this problem is foreseen. With the present backstop system, the speed of advance is limited by the maximum permissible deceleration. In turn, the maximum deceleration is determined by the coefficient of friction between the sheets to be cut, and this cannot be changed. The most promising avenue of approach for the development of a high speed cutter is to leave the stock stationary, lightly clamped at the last edge to be cut, and to move the cutting device itself, i. e., move the plane of cut. For obvious reasons, unless a new method for cutting other than by means of a mechanical device is found, a lightweight, low force, high speed device, rather than the heavy, high force, low speed, guillotine action should be sought. An additional feature contributing to the light weight of such a device will be the fact that the weight of paper to be moved away from the cut, following a cut, will be due only to the last small lift separated, not to the accumulation of all cut lifts not previously removed by the operator from in front of the blade. In this way it will be feasible to cut completely a large lift before removing any of the resultant cut stock. Therefore, it is concluded that future research in this field should be directed at the development of a low force, high speed, cutting system, such as a vibratory blade, to allow consideration of advanced alternatives to the guillotine type cutter.

Until such new cutting systems are developed, it is concluded that the principles embodied in PC 64 are adequate upon which to base a commercial model. This model should include the all-hydraulic features of PC 64 and refined versions of the other main features included. Particular changes to details of the design of PC 64 will depend on further operating tests and on a detailed study of the conclusions on which this design was based.