MEASUREMENT OF CYLINDER DEFLECTION INCLUDING FINITE ELEMENT ANALYSIS

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Abstract

Hallmark Cards has research and development projects which are concerned with the nip pressures in sheet fed, large format lithographic presses. The purpose of this report is to describe some of the methods which were used to determine cylinder deflections during a recent study of the factors which affect loading of the blanket against the sheet.

The purpose of the study was to accurately determine the unit forces which act on the sheet during printing so that they could be correlated against the other factors which control the quality of print. Three methods were employed in order to derive the unit loading: (1) empirical testing involving prototype modeling of full-sized cylinder sections, (2) calculations using classical mechanics as the basis for selecting several mathematical models which "bracketed" the values of expected deflection, and (3) a finite element solution which utilized detailed computer modeling. The three separate approaches to the same problem helped to provide a system of checks and balances to prevent the propagation of errors.

The investigation showed that for increasing wall thicknesses in a 43" x 60" sheet fed press cylinder, the deflection mode is initially one of localized radial deflection at the nip (crush) rather than axial beam deflection. By increasing the wall thickness to the point where crush is not a problem, beam deflection is automatically taken care of . Attempts to minimize rotational inertia can be best accomplished by stiffening the cylinder using a variety of internal supports.

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Introduction

One of the controlling factors in being able to print consistently high quality work is the ability to maintain uniform squeeze between the blanket and impression cylinders on a press. The ideal condition would be to have the cylinders so rigid that the squeeze would be only a function of the blanket and packing. That is, the squeeze would be the same in the center of the sheet as it is on the edges.

We know that no mechanical member is absolutely rigid, particularly the cylinder of a printing press which is a reinforced steel shell fabrication. In order to determine in a non-destructive manner the deflection which might be expected from the loading caused by a sheet being printed plus the amount of interference against the resilient blanket, it was necessary to construct some laboratory fixtures which determined the effective spring rate of that system. The results from the lab tests were then combined with measurements taken from a press nip to determine the unit loading which would be attained during printing. Having obtained representative unit loading in pounds per linear inch of nip, it is then possible to use mathematical relationships from conventional mechanics, or more detailed analysis using computer programming, to predict any cylinder wall deflection given the dimensions and materials used in constructing the cylinder.

Determination of Cylinder Loading

The success of any engineering calculations of cylinder deflection requires the accurate determination of unit loading, the pounds of force per linear inch of printing nip. The objective was achieved by a combination of two efforts. First a sheet fed press was measured in order to determine the blanket squeeze after a complete makeready had been performed. Next a laboratory fixture was constructed in order to measure the spring rate of the combination of blankets and packing which were used in the press run from which the measurements were made.

The blanket-to-impression cylinder nip was selected for evaluation. The press station was made ready as if a production job were going to be printed. A diagnostic plate was selected in which half-tone and solid areas could be examined for good dot formation, and densitometer readings could be taken. The press was brought onto impression and sheets were printed. Initially, the squeeze was so small that the printed image did not have satisfactory density, but the squeeze was increased until the quality of print was up to Hallmark standards.

After satisfactory print quality had been achieved over the entire sheet, the press was stopped and small patches were cut from the blanket, underblanket, and packing. The patches were taken from locations where cylinder stiffeners were known to exist and not exist. The press was then jogged over until the patch locations coincided with the printing nip. Feeler gages were then used to determine the metal-to-metal gap between the blanket and impression cylinder bases. The measured gap was compared with the total thickness of the blanket, underblanket, packing, and stock which was printed. The difference was the squeeze required for good print quality for those materials.

The next step was to establish the spring rate, the amount of force required to displace the blanket a specified distance. The spring rate was measured by constructing a small fixture which duplicates the press nip. The fixture (shown in Figure 1) has upper and lower anvils which are machined to the same radius as the press cylinders, approximately nine inches. The blanket and packing may be tensioned by a method which closely resembles the blanket tensioning shaft on a press.

After four inch wide sections of blanket and packing have been set up in the fixture, the entire assembly is mounted in an Instron compression tester. As the anvils are loaded against one another, the forces are measured by a load cell, and the displacements are measured by feeler gages, just as they were on the printing press. Several samples were tested in order to minimize the errors due to variations in gaging techniques. The data points were averaged and the slope of the graph of load as a function displacement (Figure 2) represented the average spring rate of the system. It was interesting to observe that significant variations in the blanket loading did occur due to variations in either the blanket material, the packing, or both. It is also important to remember that the press tests and the lab tests were static tests. One would expect dynamic measurements (had they been made) to be somewhat higher due to the lack of material creep, among other things.

Several combinations of blankets were tested. The results showed that the loading at the nip could be as small

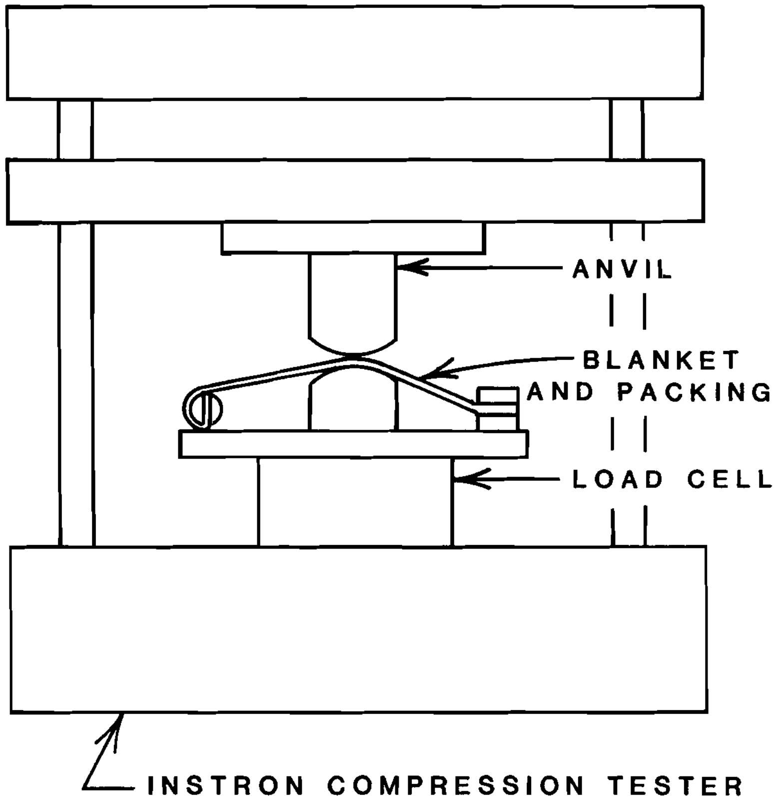


FIGURE 1

FIXTURE TO MEASURE SPRING
RATE OF LITHO BLANKET

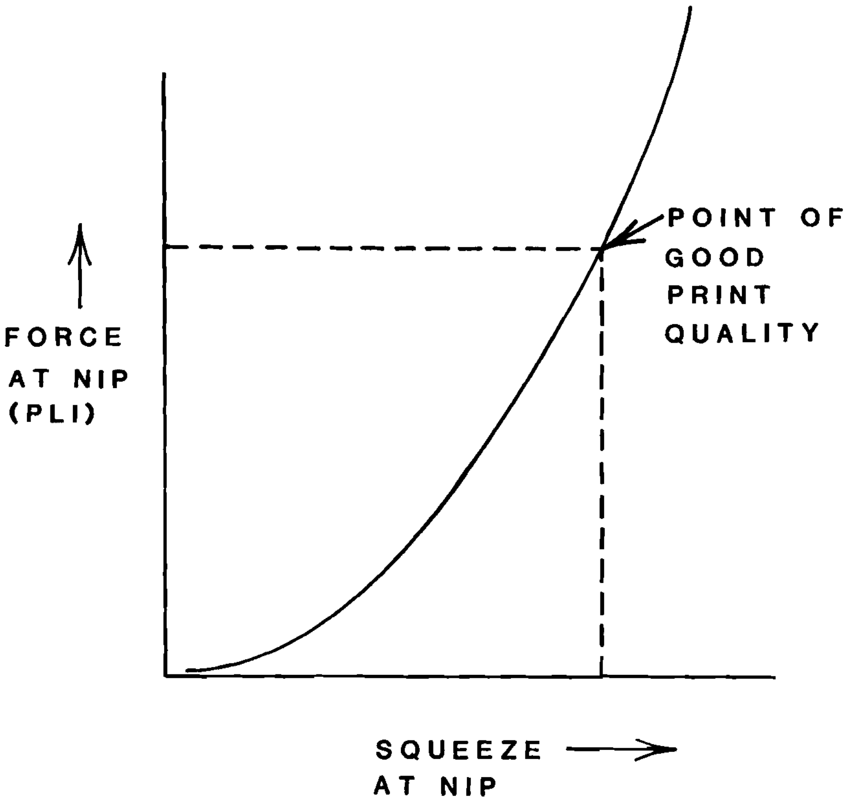


FIGURE 2

TYPICAL LOAD VS. DISPLACEMENT
FOR A LITHO BLANKET

as 70 pounds per linear inch for a compressible blanket to over 300 pounds per linear inch for conventional blankets. The selection of a blanket with a low spring rate will substantially reduce the bearing loads in a press.

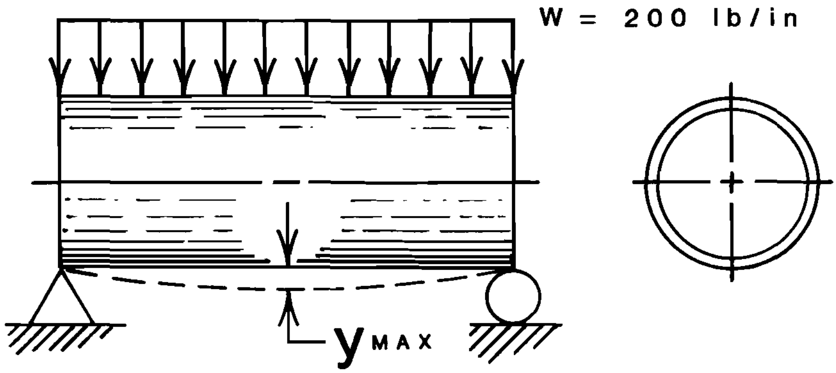
Calculation of Cylinder Deflection

Deflection calculations of a point on a sheet-fed press cylinder under load are extremely complex. The end constraints (bearings), the presence of gears and bearers, the gap, and all the internal supports contribute to making the solution in closed form nearly impossible. As a start, however, one can make some simplifying assumptions and calculate the deflections for known configurations in order to determine whether there is even a chance of there being a relevant deflection. It is usually known whether the assumption would give a falsely liberal or conservative answer.

The most simple case would be to assume the cylinder behaves as if it were a beam made from a hollow piece of pipe with open ends. Figure 3 shows that deflection mode for the theoretical case of a steel cylinder having a length of 60 inches, an outside diameter of 18 inches, a wall thickness of 1 inch, and a uniform load of 200 pounds per linear inch. The beam deflection shown in Figure 3 requires the cross-sections of the cylinder to remain circular.

If one were to look at the end of the pipe shown in Figure 4, another form of deflection becomes evident. Using the same conditions as in Figure 3, the deflection causes the pipe to become non-circular, and it is additive to the beam deflection described in the previous example. It is important to note that the deflections for the examples used show that the tendency for the cylinder to "crush" (Figure 4) is greater than the tendency to "sag" (Figure 3). This condition exists because the ratio of the length of the "beam" (width of the cylinder face) to its cross-section (diameter) is relatively small. The calculations shown in Figures 3 and 4 demonstrate that the designs of cylinders of these dimensions should be concerned more with radial deflections than axial deflections.

Figure 5 demonstrates the case of wall deflection on a cylinder with capped ends using the same dimensions and loads as in Figures 3 and 4. This example comes closer to describing the structure of a printing cylinder as well as the resultant deflection.



$$y_{max} = \frac{5}{384} \frac{wL^4}{EI}$$

y_{max} = Maximum Deflection at Center of the Beam

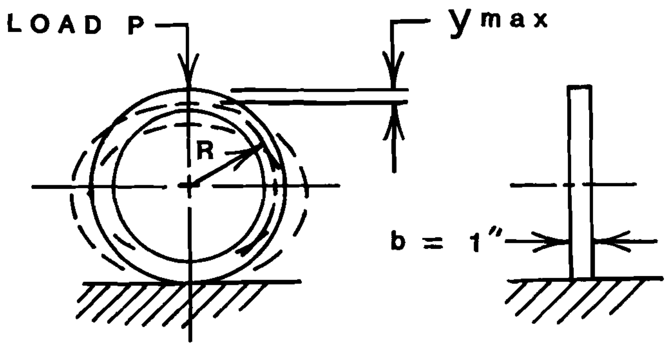
W	= Distributed Load	200 lb/in.
L	= Length of Beam	60 in.
E	= Young's Modulus	30×10^6 psi
I	= Moment of Inertia = $\frac{\pi}{4}(R_0^4 - R_I^4)$	
R_0	= Outer Radius	9 in.
R_I	= Inner Radius	8 in.

$$y_{max} = \frac{5}{384} \frac{(200)(60)}{30X10^6 \times \frac{\pi}{4}(9^4 - 8^4)}$$

y_{max} = 0.000581 in.

FIGURE 3

HOLLOW CYLINDER CALCULATION



$$y_{max} = \frac{PR\pi}{4\,Ea} + \frac{PR\pi}{4Ga} + 0.3\,\frac{PR^3}{EI}$$

y_{max} = Maximum Deflection, "Crush"

P =	Load	200 lb.
R =	Mean Radius	8.5 in.
E =	Young's Modulus	30×10^6 psi
G =	Modulus of Rigidity	12×10^6 psi
I =	Moment of Inertia	$b(R_O - R_I)^3/12$
a =	Area of Section	$1 \times 1 = 1$ in^2
R_O =	Outer Radius	9 in.
R_I =	Inner Radius	8 in.

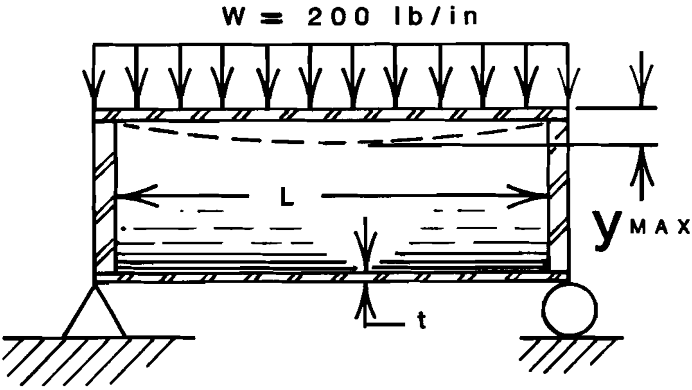
$$y_{max} = \frac{200(8.5)\pi}{4\times30\times10^6\times1} + \frac{200(8.5)\pi}{4\times12\times10^6\times1} + \frac{0.3(200)(8.5)^3}{30\times10^6\times\frac{1}{12}(1)(1)^3}$$

$$= .000045 + .000111 + .007369$$

$$y_{max} = .007525 \text{ in.}$$

FIGURE 4

RADIAL DEFLECTION OF
HOLLOW CYLINDER



$$y_{max} = .0305\ [12(1-\upsilon^2)]^{5/8}\ R^{3/4}\ L^{3/2}\ t^{-9/4}\ \frac{W}{E}$$

y_{max} = Maximum Deflection at Center

υ =	Poisson's Ratio	0.27
R =	Mean Radius	8.5 in.
L =	Length of Beam	60.0 in.
t =	Wall Thickness	1.0 in.
W =	Uniform Load	200.0 lb/in.
E =	Young's Modulus	30×10^6 psi

$$y_{max} = 0.0305\ [12(1-.27^2)]^{5/8}\ (8.5)^{3/4}\ (60)^{3/2}\ \frac{(1)^{-9/4}(200)}{30\times10^6}$$

$$y_{max} = 0.002120$$

FIGURE 5

RADIAL DEFLECTION OF CYLINDER
WITH CAPPED ENDS

Finite Element Analysis

The previous discussion has been helpful in pointing out where the weaknesses of a cylinder might be found. The calculated values of the deflections, while correct for the assumed structures, do not tell us very much about the behavior of an actual press cylinder. More accurate results may be obtained with the addition of calculations which consider internal bracing, but an accurate model would be difficult to solve. Another consideration is time. One should not invest large amounts of engineering time in the pursuit of a mathematical model whose validity is subject to question.

Hallmark selected the alternative which many engineering groups are using, finite element analysis. The ANSYS program was considered to be well suited for the types of calculations which were needed for analyzing a printing press cylinder. The use of finite analysis consists of building a computer model of the desired structure, applying constraints, loads, or known deflections as boundary conditions. The result is a framework of nodes and joining members which looks like a skeleton of the part being analyzed. Figure 6 shows one print-out of a cylinder being modeled using finite element analysis.

The advantage of the technique is that extremely complex structures may be analyzed. It is possible to incorporate internal braces, bearing loads, variations in nip loading, different materials, and dynamic effects into a finite element analysis. Further enhancements are having the weight, moment of inertia, and localized stresses calculated. The cylinder may be "turned" by the computer in order to determine the effects of the cylinder gap and internal bracing on cylinder deflection. In general, the more nodes selected, the more accurate the results. The program provides for being able to print out a picture of the model without performing the analysis as a check of the validity of the construction, thus saving the cost of a computer run if a problem does exist.

The disadvantages of finite element analysis are that the results are no better than the model. The computer will generate an output if there are no syntax errors in the program, but the calculated results may have significant inaccuracies due to inadvertent errors in constructing the model. The technician alone must decide whether the results seem feasible. The addition of nodes greatly increases the

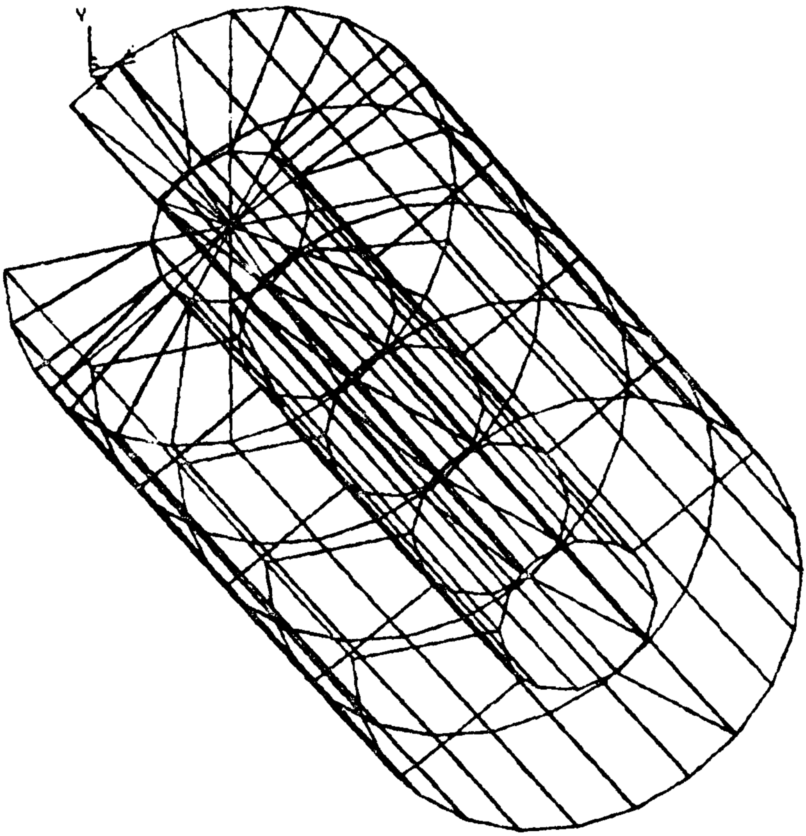


FIGURE 6

NODES OF FINITE ELEMENT
ANALYSIS

amount of work required by the computer . . . and the expense. Items which have symmetry are the easiest, and least expensive to analyze. Problems involving large numbers of nodes must be solved on large computers. Since the process involves the solution of many simultaneous equations by matrix inversion, the computation time with a small computer becomes excessive.

Conclusion

The engineering analysis of a printing press cylinder as a structure demonstrated the use of three stages of evaluation. The first stage was the gathering of data from printing press measurements and laboratory equipment designed to simulate a portion of a printing press. The second stage was to perform calculations from known examples which resemble the desired structure as closely as possible, realizing that the simplifying assumptions can give only approximate results. The third stage was to use the experimental data gained in the first stage to model a computer solution by finite element analysis.

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