INTERN EXPERIENCE AT PACKAGES LIMITED

An Internship Report

by

REFAAT SHAFKEY

Submitted to the College of Engineering
Texas A&M University
in partial fulfillment of the requirements for the degree of

DOCTOR OF ENGINEERING

August 1985

Major Subject: Mechanical Engineering

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ABSTRACT

INTERN EXPERIENCE AT

PACKAGES LIMITED. (August 1985)

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This report describes the internship experience of Refaat Shafkey at Packages Limited, Pakistan, where he worked as Senior Design Engineer from March 17, 1984, to January 14, 1985. The internship was undertaken to fulfill the requirements of the Doctor of Engineering degree at Texas A&M University.

The intern worked in the Design and Development department of the Technical Division. The department served as an in-house consultancy for the production sections of the company. The technical assignments during the course of internship covered new developments, modifications and maintenance related functions. This provided exposure to problems in various sections, each subject to different technical and non-technical constraints. The exposure to cost estimation, industrial communications, and management decision-making were all a source of professional development. The internship provided a valuable "real-life" addition to the intern's education.

ACKNOWLEDGEMENT

I wish to express my sincere gratitude to Dr. Don Bray, my program advisor and committee chairman, for his invaluable advice, support and expert handling of so many intricacies associated with this program. Thanks are also due to Mr. Sheikh Suleman Elahi, my internship supervisor, for his guidance and support during my internship at Packages. I am also grateful to my committee members: Dr. Ray Goforth, Dr. Tom Tielking and Professor Stan Lowy for their assistance and understanding.

Finally, I would like to thank my family members for their constant help and encouragement which provided the motivation to carry out this work to its conclusion.

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CHAPTER I

INTRODUCTION

The report describes the intern's experience at Packages Limited to fulfill the internship requirements of the Doctor of Engineering degree. Packages has considerable experience and expertise in the paper, printing and packaging industries both in Pakistan and abroad.

The job assignments for the intern, who served as Senior Design Engineer, were usually technical in nature. The exposure, however, was both technical and non-technical. The technical contributions were not confined to one or two narrow problems, but covered a variety of jobs for different departments of the company. This required making educated decisions in a typical industrial environment characterized, sometimes, by incomplete and even inaccurate information.

The non-technical exposure included interaction with various departments to understand problems and extract pertinent information useful for subsequent design and development. The internship provided an opportunity to see problems in industry as a whole and to understand an engineer's role in tackling such problems.

INTERNSHIP OBJECTIVES

To fulfill the College of Engineering requirements at Texas A&M, the following internship objectives were established:

- Gain practical engineering experience in an industrial environment.
- Make technical contributions pertaining to design and analysis in areas of concern to the company.
- Develop an understanding of the organizational approach of solving problems.
- Develop familiarity with the organizational set-up and management methods.

During the internship the above objectives were adequately fulfilled. Technical assignments from different departments were handled which provided an understanding of problems in various sections of the company. Also, it usually led to a useful exchange of ideas and information with different employees and was a source of development for the intern.

INTERNSHIP COMPANY

Packages Limited was established in Lahore in 1957 in collaboration with AB Akerlund and Rausing of Sweden. The objective was to build local skills and competence in paper, printing and packaging industries.

Over its twenty-seven years of existence, the company has grown and expanded progressively. Starting with 500 employees in 1957, the company presently has a workforce of more than 3000. About 250 of these are qualified engineers, planners and specialist technicians. In 1983 alone an expansion program costing more than 390 million rupees was undertaken in the Board Mills and the Packaging divisions. The company has a wide range of processing equipment to convert paper and board into packaging for various industries within Pakistan, and also for some countries in Asia and Africa.

Packages is a highly integrated company. Its business is not merely folding and gluing of paper to form cartons, but it also maintains extensive maintenance and development facilities. A sizable part of its power requirements is met through the company's power house. It has capability to fully develop all types of printing inks to meet its requirements. Also, it has adequate paper and printing related research and quality control facilities.

Packages Limited has several associated companies in Pakistan and abroad. It also assists other developing countries in setting up similar industries and provides services, such as feasibility studies, process and equipment selection, installation, and training services for management, marketing and labor.

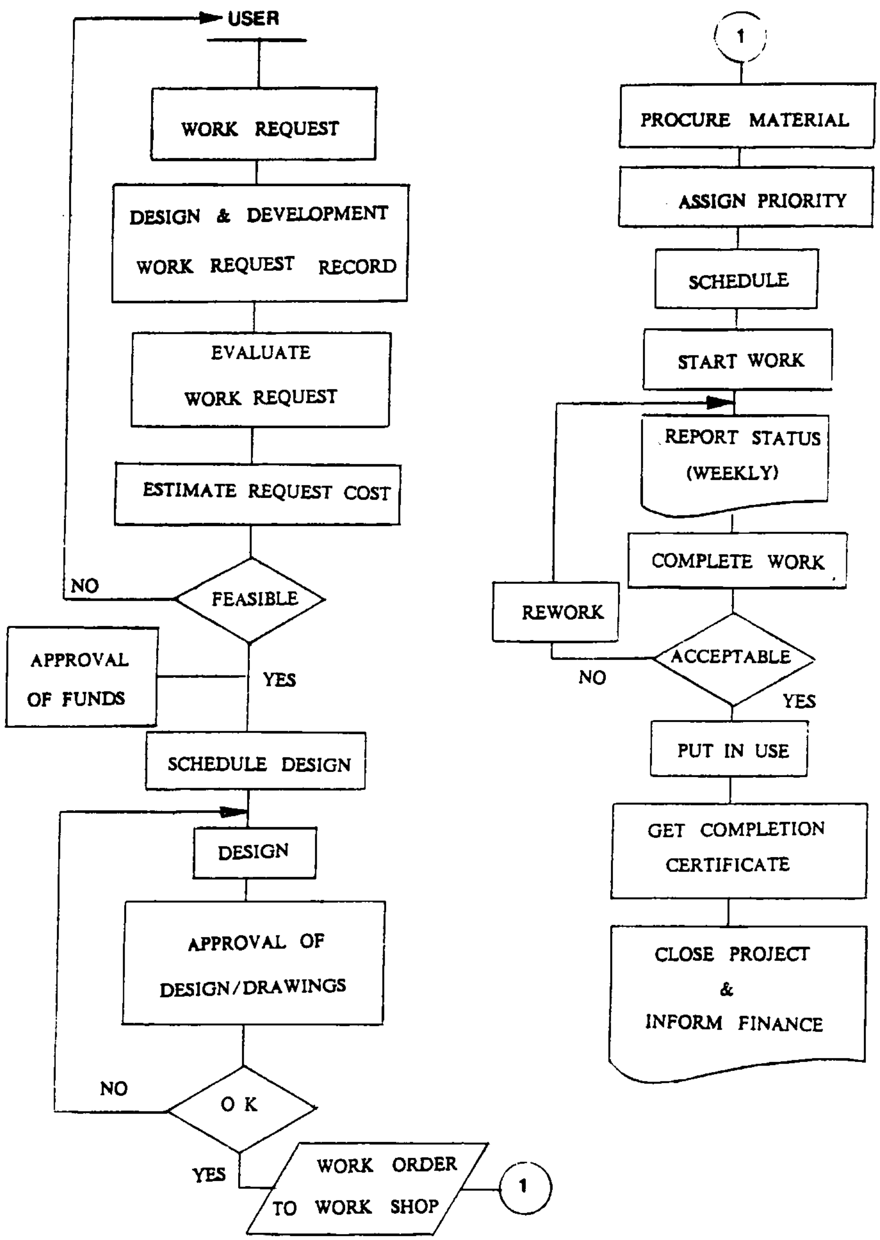
INTERNSHIP POSITION

The intern was employed as Senior Design Engineer in the Design and Development department (D&D) of the Technical Division of Packages Limited. The department provides design and development related support services to all other departments of the company and functions as an in-house consultancy group.

To ensure proper transformation of the design ideas, this department works in close association with the extensive workshop facilities of the company. The usual workload consists of new developments, modifications, cost estimation and maintenance-related services for the production departments.

A flow chart of the work procedure of the D&D department is shown in Figure 1. The general organizational outline is presented in Figure 2, whereas the structure of the Technical Division and location of the internship position is given in Figure 3.

The internship supervisor was Mr. Sheikh Suleman Elahi, who is the Technical Manager at Packages Limited. He has considerable work experience at Packages in different engineering and managerial capacities. He has also served abroad for several years in senior engineering and management positions. Subsequent sections of this report describe different projects and the experience acquired during the internship.



Flowchart of Work Procedure for the Design & Development Department.

Figure 1.

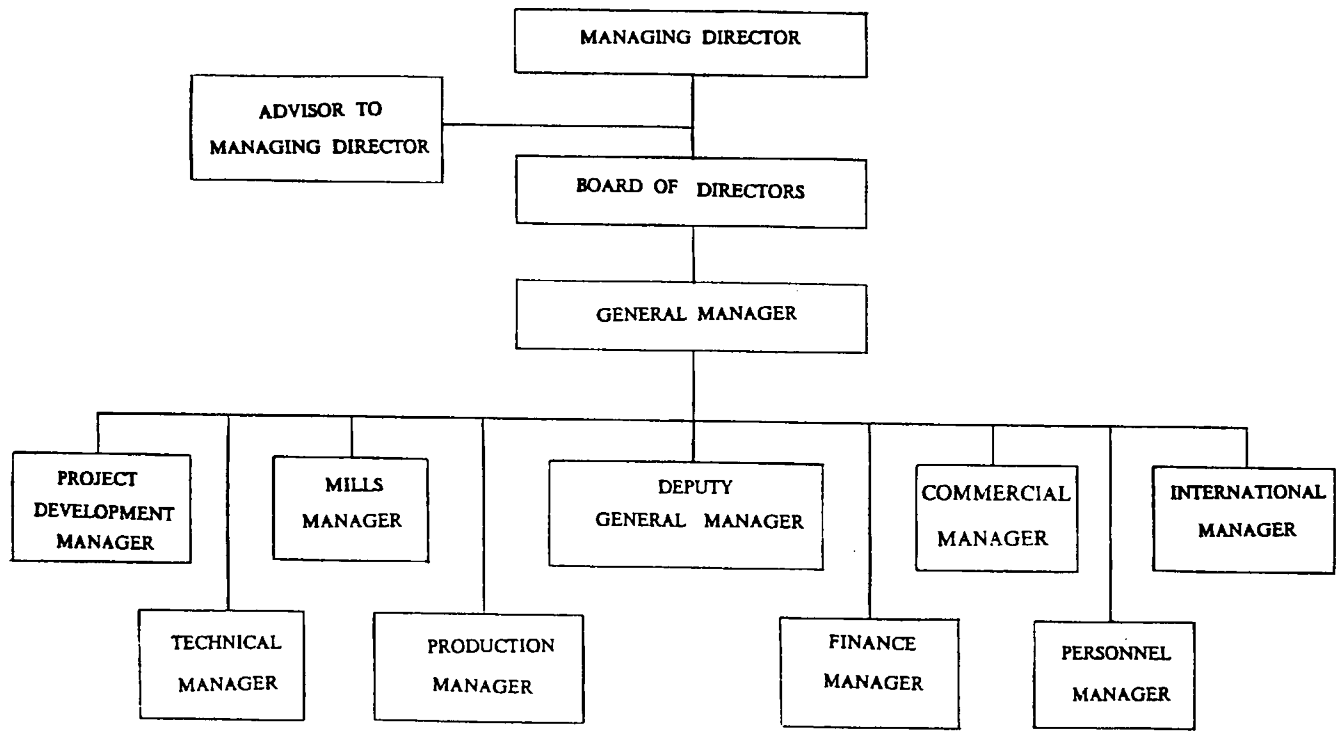


Figure 2. Organizational Outline of Packages Limited.

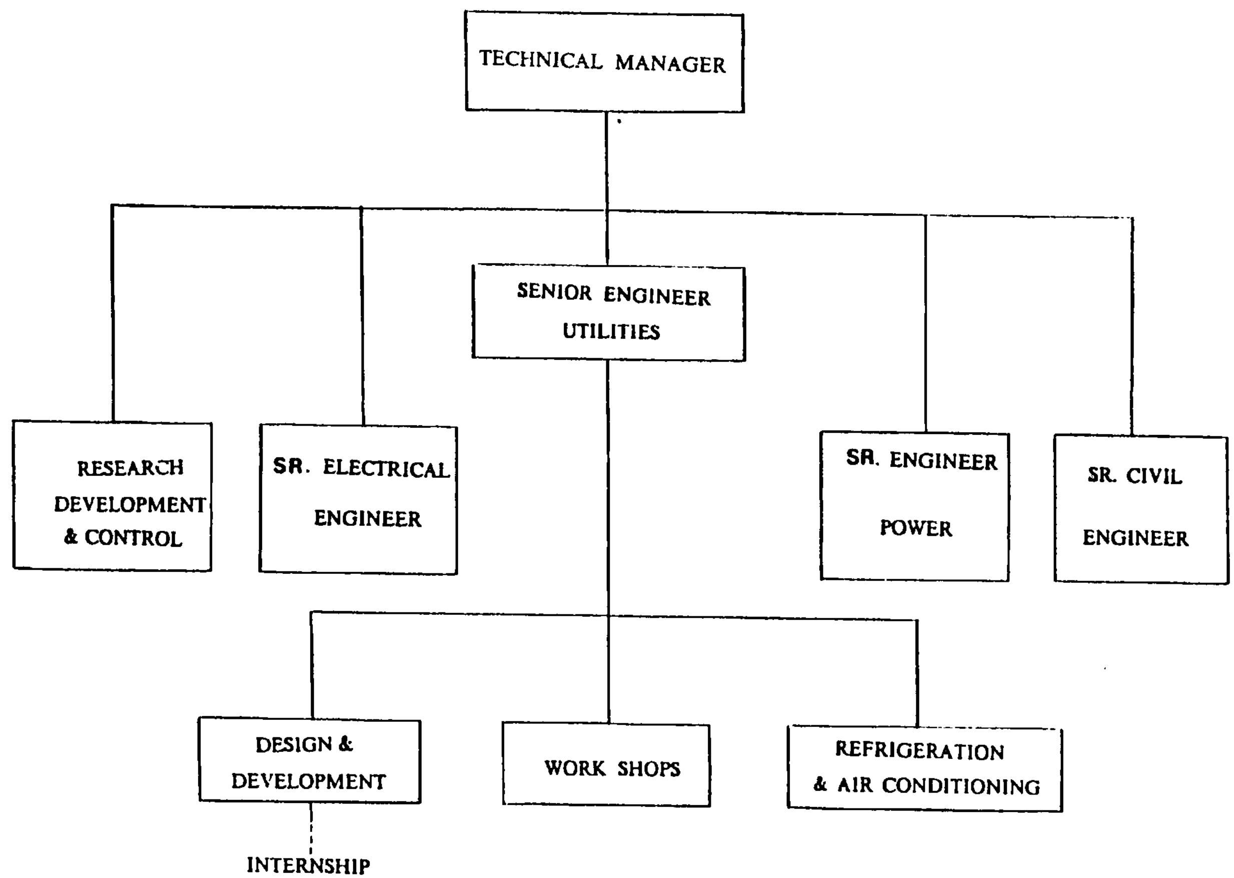


Figure 3. Organization of Technical Division and Location of Internship.

CHAPTER II

ENGINEERING ASSIGNMENTS

The engineering assignments during the internship covered projects relating to new developments, modifications and maintenance. The following assignments will be discussed in detail in this report.

1. Development of a gusset unit for a polyethylene bag machine.

2. Design of truss roofs for a paper storage and cattle shed.

3. Design of a steam jacketed vessel.

4. Design and development of a belt conveyor to handle wheatstraw.

5. Preliminary design of a pneumatic system to convey wheatstraw from storage stacks to the straw preparation plant.

6. Modifications in the pneumatic paper-trim handling system.

DEVELOPMENT OF A GUSSET UNIT FOR A BAG MACHINE

Introduction:

This project was undertaken for the paper converting department of the packaging division. The department, besides flexographic printing, also handles the extrusion of polyethylene tubes of various sizes. These polyethylene tubes are converted to shopping bags after the desired printing.

The bag-making machine basically consists of a reel-unwinding unit and a set of dancing rollers which maintain proper tension in the polyethylene tube. This is followed by a cutting and a thermal sealing die to cut and seal one end of each bag. The need for a gusset-forming unit was realized because without side webs or gussets, the small bag capacity made the bags unsuitable for supermarket shopping.

The earlier arrangement of producing gussets required them to be produced at the extrusion stage. This resulted in proper gussets, but created problems in printing which was the next operation before converting the tube into bags. The problem was poor quality printing, which resulted due to four layers of polyethylene tube towards the reel edges while there were only two of those in the middle portion of the reel. As in flexographic printing the sheet on which impression is to be made is passed between a stereo and a blanket roller, so for uniform print the thickness of the sheet across the roller length should be uniform. This obviously was not the case when the tube with side webs produced during extrusion was used for printing. As much as 25% rejection due to poor printing was not unusual for such jobs.

An obvious solution to this problem lay in producing gussets in the tube during the bag-making stage (after required printing on plain tube) rather than during extrusion. A gusset-making unit was therefore developed which would permit gusset formation on the existing bag-making machine. The engineering assembly drawing for the unit is shown in Figure 4. The unit can be placed on the machine between its existing reel-unwinding unit and the dancing roller set. Thus it would be possible to use it independently as and when required.

Operation Procedure:

To use the gusset unit, the polyethylene tube is passed through two sets of nip rollers at the lower and upper end of the unit frame. Compressed air is then introduced in the tube via a needle to slightly inflate the tube like a balloon. The needle mark is later closed by tape. The tube is squeezed to a wedge shape as it passes through an adjustable guide frame. The guide blades on the sides of the frame are then used to produce webs of the required depth.

After passing through the upper nipping roller set, the tube passes through a set of dancing rollers. These rollers maintain continuous uniform unwinding at the reel end of the machine despite intermittent stop-and-go action at the cutting and sealing end. The stop-and-go action is not desirable at the reel-unwinding end to prevent the tube from running tight or loose at different times. The dancing rollers basically consist of a set of rollers in a frame pivoted at one end with the other end free to move up and down to accommodate a tight or a loose running tube.

Problems During Design & Development:

Some of the problems encountered during the development of this unit were as follows:

First, adequate literature was not available for designing such equipment. A lot, therefore, had to be based on little information or exposure that was available. Some ideas about the rollers for such units were possible by checking the existing bag-making machines. The idea for the suitable frame length, to produce a wrinkle free web of uniform depth, was obtained from the existing gusset unit of the extruder and through discussions with the department supervisors.

The gusset unit of the extruder used a set of nip rollers at one end, and the extruder die at the other end, to maintain proper air pressure in the tube for gusset forming. In the gusset unit for the bag-making machine the nip roller concept of the extruder was extended to both ends of the frame to retain air in the tube. To minimize air leakage, one of the rollers of each of the nipping roller set was rubber lagged. The contact pressure between the rollers was maintained by putting the rubberized roller of each set on springs. A knob was provided to slightly separate the rollers for initial manual feeding of the tube prior to inflation.

Another problem during the development of this unit was the selection of a suitable drive for the nip rollers. The bag machine drive system, though strong enough to pull the polyethylene tube through the rollers, could stretch or break the tube, particularly when thin (low grammage) tubes were used. The constraints on the choice of the drive system were low cost and local availability. This eliminated the ideal variable speed DC drive system with feedback control to automatically adjust the roller speed (unwinding end) according to the machine requirements at the sealing end. However, a PIV (Positive Infinitely Variable) drive unit available to the paper converting department proved useful. This along with a set of dancing rollers on the gusset unit, provided the required speed range adjustment to permit synchronization with the intermittent stop-and-go action at the sealing end. The same drive was used for both the nip rollers by linking them through a chain and sprocket set. The unit was given test runs and except for minor adjustments, it performed satisfactorily.

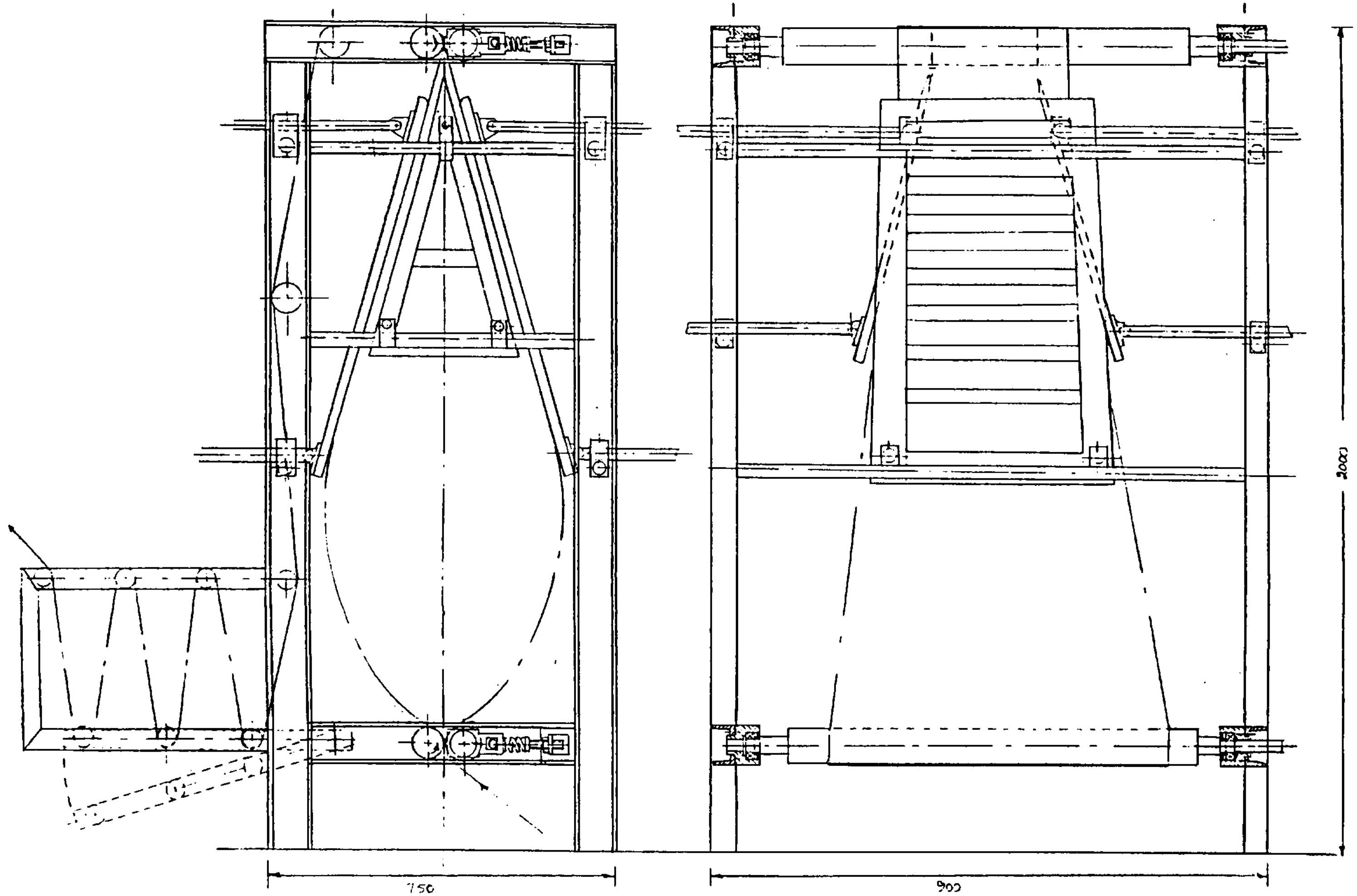


Figure 4. Gusset Unit for Polyethylene Bag Machine.

DESIGN OF TRUSS ROOFS

This involved two projects:

1. Truss roof for the waste paper storage shed.
2. A cattle shed for the Milkpak project.

The work required establishing design parameters, the actual design and cost estimates.

1. Truss Roof for the Waste-Paper Storage Shed:

Introduction:

The shed consisted of covered area of 9600 sq. ft. and housed a waste paper pulping plant and storage area for waste paper. The waste paper is converted into pulp and subsequently recycled into different types of paper and board. Before changes, the shed had a concrete slab roof. The roof had developed several leaking cracks and it was decided to replace it with a light galvanized iron (GI) sheet roof. This roof would not only be functional but also considerably cheaper compared to the concrete slab roof. Besides, the design and fabrication would be possible internally, thereby cutting time and cost for engaging external parties.

Design Procedure:

The plan drawings of the storage shed were obtained from the civil works department. These were used to establish the location of the trusses needed to support the roof without lowering the existing ceiling clearance. The spacing between the purlins was decided on the basis of bending stiffness of the corrugated GI sheets. A 22-gauge sheet would not sag if supported at five feet. To be on the conservative side this was reduced to four feet in actual usage. Sheets were available in 8 × 3 feet and 6 × 3 feet sizes. A suitable sheet overlap (based on the slope of the roof) had to be provided at each joint to avoid the back-flow of water during heavy rain. Usually an overlap of about one foot is sufficient. Having decided the general layout of trusses and purlins, the next step was to estimate design loads, calculate forces and size individual members.

Design Parameters:

Dead loads: These include the weight of the structure itself or those loads that are permanently attached to the structure, such as the weight of trussses, purlins and sheets.

Live loads: These include the loads that are not permanently applied to the structure, such as the wind or snow loads.

The dead load was estimated by considering the total covered area and the number of GI sheets needed to cover it. The GI sheet weight was supported on purlins, which in turn were supported on the trusses. The sizing of the purlins is given in Appendix A. With sheet and purlin load known, the weight of the support trusses was estimated. These weights were added to get the total dead load and hence the dead load per truss.

For our design, the only live load which needed consideration was the wind load. The wind pressure depends on several factors, such as the building height, its location and shape. A wind pressure of 16 lbs. per sq. ft., which corresponds to around 70 miles per hour, was considered satisfactory. Complete listing of design data is given in Table 1.

Table 1. Design Data for the Paper Shed

Dead Load:

- Total covered area : 48 × 200 feet .... 9600 sq. ft.
- GI corrugated sheets required:

  (i) 22–gauge 8×3 : 4×200/2.25 .... 356 Nos.

  Sheet weight (at 30 lbs/sheet) : 30 × 356 .... 5.34 tons.

  (ii) 22–gague 6×3 : .... 356 nos.

  Sheet weight (at 24 lbs/sheet) : 24 × 356 .... 4.26 tons
- Total sheet weight : 5.34 + 4.26 .... 9.6 tons
- Total purlins, each consisting of C 3×4.1×200 ft. .... 14 Nos.
- Total purlin weight (at 4.1 lbs/ft) = 4.1×200×14 .... 5.74 tons
- Total truss weight (at 250 lbs/truss) = 250×18 .... 2.25 tons

∴ Total dead load = 9.6 + 5.74 + 2.25 = 17.59 tons

Live Load:

It is based on wind pressure of 16 lbs per sq. ft., and using a roof slope of 5°. The projected area on which pressure acts amounts to : 400 sq. ft.

- Total wind load = 400 × 16 = 6400 lbs. .... 3.2 tons
- Total Load : 17.59 + 3.2 .... 20.79 tons

∴ Load per truss = 20.79/18 = 1.15 tons.

This load forms the basis of design calculations leading to forces in individual members. Once the forces are known, the members can then be sized to withstand those forces.

Analysis:

The line diagram for the configuration used for the truss is shown in Figure 5a. It also illustrates the manner in which the load is considered acting at the nodes (purlin positions). Before proceeding with the force analysis, the stability of the structural arrangement has to be checked. This is done by using the following equation for plane trusses:

$m + 3 = 2j$: stable statically determinate.

$m + 3 < 2j$: Unstable, nonrigid.

$m + 3 > 2j$: statically indeterminate, redundant.

where:

m = number of members in the truss.

j = number of joints in the truss.

The truss configuration used, as shown in Figure 6, results in : $j = 12$, and $m = 21$. For this the above equation gives:

$$21 + 3 = 2 \times 12$$

$$24 = 24$$

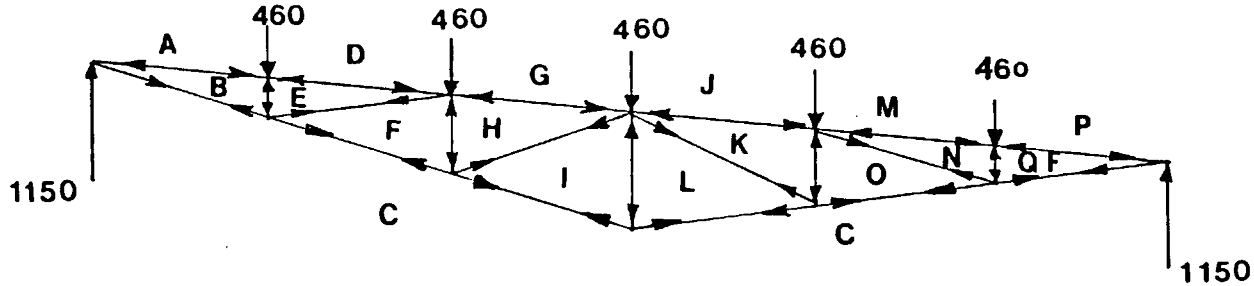
$\therefore$ the chosen arrangement is stable and statically determinate.

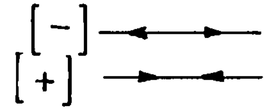
Forces in Truss Members:

As the structure is statically determinate, the reactions and forces in individual members can be determined using basic conditions of equilibrium. First, the reactions are determined using the three equlibrium equations for the whole frame. Then the equations $\Sigma F_x = 0$ and $\Sigma F_y = 0$ can be applied at each joint in turn, when considered as a free body, to calculate forces.

The analysis can also be conducted by graphical means after the initial calculation for the reactions. In this case, a force polygon can be drawn, either separately for each joint or as a composite diagram for the whole frame.

A graphical method using Bow's notation was used for the analysis of forces in this problem. The resulting force polygon is shown in Figure 5b. The forces in individual members of the truss were directly read from this diagram and used to size different elements in the truss framework. The final truss configuration is shown in Figure 6, whereas the material list and cost estimate is given in Table 2.





BAR	F(lbs)
AB	-5400
DE	-5400
GH	-4325
JK	-4450
MN	-5725
PQ	-5725
BC	+5650
FC	+4475
IC	+3400
LC	+3275
OC	+4500
QC	+5750
BE	-475
EF	+1100
FH	-675
HI	+1100
IL	-1375
LK	+1300
KO	-700
ON	+1300
NQ	-500

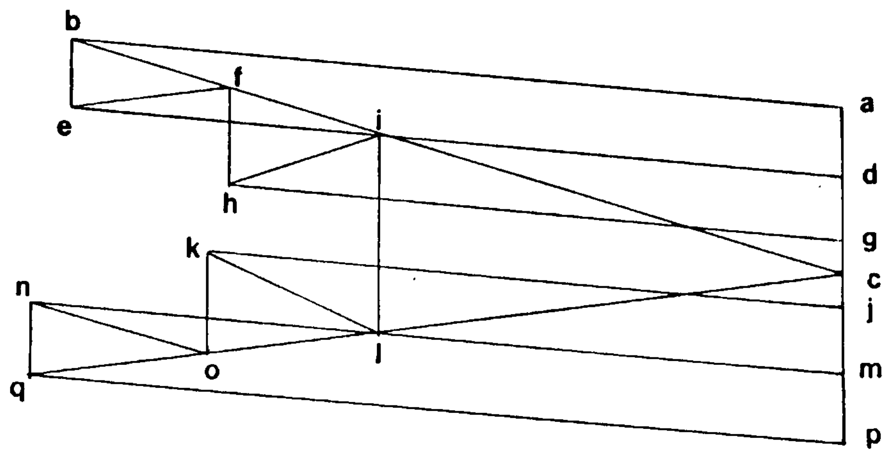


Figure 5. (a) Line Diagram and (b) Force Diagram for the Paper Shed Truss.

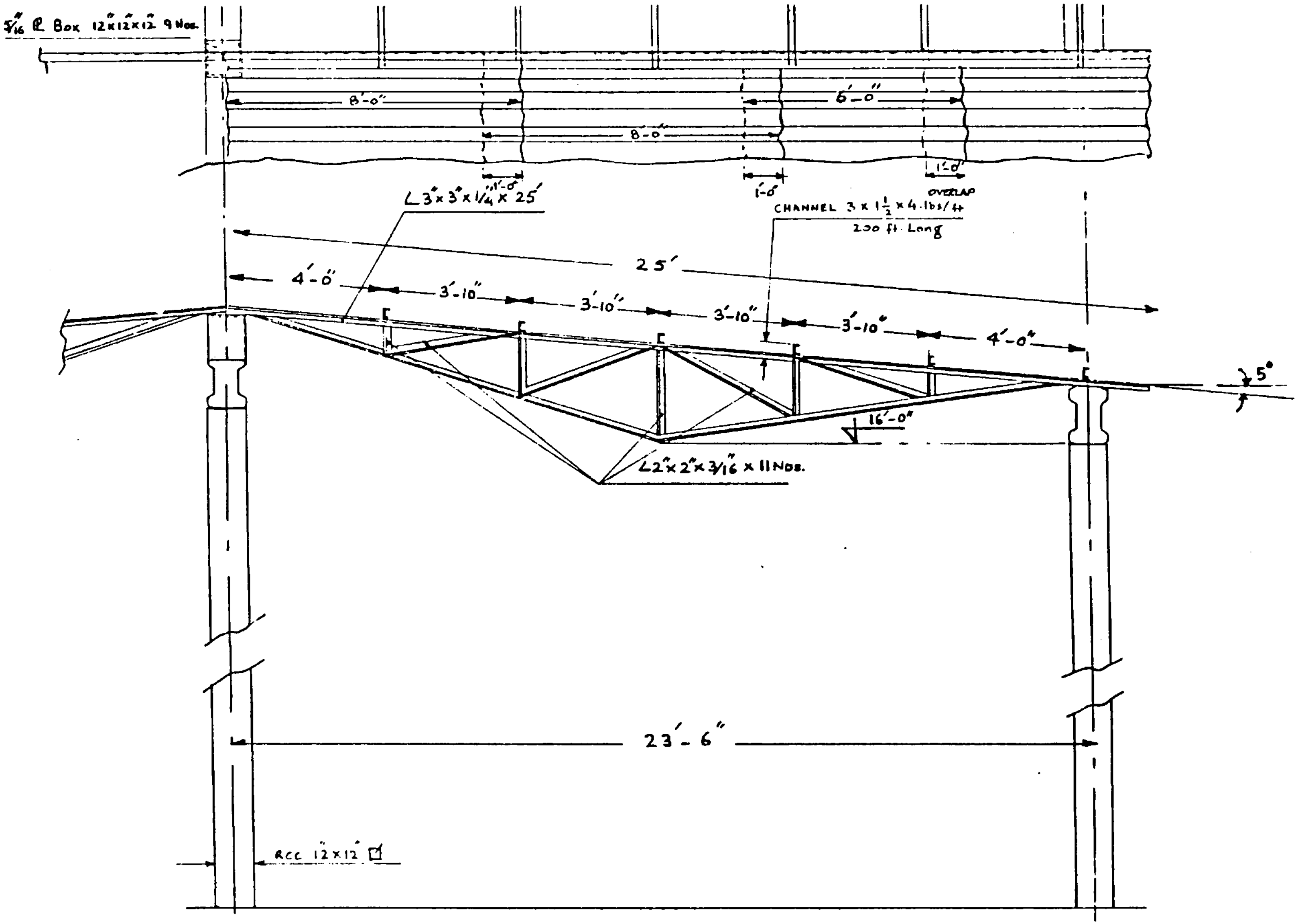


Figure 6. Final Configuration for the Paper Shed Truss.

Table 2. Material List & Cost Estimate for the Paper Shed

TRUSS 25′ LONG 18 NOS.

- L 3 × 3 × 1/4 × 25′ (at 4.9lbs/ft) .... 125 lbs.
- L 2 × 2 × 3/16 × 47′ (at 2.44 lbs/ft) .... 115 lbs.
- Actual truss weight .... 240 lbs
- Total weight of 18 trusses = 240 × 18 .... 2.16 tons
- Truss material cost (at Rs. 5200/ton) .... Rs. 11,232

PURLINS 14 NOS.

- C 3 × 1.5 × 200′ (at 4.1 lbs/ft) .... 820 lbs.
- Total purlin weight (820 × 14) .... 5.74 tons
- Purlin material cost (at Rs. 5200/ton) .... Rs. 29,848

GI CORRUGATED SHEETS

- 22-Gauge 8 × 3 ft .... 350 Nos.
- 22-Gauge 6 × 3 ft .... 350 Nos.
- Sheet cost (at Rs. 96/sheet) .... Rs. 67,500

- Sub-Total .... Rs. 108,580
- U-Clamps, nuts & bolts, welding rods etc. .... Rs. 10,000
- Labor (at 50% material cost) .... Rs. 59,290
- TOTAL .... Rs. 177,870
- Total area .... 9200 sq. ft.
- Cost per sq. ft. .... Rs. 19.33

Deflection of truss:

The unit or dummy load method was used to determine the deflection of a joint for the truss framework shown in Figure 6. The method is based on the principal of virtual work and utilizes the following expression to determine deflection at the desired node.

$$\delta = \Sigma\ [F_1/Q]FL/AE$$

where:

F = Force in each bar due to applied loads

Q = Dummy load, usually taken as unit force, assumed acting at a point where deflection is required.

F_1 = Force in each bar due to Q

l = length of the bar

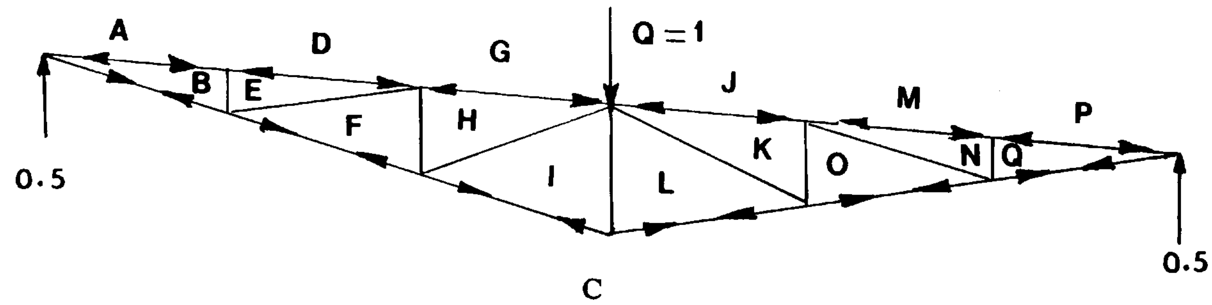
A = Cross-sectional area of the bar

E = Modulus of elasticity

δ = Deflection of joint

As the maximum deflection would be expected towards the center of the truss, the unit load Q is applied at the center joint. The resulting unit load diagram is shown in Figure 7 and the calculations using the above equation for deflection are given in Table 3. The calculated deflection is 0.22 in. which is well below the permissible limit of about 1-in. obtained by the general rule of thumb:

$$\delta = 1/300$$



BAR	F,(lbs)
AB	-2.31
DE	-2.31
GH	-2.31
JK	-2.30
MN	-2.30
PQ	-2.30
BC	+2.40
FC	+2.40
IC	+2.40
LC	+2.30
OC	+2.30
QC	+2.30
BE	0
EF	0
FH	0
HI	0
IL	-1.00
LK	0
KO	0
ON	0
NQ	0

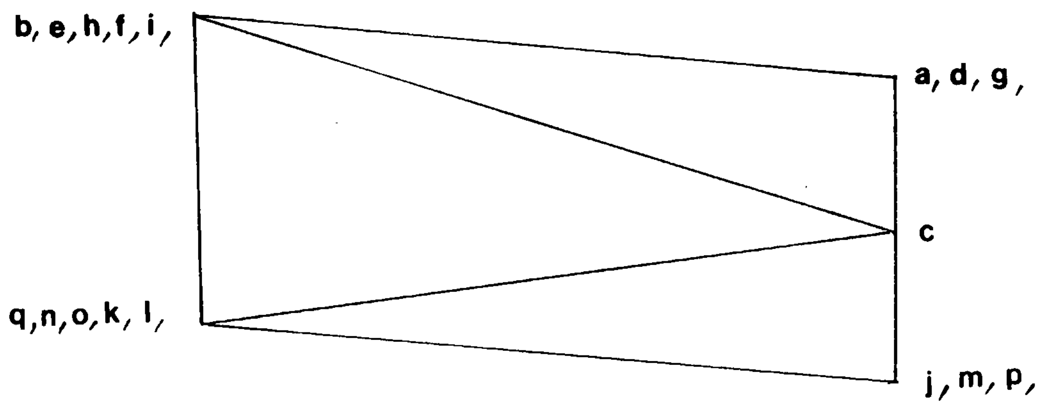


Figure 7. Unit Load Diagram for the Paper Shed Truss.

Table 3. Truss Deflection Calculations

BAR	F(lbs)	F_1(lbs)	L/AE (in/lb)	$[F_1/Q]FL/AE$
AB	−5400	−2.31	1.11×10^{-6}	0.013846
DE	−5400	−2.31	1.11×10^{-6}	0.013846
GH	−4325	−2.31	1.11×10^{-6}	0.011090
JK	−4450	−2.30	1.11×10^{-6}	0.011361
MN	−5725	−2.30	1.11×10^{-6}	0.014616
PQ	−5725	−2.30	1.11×10^{-6}	0.014616
BC	+5650	+2.40	2.22×10^{-6}	0.030103
FC	+4475	+2.40	2.22×10^{-6}	0.023843
IC	+3400	+2.40	2.22×10^{-6}	0.018115
LC	+3275	+2.30	2.22×10^{-6}	0.016722
OC	+4500	+2.30	2.22×10^{-6}	0.022977
QC	+5750	+2.30	2.22×10^{-6}	0.029359
BE	−475	0	nn*	0
EF	+1100	0	nn*	0
FH	−675	0	nn*	0
HI	+1100	0	nn*	0
IL	−1375	−1.00	1.39×10^{-6}	0.001911
LK	+1300	0	nn*	0
KO	−700	0	nn*	0
ON	+1300	0	nn*	0
NQ	−500	0	nn*	0

* Not Needed $\qquad \delta = \Sigma\ [F_1/Q]FL/AE = 0.22$ in.

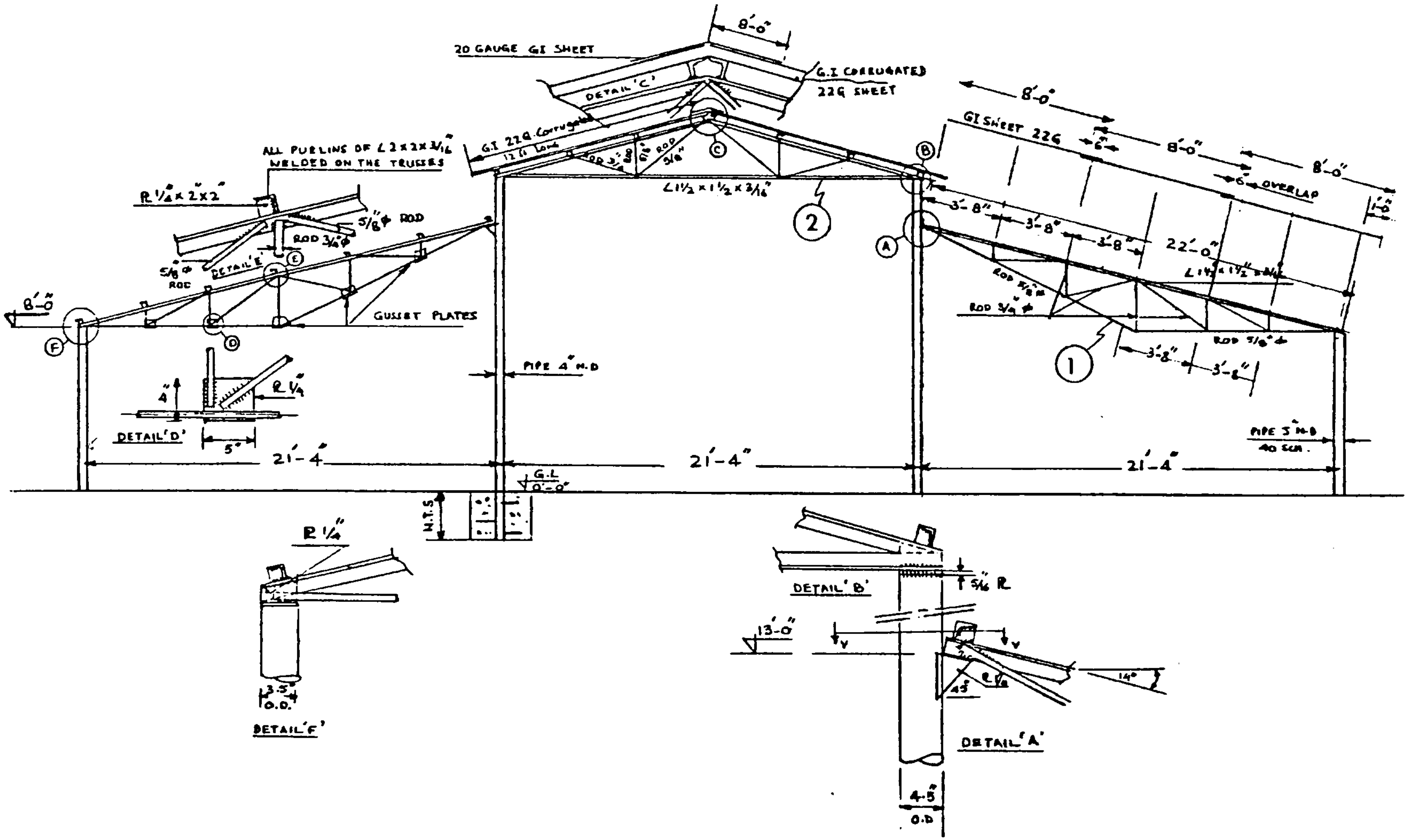
2. Design of a Cattle Shed:

Introduction:

This was basically similar to the paper shed roof project. In fact due to exposure and experience with the previous project, the intuition was more developed to estimate various loads and to select suitable structural steel sections. Thus the design effort proceeded smoothly and better decisions were possible.

As most of the design procedure has already been explained in the earlier paper shed project, any repetitious details will not be given here. The civil department provided the layout plan drawings for the proposed shed. These drawings were used to plan and design a suitable shed structure. Unlike the paper shed, where only waste paper was to be stored, this shed was for keeping a special breed of cattle. So proper ventilation to keep the temperature within limits, particularly during the excessive summer heat, also had to be considered in the design.

The shed was kept open on the sides so that adequate cross-flow of air would be possible to prevent foul air build-up. However some foul air, being lighter, manages to get trapped in the ceiling unless proper draft ventilation exists there. During night when the atmosphere gets heavy, this air can actually settle down and may harm the cattle. To prevent this foul air build-up, cross ventilation was provided near the ceiling top. During summer, the GI sheet roof would not provide proper protection from the heat. However, by maintaining adequate water spray over the roof, it should be possible to cool the shed to the desired level. Keeping these factors in view, the design shown in Figure 8 was proposed. The relevant design data is presented in Table 4.



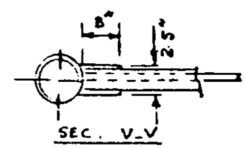


Figure 8. Front Profile of Cattle Shed.

Table 4. Design Load for Truss-1 and Truss-2 of Cattle Shed

Two types of trusses are used in the design: truss-1 and truss-2 as shown in Figure 8. Design load on each is estimated as follows:

Dead Load Truss-1:

- Sheet load per truss-1 = $\frac{\text{(total sheets)} \times \text{(wt. per sheet)}}{\text{No. of supporting trusses}}$
  = (126 sheets) × (30 lbs/sheet) / 8 = 473 lbs
- Purlin load per truss-1 = (lbs/ft.)×(ft.)×(purlins/truss-1)
  = 2.5 × 13 × 7 .... 228 lbs.
- Estimated weight of each truss-1 .... 150 lbs.
- Total dead load per truss-1 .... 850 lbs

Live Load:

- Live load/truss-1 = (design wind Press.)×(area resisting wind)
  = 16 lbs/sq. ft. × 5 × 13 .... 1040 lbs

∴ Total design load per truss-1 .... 1890 lbs

Total Load on Truss-2:

The calculations for this are similar to the above and work out to the following values:

- Total dead load/truss = sheet load/truss + purlin load/truss + Truss wt.
  = 484 + 260 + 150 = ... 894 lbs.

- Live load per truss-2 = 16 lbs/sq.ft. × (3 × 13) = ... 624 lbs

∴ Total design load per truss-2 .... 1518 lbs.

Forces in Truss Members:

The estimated design loads given in Table 4 were used to find the forces in individual truss members. These forces were obtained by applying the basic conditions of static equilibrium at each node. Once the forces were established, standard structural steel sections that would effectively withstand these forces were chosen to form different truss members. For the members in tension, the selection was simple--the resisting area of the member had to be such that the stress due to tension would not exceed the yield strength (with a factor of safety) of steel. Rod is usually used as a good tension member. The compression members, however, require more careful selection of different structural sections. Every compression member was treated as a strut or a column (depending on the selenderness ratio) and relevant column formulas were used to estimate their load bearing capacity, that would not cause compression overload or buckling failure.

The design calculations are similar to those for the truss roof for the paper shed. Figures 9 and 10 show the force diagrams and forces in individual members of truss-1 and truss-2. These forces were used to size different members in the truss framework shown in Figure 8.

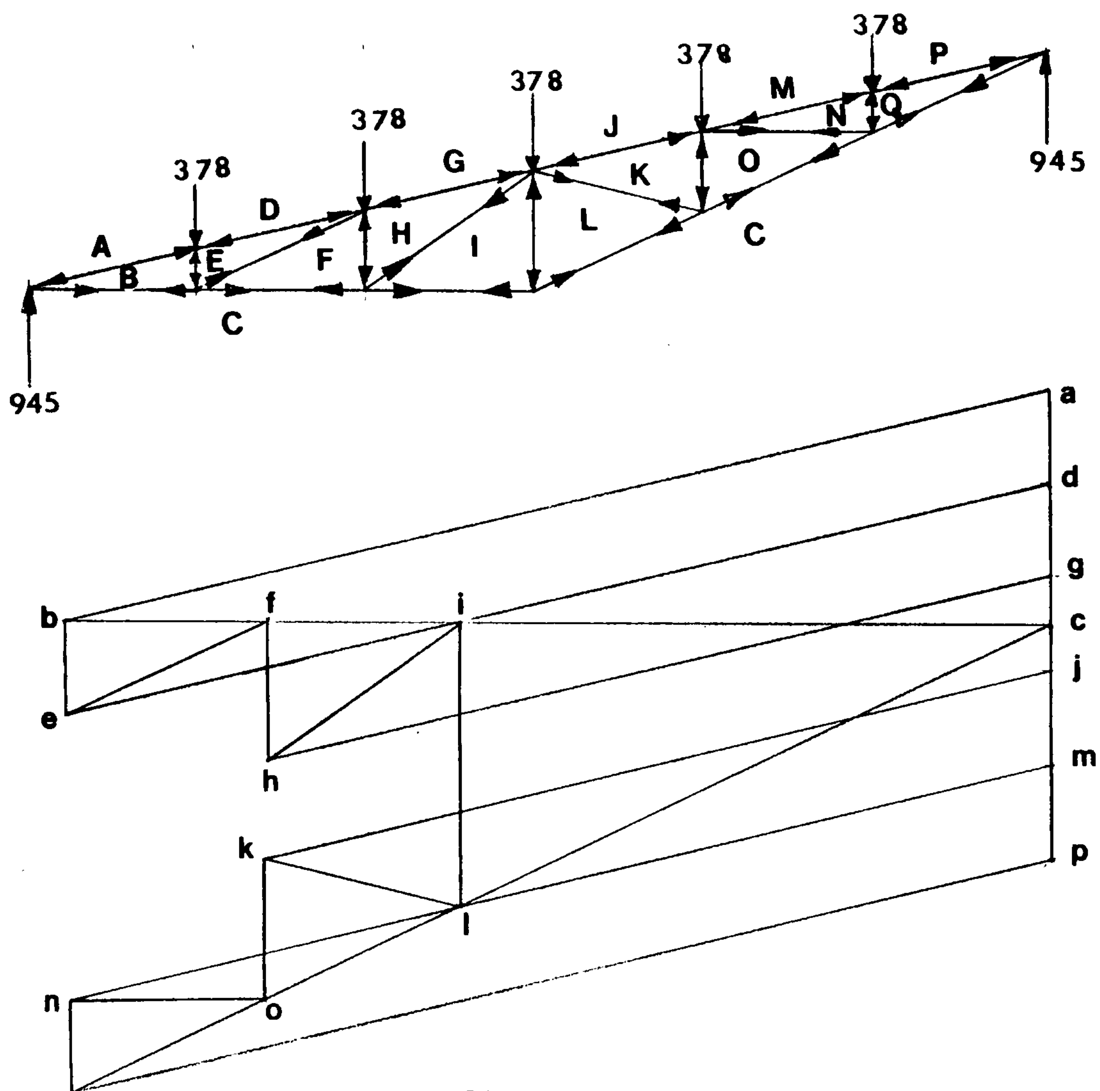


Figure 9. Force Diagram for Truss-1.

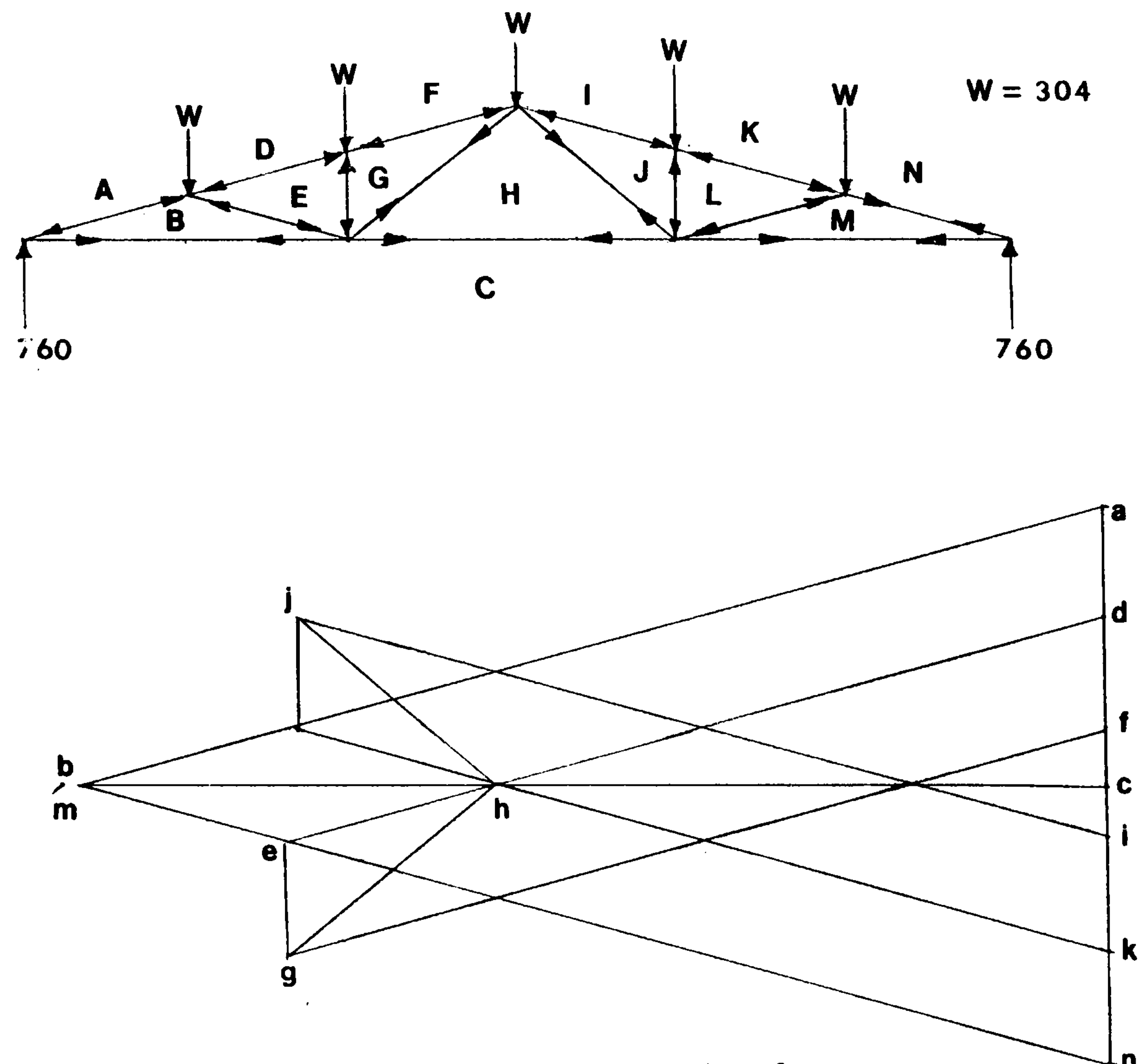


Figure 10. Force Diagram for Truss-2.

DESIGN OF A STEAM JACKETED VESSEL

A steam jacket shell for a mixing tank was designed using the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code. The physical requirements for the vessel, based on the solution batch size, were specified by the user department. Steam was available at 4 atmospheres. The design drawing is given in Figure 11.

When steam pressure is applied it results in a bursting force on the outer jacket shell, while the inner shell experiences a collapsing pressure. Collapse can occur in a variety of modes depending on the relative position of the reinforcing rings and the vessel size. Thus the design of the outer and the inner shell requires separate treatment. The design procedure and the calculation details are given below.

Known information:

- Operating pressure is 4 atmp. steam (60 psig)
- The solution to be used in the tank is slightly corrosive, so stainless steel should be used for the inner shell.
- The vessel size is specified by the user department.

To design a vessel the plate thickness from which the shell is to be made, the size of the stiffeners (if required), and the vessel ends need to be specified. The ASME pressure vessel code is an extensive source of information for pressure vessel designers and fabricators. It contains a complete range of up to date information relating to design calculations, materials, welding and fabrication specifications, inspection and testing procedures, maintenance and other major or minor details that are of use to the designer or the fabricator. The main resource for this work was section–VIII, division I of this code. As it is not possible to refer to the entire material in this report, only relevant parts necessary to understand the calculations are given in Appendix B.

Key Definitions:

Design Pressure: The code defines it as the maximum difference in pressurre between the inside and outside of a vessel or between any two chambers of a combination unit, based on the most severe conditions of coincident temperature and pressure expected in normal operation. For our conditions this is 60 psig.

Design Temperature: This is defined as the maximum mean temperature expected through the thickness. In the present design, the maximum expected temperature can be the temperature of the fluid which is steam at 4 atmospheres. At this pressure the saturation temperature is 350 °F. For design consideration we can expect maximum temperature of up to 500 - 600 °F depending on the dryness of steam.

Maximum Allowable Stress: The maximum permissible values for different materials are given in sub - section C of the code. Relevant sections of table UCS–23 for carbon and low alloy steels are given in Appendix B. Calculations are based on low carbon plate steel SA 283 Grade D and SA 410 stainless steel.

Corrosion Allowance: Vessels subject to thinning by corrosion or erosion should have provision made by a suitable increase in thickness of the material over that determined by the design formula. For only nominal corrosion such an increase may not be made.

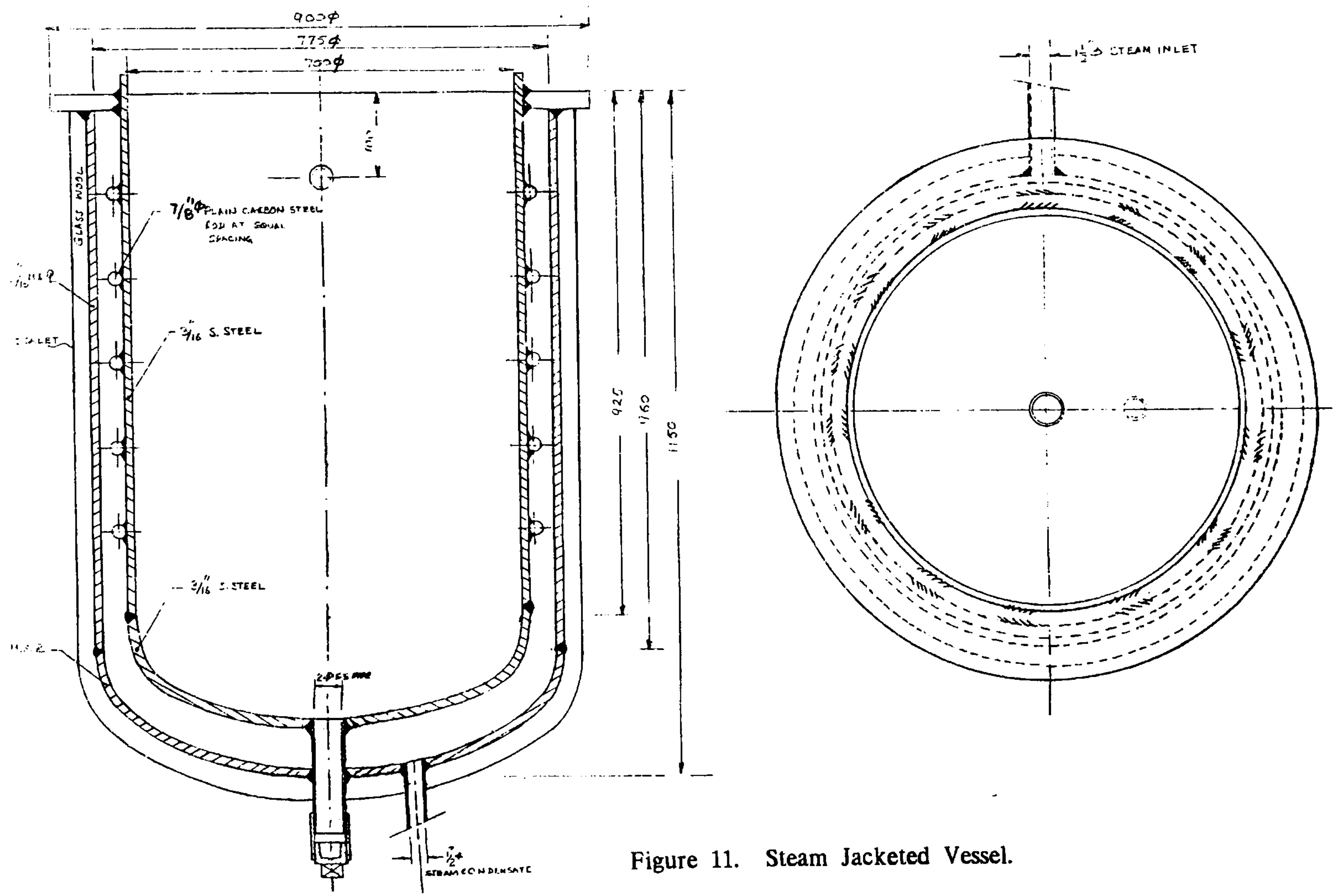


Figure 11. Steam Jacketed Vessel.

Thickness Calculations for the Outer Shell:

For thin walled pressure vessel treatment to be valid: $t/D \leq 1/10$. In this problem D = 30.7 in., so even if t = 1 in. (not known yet), $t/D \leq 1/10$, and we can conveniently expect that thin wall treatment would be valid. For cylindrical shells under internal pressure the relevant section of the code, UG-27, requires the thickness to be calculated by the following formula when $t < 0.5R$ or $P < 0.385SE$:

$$t = P\,R \,/\, (S\,E - 0.6\,P) \tag{1}$$

where:

t = minimum thickness

P = allowable pressure, psi

S = allowable stress, psi

E = joint efficiency

R = inside radius, inches.

Using: $P = 60$ psi, $R = 30.5/2 = 15.25$

$S = 12{,}650$ psi, $E = 0.70$

(The above values for S and E are based on the code requirements for low carbon steel and non-radiographed butt welded joints.)

Equation (1) gives : $t = 0.11$ inch.

According to section UCS-25 of the code, vessels with a required thickness of less than 1/4 in. should be provided corrosion allowance not less than 1/6 of the calculated plate thickness. Accordingly, a suitable thickness for the shell is 3/16 inch.

$\therefore$ Use 3/16 in. M.S. plate for outer shell.

Calculations for the Inner Shell:

This shell is subjected to external pressure and the rules for designing such vessels are covered in section UG-28 of the pressure vessel code. The design procedure basically requires to start with assumed values for thickness and distance between stiffening rings. These are used to calculate ratios L/D_o and D_o/t which are then used with graphs UCS-28.1 and 28.2 (Appendix B) to calculate factors B and A. The allowable working pressure is then determined by equation:

$$P_a = B/[D_o/t] \tag{2}$$

If calculated pressure P_a, is greater than the design pressure P, the assumed thickness is satisfactory. $P_a < P$ indicates the assumed thickness is not sufficient. The process then has to be repeated with increased thickness.

For our design situation:

D_i = 27.5 in.

P = 60 psi

Assume t = 3/16 in. = 0.1875 in.

and L, the greatest center to center distance between stiffening rings = 5 in.

D_o, the outside diameter of shell = 27.875 in.

Therefore:

L/D_o = 5/27.875 = 0.18

D_o/t = 27.875/.1875 = 148.67

Using these values on chart UCS-28.2 for stainless steels, as shown in Appendix B, the factor B is read as:

$$B = 12000$$

Using Equation (2) :

$$P_a = 12000/148.67 = 81 \text{ psi}$$

Thus allowable pressure P_a = 81 psi, is greater than the design pressure P = 60 psi. Therefore the assumed thickness, t = 3/16 in. is correct.

Use t = 3/16 in. stainless steel plate for the inner shell.

Determination of Stiffening Ring Size:

For this section UG-29 of the code is applicable. The moment of inertia I_s of the chosen ring should not be less than that given by the formula:

$$I_s = D_o^2 \, L_s \, (t + A_s/L_s) \, A \, / \, 14 \qquad (3)$$

Where:

I_s = The required moment of inertia of the stiffening ring about its neutral axis. - in.^{4}

A_s = Cross sectional area of the stiffening ring. - in^2

A = Factor determined from appropriate chart - (UCS 28.1, Appendix B)

L_s = Maximum spacing of the stiffening rings - in.

Assuming the stiffening ring to be 7/8 inch diameter plain carbon steel rod. Then:

$$A_s = \pi(d)^2/4 = 0.60 \text{ in}^2$$

$$I = \pi(d)^4/64 = 0.03 \text{ in}^4$$

Factor A, as read from chart UCS-28.1 in Appendix B, is: A = 0.00035

∴ The required moment of inertia I_s from Equation 3 is: I_s = 0.029 in^4

As the moment of inertia I for a 7/8 in. rod (= 0.03 in^4) is greater than the required I_s for the ring, therefore the assumed size of the ring is satisfactory.

Attachment of Stiffening Ring to the Shell:

This is covered by section UG-30 of the code. Rings may be placed inside or outside the vessel and can be welded, brazed or riveted. For the design under consideration here, the rings were placed outside and intermittently welded on each side of the ring with a maximum spacing of $8 \times t$ between the welds as required by the code. (Fig. UG-30, Appendix B).

Calculations for the Formed Heads:

Various configurations for vessel heads are possible. The simplest one being a flat head, but it usually requires thicker plates particularly if the diameter is large. The dished head forms, like ellipsoidal or hemispherical heads, though requiring thinner plates are expensive to fabricate. Ellipsoidal head shapes were chosen in this problem. The calculations based on sections UG-32 and 33 of the code are given below:

For the **outer or the jacket shell** the pressure is on the concave side (internal pressure). The required thickness is calculated using the equation:

$$t = P\ D\ /\ (2\ S\ E - 0.2\ P) \qquad (4)$$

where the symbols have usual meaning. The equation is to be used when the inside depth of the head (minor axis) equals one-fourth of inside diameter of the head skirt. (i.e. $h = 1/4 \times D$).

Thus for the given conditions:

$P = 60$ psi

$D = 30.5$ in.

$E = 0.70$ (joints not radiographed)

$S = 12{,}650$ psi (for low carbon steel)

Using the above values in Equation 4, $t = 0.10$ inch.

$\therefore$ Use 1/8 in. M.S. plate for the jacket head.

Inner Shell Head (convex head):

In this case the ellipsoidal head is subjected to external pressure. The code requirement for its thickness calculations is the greater of the following:

1. The thickness is computed as for internal pressure but using a design pressure of 1.67 times the external pressure, assuming a joint efficiency $E = 1.00$ for all cases.

2. The thickness computed by using chart UCS-28.2 in Appendix B and the formula:

$$P_a = B / (L_1 / t_h) \tag{5}$$

Calculations:

(1). In this case the following values are used to calulate thickness from Equation 4:

$P = 1.67 \times 60 = 100$ psi

$D = 27.5$ inch, $S = 15{,}000$ psi, and $E = 1.00$

Equation 4 gives thickness, $t = 0.093$ inch.

(2). This requires establishing the following ratios first:

$$L_1/100\ t_h\ , \qquad L_1/t_h\ , \qquad D/2h$$

where: $L_1 = K_1\ D$, and t_h = thickness of the head.

The factor $D/2h$ is the ratio of the major to minor axis of the ellipse from which ellipsoidal head is assumed to be generated (surface of revolution). Table UG-37, Appendix B, is used to determine K_1

For $D/2h = 2$, $K_1 = 0.90$. This gives: $L_1 = 0.90 \times 27.5 = 24.75$

Assuming t_h = 3/16 inch we get: $L_1/100 = 1.33$, and $L_1/t_h = 133$

From chart UCS-28.2 in Appendix B : Factor B = 90000

Therefore from Equation 5 : $P_a = 9000/133 = 67.7$ psi

$\therefore$ P_a, allowable pressure (67.7 psi) > P, design pressure (60 psi)

$\therefore$ Use 3/16 inch stainless steel plate for the inner shell head.

Jacket Closure:

This refers to the manner in which the jacket is attached to the vessel shell. Section UA-104 of the code gives the rules for its design. The type of closure bar used is illustrated in figure UA-104 (f-1) and (g-3) in Appendix B. The formula used for determining the bar thickness is given below:

$$t_{rc} = 1.732 \sqrt{PRJ/S} \tag{6}$$

where :

P = design pressure

R = radius of shell

J = jacket width

S = allowable stress

For our design conditions, the following value is obtained for t_{rc} :

$$t_{rc} = 1.732 \sqrt{60 \times 13.75 \times 1.50/12{,}650} = 0.53 \text{ inch}$$

$\therefore$ Use 1/2 inch plate

The welding details are according to the specifications given in figure UA-104 (f-3) and (g-3) in Appendix B.

BELT CONVEYOR FOR HANDLING WHEATSTRAW

A belt conveyor was designed for the cooking house of the paper and board mills division. The objective was to mechanically feed wheatstraw from the mixing chamber, where dry wheatstraw was mixed with the desired chemicals, to the digestors. The digestors were used to steam-cook straw to separate pulp fiber.

The position of the digestors, the mixing chambers and the belt conveyors is shown in Figure 12. The system had to be flexible to feed any particular digestor from either of the two mixing chambers. The design of the main conveyor will be described in detail in the following pages.

Belt Conveyor Components:

The belt conveyor consists of the following components:

(i) The belt

(ii) The carrying idlers

(iii) The return idlers

(iv) The head pulley or the head end

(v) The tail pulley or the tail end

(vi) Drive unit

(vii) Belt tightening unit

(viii) Feed and discharge equipment

(ix) Support structure or frame

The engineering drawing of the main belt conveyor designed for the cooking house is given in Figure 13. The design data and the calculation procedure used to size different members is as follows:

Design Data:

- Specfic weight of wheatstraw (from mixing chamber), W $\simeq$ 20 lbs/ft^3
- Belt conveyor is horizontal and has trough rollers.
- Capacity to be handled per hour, C $\simeq$ 25 tons / hr.
- Center – to – center length of conveyor, L = 85 ft.
- Belt width: A one meter wide belt is used. Effective width, b = 36 in.

Once the above basic design parameters are established, the remaining information for the system can be determined from standard belt conveyor equipment catalogues or the materials handling handbooks.

1. Idler spacing and size:

Using Appendix C1, the idler spacing when a 36 inch belt is used to convey material weighing up to 30 lbs per cubic ft. is as follows:

- Carrying idler spacing: 4.5 ft.
- Return idler spacing: 10 ft.
- Idler diameter: 4.5 in.

2. Belt speed:

Refering to Appendix C2, the following information is obtained:

Conveying capacity for a 36-in belt at 100 ft/min: 235×0.20 = 47 tons/hr.

∴ For 25 tons / hr capacity, the belt speed is: (100/47)×25 = 53.2 fpm.

∴ Design belt speed = 55 fpm

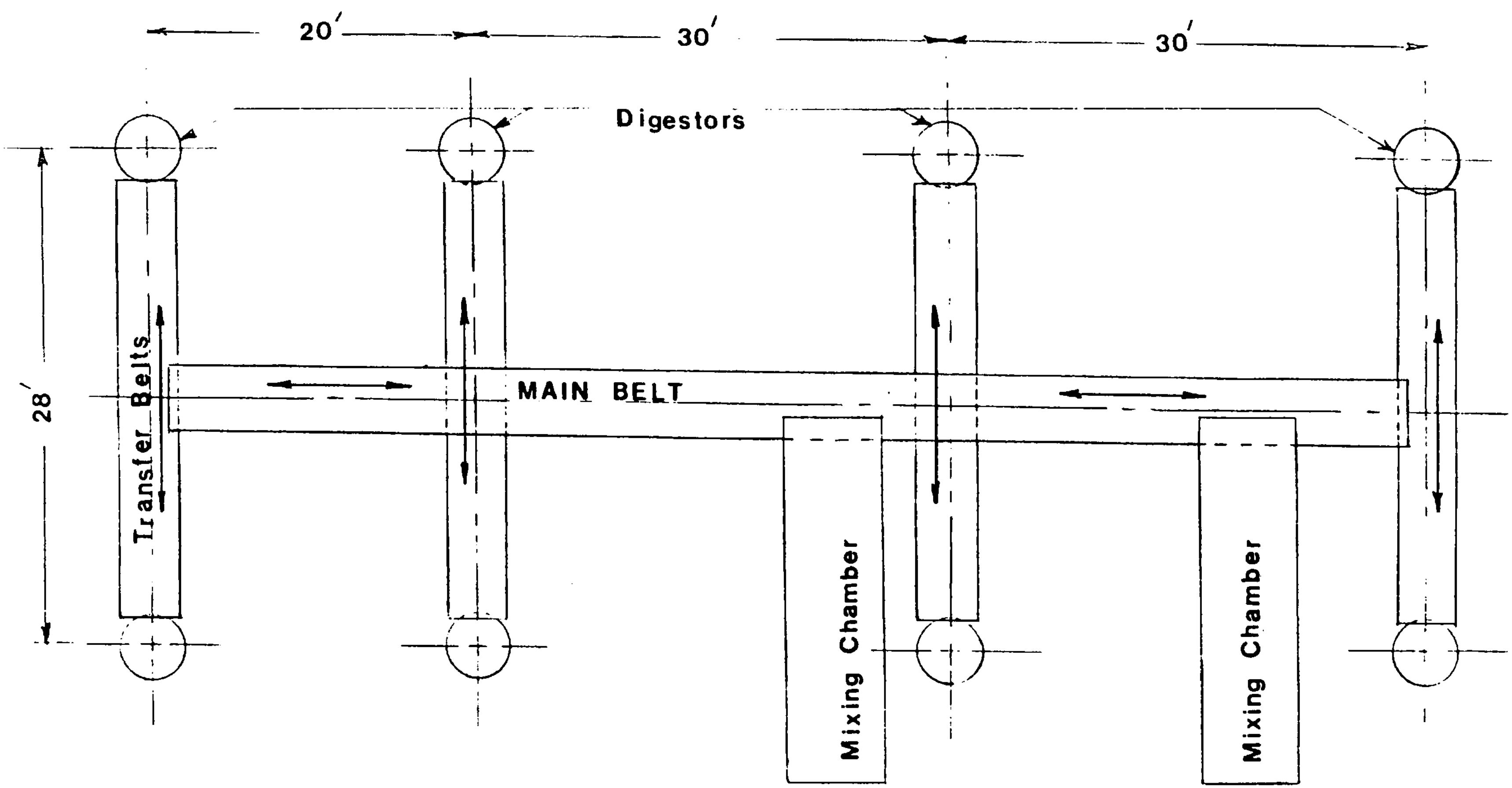


Figure 12. A Layout Showing Position of Belt Conveyors and Digestors.

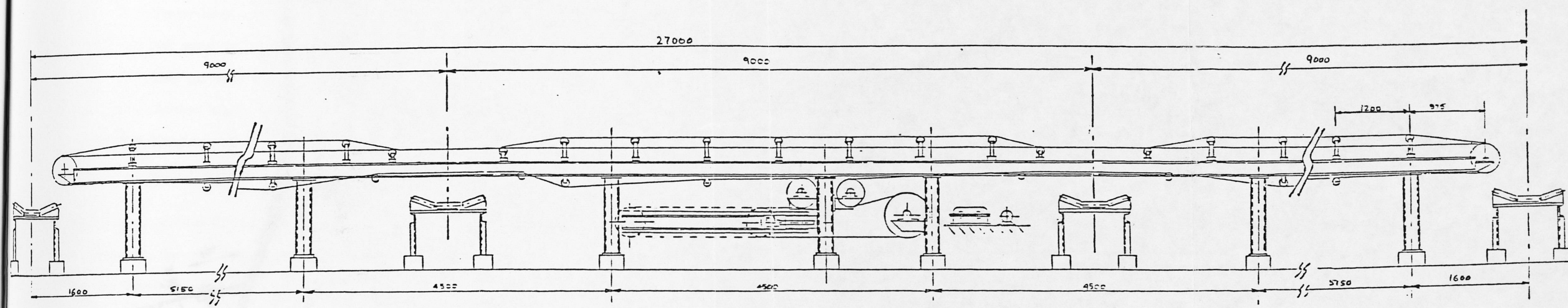


Figure 13. Engineering Drawing of the Main Belt Conveyor.

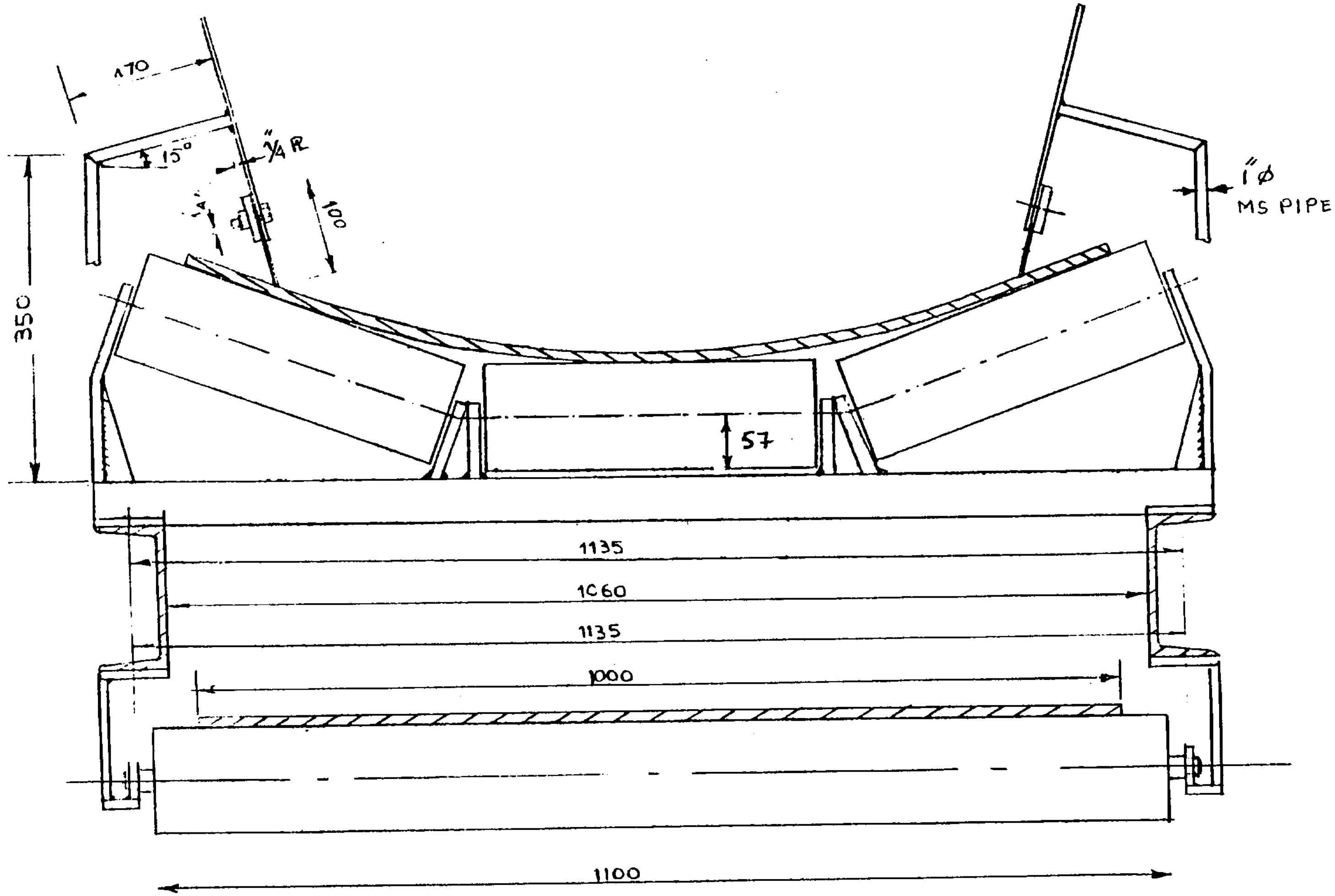


Figure 13b. Detail View of Troughing Idlers. (20° Trough, Equal Rolls)

3. Horsepower determination:

A belt horsepower formula developed by Goodyear Rubber and Tire Company relates the total power requirements as follows:

Total HP = [HP to move empty belt] + [HP to move load horizontally]

+ [HP to elevate load (added) or lower load (deducted)]

$$= \frac{F(L + L_0)(0.03QS)}{990} + \frac{F(L + L_0)T}{990} + \frac{T\ H}{990}$$

where:

F = Friction factor

L_0 = Length factor

L = Center-to-center distance, in ft., between the head and the foot pulley.

S = Belt speed in ft. per min.

T = Tons per hour of material handled.

H = Vertical height in ft. to which the material is raised or lowered.

Q = Weight of moving parts of conveyor in pounds per foot of center-to -center distance, carrying and return runs.

Tables in Appendices C3 and C4 are based on the above formula. They can be used to determine the belt horsepower requirements directly. For normal conveyor in average industrial applications, the belt sag is limited to around 2 % of the idler spacing. This corresponds to:

Fricton factor, F = 0.030

Length factor, L = 150 ft

For a conservative estimate, using Appendix C3, for a 42-inch belt :

HP (for an empty horizontal belt at 100 fpm) = 1.15

HP (at 55 fpm) = (1.15/100) × 55 = 0.64

Horsepower requirement to move the material, as given in Appendix C4 for 100 feet center to center distance is:

$$HP \text{ (for handling capacity of 100 tons/hr.)} = 0.76$$

$$HP \text{ (for handling capacity of 25 tons/hr.)} = 0.76 \times 25/100 = 0.19$$

As there is no vertical movement, the total horsepower requirement is:

$$\text{Total horsepower} = 0.64 + 0.19 = 0.83$$

Considering the overall efficiency of transmission system to be around 60 % :

$$\text{Total horsepower requirement} = 0.83/0.60 = 1.38$$

A 3 HP, 1500 rpm electric motor with a speed reduction gear box was used to drive the belt.

4. Effective horsepower pull:

It is given by the formula:

$$E = HP \times 33{,}000 \; / \; S$$

$$= 3 \times 33{,}000/55 = 1800 \text{ lbs.}$$

5. Preliminary determination of operating belt tension:

Assuming a LPS 220° (rubber lagged pulley snubbed at 220° of the belt contact), Appendix C5 gives:

$$T_1 = 1.35 \times E = 1.35 \times 1800 = 2430 \text{ lbs.}$$

6. Slack side tension at the drive pulley:

$$T_2 = T_1 - E$$

$$= 2430 - 1800 = 630 \text{ lbs.}$$

7. Tentative belt selection:

Referring to Appendix C6, for handling light materials like wheatstraw, a 36 inch belt requires:

A minimum of 4-ply of 28-oz duck. Eight ply is the maximum.

8. Check for maximum belt stress:

If 4-ply belt is used:

$$\text{Unit belt stress} = 2430/4\times36 = 16.87 \text{ lbs/in./ply}$$

The permissible belt stress (lbs/in./ply) as given in Appendix C7, is as follows:

28 oz duck : 35 lbs., Using vulcanized splice

: 26 lbs., Using metal splice

Thus a 4-ply belt has sufficient strength. Recommended cover thicknesses as given in Appendix C2 are :

Top rubber cover : 1/8 in.

Bottom rubber cover : 1/16 in.

9. Unit weight for belt:

From Appendix C8, for a 28 oz duck, 4-ply belt with total rubber thickness of 3/16 in. on both sides, The total belt weight per ft. is given as:

$$\text{Weight / in. width / ft. length} = 0.201 \text{ lbs}$$

$$\text{Weight / ft of belt} = 0.201 \times 36 = 7.24 \text{ lbs}$$

10. Load of raw material per foot of belt:

$$= \text{Capacity in lbs per min. / Belt speed in fpm.}$$

$$= [25 \times 2000 / 60] / 55 = 15.15 \text{ lbs.}$$

11. Minimum belt tension (sag tension) at the front end:

To limit the belt sag between the troughing idlers to around 2 % of the idler spacing (an accepted practice), the tension in the carrying run of the conveyor should not be less than:

$$\text{Minimum belt tension} = 6.25 \text{ w l}$$

where:

w = weight per ft. in lbs. of the belt and its load.

l = idler spacing in ft.

∴ Minimum sag tension = $6.25\times(15.15 + 7.24)\times4.5 = 630$ lbs.

As there is no inclined portion in the belt, this will be the total fixed tension.

12. Total operating tension in the belt:

This is the sum of the fixed tension and the effective horsepower pull:

i.e. $630 + 1800 = 2430$ lbs.

∴ Unit belt stress = $2430/4\times36 = 16.88$ lbs./in./ply

This is well below the permissible value of 35 lbs. as given in Appendix C7.

Ratio of operating stress to the maximum permissible = $16.88/35 = 0.48$

As the operating stress level is only 48% of the maximum permissible, the conveyor is classed as "normal" and the starting stresses need not be considered.

13. Drive - pulley ratio T_1/T_2 :

Total operating tension T_1 = 2430 lbs.

Minimum fixed tension T_2 = 630 lbs.

$T_1/T_2 = 3.86$ (max.)

From Appendix C5, for an LPS 220 drive, the permissible $T_1/T_2 = 3.83$ for operating. This can be increased 50% for starting to 5.75. Thus the drive ratio is satisfactory.

14. Minimum pulley diameters:

For a 4-ply belt at operating stress level of around 60% of the maximum permissible, the minimum recommended pulley diameter in Appendix C7 is 16 inch. Usually for low stresses, a smaller diameter like 12 inch would also be satisfactory.

15. Conveyor structure/frame:

Structural supports for the usual kinds of conveyors generally consist of 5, 6, or 7 inch channel (C-section) stringers, depending on the width of the belt and the support span. For very long conveyors, light weight truss may be used to permit longer spans for more economical construction. Since for the present design, only light loads are involved, a C 6 × 13 structural steel section (channel section having 6 inch depth and 13 lbs. per ft. weight) should be satisfactory for the support spacing shown in Figure 13.

PNEUMATIC CONVEYING OF WHEATSTRAW

Introduction:

The board mills of Packages Limited, being a paper mill, uses fibrous raw material to produce pulp which is subsequently converted to paper or paperboard. There are several sources of these fibers, such as soft wood chip, baggase, wheatstraw and kahi- a kind of wild grass. The pulp at Packages is produced from soft wood, recycled waste paper and from locally available farm by-products such as wheatstraw and kahi.

As wheatstraw and kahi are both seasonally available raw materials, they are purchased during their peak availability seasons and stored at site premises for subsequent use throughout the year. The storage is in the form of huge stacks in the open space available at plant site. As the material is low density straw, the storage occupies considerable space. Thus, some stacks are quite close to the straw preparation plant while others are away towards the factory boundary. The total area covered, and the factory layout plan is given in Appendix E.

At present there is no adequately mechanized system to handle this material from its stacked locations to straw preparation plant. Two tractor trollies are used to transport this material from the stacks to the preparation department. From there, after cutting and cleaning, it is conveyed to the cooking house for further processing. The trollies require manual loading of straw which makes its handling very slow and labor intensive. This is undesirable because, if allowed to remain for too long, the temperature in a particular stack might build up to its flash point. Quick transport to the straw preparation section for processing is therefore very desirable to avoid fire hazard. All these factors demand an efficient, fast and reliable system with several pick up points to suit different stack groups.

Two of the common bulk material handling systems are: belt conveyors and pneumatic conveyors. Belt conveyors usually result in low running costs, but initial installation costs can be substantial. Besides the belt can cause undue hinderances if it has to run along the plant buildings or streets. Moreover, in this particular case, belt conveying would not mechanize the material loading process and the conveying would be slow and sluggish. Pneumatic conveying can easily overcome these problems. It is flexible and the installations can conveniently run along plant buildings. Also it is weather proof, rugged and maintenance free. However, the mechanical efficiency of pneumatic systems is low. Sometimes it can result in a very high power consumption per ton of material which can upset its other advantages. Moreover, the design process is still an art which has to be learned through experience and trial runs on experimental lines with the materials which are to be conveyed pneumatically.

This project basically covered the preliminary design calculations for the test line. It should provide further information about the overall system design and the feasibility of extending the system to the entire stacked area. The proposed test line (about 400 ft) would handle straw from the stacks nearest to the straw preparation department. It is shown in the site layout plan in Appendix E along with the future possible line to link all stacks through a common pneumatic system. Some modifications in the stacking arrangement would also be required to limit the number of pick-up points.

At any time, the material would be conveyed from one stack only. Therefore, it would not be feasible to use only one blower for the entire line as this would require expensive compressing equipment capable of handling losses in the entire length. This would push the initial capital and running costs unnecessarily high even when conveying material from a closeby stack.

A tandem fan arrangement might provide a better alternative to the medium pressure equipment. In this arrangement, each blower would actually convey material from one stack location to up to, say about, 400 ft. (about 125 m). From there the next fan would pick it up for further delivery. This would continue until the material has been transported to the desired location (straw preparation plant). With this arrangement it might be possible (depending on pressure requirements) to use locally manufactured centrifugal blowers satisfactorily. This would considerably cut the cost and time to import expensive equipment that might be necessary otherwise.

The following section gives the design procedure as described by Stoess* for sizing different parts of pneumatic systems. The calculations are presented for the proposed experimental line.

Design Procedure:

To design a pneumatic conveying system, decision has to be made about several variables such as:

- Conveying air velocity to keep the material air borne.
- Cubic ft/min (cfm) of air required per pound of material to be conveyed.
- Pipe size to suit the air flow requirments.
- Estimate of energy loss due to air and material flow.
- Pressure requirements in the line.

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*H. A. Stoess Jr. P.E., "Pneumatic Conveying", John Wiley and Sons, 1983.

- Air mover or blower selection.

- Horsepower requirments for the air mover.

The calculations for the above factors are mainly based on information developed during actual tests with different materials.

Design Calculations:

- Material to be conveyed : Shredded wheatstraw.
- Specific weight of material, $W \simeq 8$ lbs/ft^3 (dry)

  $\simeq 16$ lbs/ft^3 (wet)
- Capacity required, $C \simeq 600$ lbs/min.
- Conveying length, $L = 125$ meters $\simeq 400$ ft.

1. Saturation and conveying velocity requirement: From Appendix D1, to convey wood flour (similar to shredded wheatstraw) to a distance of around 400 ft., the following is applicable:

- Saturation (cubic ft. of air/lb of material conveyed per minute) = 4.4
- HP/ton (hp required to convey 1 ton of material in one minute) = 6.5
- Conveying velocity, $V = 100$ ft. per sec.

As friction depends on velocity and pipe diameter, so for larger pipes lower saturations can be used. For an 8–in. and 10–in. pipes, saturation and hp/ton can be reduced 15% and 25% respectively.

2. Capacity required: For the required capacity of 600 lbs/min, if an 8–in. pipe is sufficient, then the volume of free air required for the system is given by:

$$SCFM = \text{Saturation} \times \text{Conveying rate (lbs/min.)}$$

$$= [4.4 - 4.4 \times 0.15] \times 600 = 2244$$

3. **Pipe size:** The size of the conveying pipe is determined by computing the pipe constant given by the relation:

Pipe constant = SCFM/Velocity(fps) = 2244/100 = 22.44

Refering to Appendix D2 for pipe constants, an 8-in. schedule 10 pipe has a pipe constant of 22.7. As this is larger than the required 22.44, so this pipe should be satisfactory. In order to maintain proper velocity in the conveying pipeline, the volume of free air (scfm) would be calculated again based on pipe constant of 22.7.

$$\text{SCFM} = \text{pipe constant} \times \text{Velocity}$$
$$= 22.7 \times 100 = 2270 \text{ cubic ft./min.}$$

Corrected saturation will be : SCFM/Conveying rate = 2270/600 = 3.78

4. **Vaccum required:** The vaccum at which the system will operate when conveying at its rated capacity is determined by:

$$\text{Vaccum factor} = [\text{hp/ton}]/\text{Saturation}$$
$$= [6.5 - 6.5\times.15]/3.78 = 1.46$$

This corresponds to a vaccum of about 10-in. of Hg.

With SCFM and the vaccum requirement of the system known, the size of the blower needed to activate the system can now be determined. This is done by determining the actual amount of air, ACFM, the unit is to inhale. The ACFM represents the expanded air at intake conditions to the blower. It is calculated as follows:

$$\text{ACFM} = [\text{SCFM} \times 30 \text{ (in. of Hg)}]/[30 \text{ (in. of Hg)} - \text{Operating vaccum}]$$
$$= [2270 \times 30]/[30 - 10]$$
$$= 3405 \text{ ft}^3/\text{min.}$$

6. **Blower size:** This is determined by consulting manufacturer's performance catalogues. It is recommended that the blowers in pneumatic conveying, service at

15% below the maximum operating speed recommended by the manufacturer. This is to allow some margin for the final tune-up which might be required once the system operation gets underway. The choice on the type of blower is based on the severity of operation. However for the problem under consideration, a locally available blower has to used for which performance data may not be available. Therefore, the selection would have to be based on experience or trial and error.

7. **Blower speed:** The speed of the blower is determined using the following expression:

$$\text{RPM} = \text{ACFM/Blower displacement} \quad + \quad \text{slip allowance}$$

where information regarding slip and displacement is given in manufacturer's catalogues.

8. **HP requirment:** The horsepower required to drive the blower is calculated by:

$$\text{HP} = \text{rpm} \times \text{displacement (cf/r)} \times [\text{ vaccum/2}] \times 0.005$$

In case complete information about blower performance is not available, the following formula can be used for an approximate estimate of hp requirements.

$$\text{HP (approx.)} = \text{ACFM} \times 1.20 \times [\text{Vaccum/2}] \times 0.005$$

Thus for the problem under consideration,

$$\text{HP (approx)} = 3405 \times 1.20 \times 5 \times 0.005$$

$$= \quad 102 \text{ bhp} \simeq 77\text{kw motor required}$$

The above calculations give an idea of the amount of power requirements for the pneumatic systems. As can be seen the power requirements are substantial, but considering that the system will be needed for only a few hours per day, it might still be economically feasible. An actual system evaluation should be possible after the experimental line with a locally manufactured blower proves successful.

All the preceding formulas and calculations were based on sea level conditions of 29.92 in. of Hg absolute pressure and 70 °F. At elevated altitudes, the weight of air is reduced, thus to approximate the same conveying conditions there as at sea level, corrections are necessary to the SCFM determined in step 2. The pipeline size remains more or less the same as at sea level.

MODIFICATIONS IN PNEUMATIC PAPER TRIM HANDLING SYSTEM

This consisted of two modification assignments: First two separate pneumatic lines of the paper converting (PC) and solid board departments (referred subsequently as line-1 and line-2) were merged into a single integrated line. The second job involved shifting blower-2 to a new location and the associated route change of the pipeline. This was necessitated as blower-2 was located very close to the factory boundary wall where its noise and vibration was creating a nuisance for the nearby residents. They had warned of a legal action against the company if nothing was done about it. The line diagram and the system details are given in Figure 14. Also see the site layout plan in Appendix E for actual pipeline route indication.

In the first phase of connecting two separate pneumatic lines, there were not many problems. Line-2 conveyed the solid board wastage (after shredding) to the waste paper shed for subsequent recycling. The overall line length was about 285 meters (900 ft.), with two pick up points. An in-line centrifugal fan with a 920 mm diameter (about 36 in.) impeller, running at 1900 rpm with a 90 kw motor was used as air mover for this line. As the paper trim passed through the fan wheel it had to be structurally rugged to withstand the resulting shock loads. The other line, line-1, was about 135 meters (430 ft.) long and conveyed paper trim from paper converting department to the bailing plant. From there the compact bails were moved to the waste paper shed on trollies for recycling.

The bailing was unnecessary as line-1 could directly feed into line-2 which could then convey the PC trim further to the waste paper shed. The only requirement would be that at any time the entire integrated line would be available to handle waste from either PC or solid board. This should not create any difficulty as the average daily usage of this system for PC or solid board would not be more than 2 to 3 hours. The line-2 was larger (500mm or 19 in. dia.) and its fan cfm were higher compared to the PC line (350mm or 14 in. dia.). It was realized that if the two lines were connected with an expander, the pressure side of blower-1 leading into the suction side of line-2 might substantially reduce the suction pressure in line-2. To keep changes to a minimum, line-1 was extended to just lead the trim into the solid board line as shown in Figure 15. From there the blower-2 picked up the trim, and the additional air necessary to make up for its cfm, to convey the material to its final destination - the waste paper shed. When the system was put to work, it functioned properly requiring only a slight increase in blower-1 rpm. This was needed to compensate for the increased frictional loss due to an increase in the line length by about 20 meters along the new line-1 route.

The second assignment on this system basically involved line-2. In the integrated system the performance of this part obviously effected the entire system. It was decided to move blower-2 which was the source of noise and vibration, from its existing location near the factory boundary to a new location removed from the boundary. The blower noise consisted of the high frequency motor noise and the noise due to the passage of the paper trim through the impeller. It was necesary to pass the material through the impeller as it helped in shredding and dispersing the trim in the pressure line, resulting in better conveying with reduced chances of choking.

Locating a new spot for the blower was not an easy task. For this the towers supporting the pipeline and the entire blower support structure also had to be moved. Also, the power connection for the blower had to be easily available at the new site.

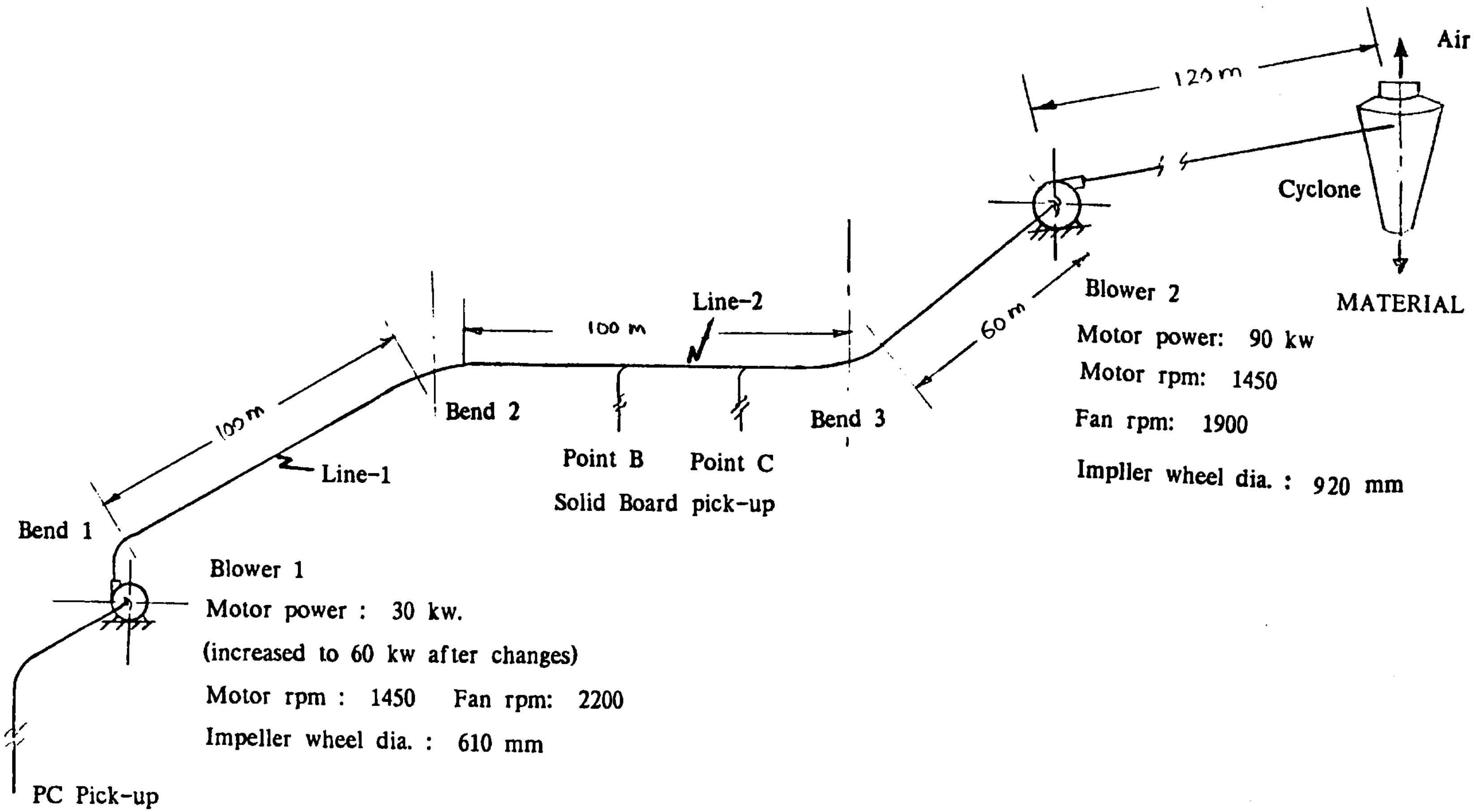

Figure 14. Pneumatic Paper Trim Handling System.

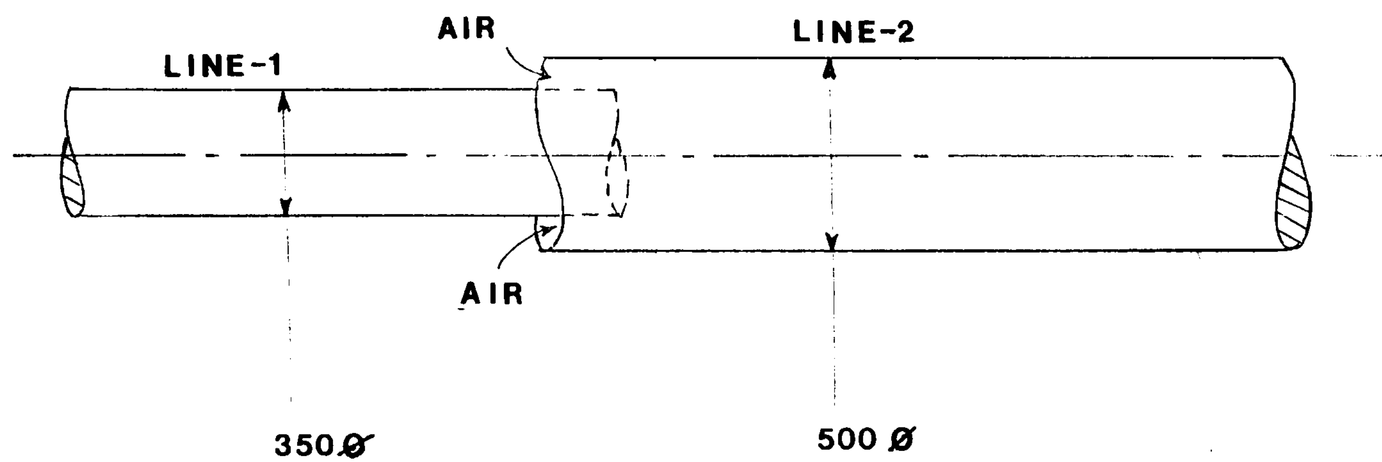

Figure 15. Line-1 and Line-2 Connection for the Paper Trim Handling System.

Keeping all these factors in view along with the effect it would have on the system performance, a decision had to be made to move the blower. This was particularly difficult as the pressure, velocity, frictional loss and actual cfm were all unknowns. Therefore, the decisions had to based on engineering intuition and experience.

After some checks and surveys, a new location that would remove the blower from creating public nuisance and also require minimum changes was established. This required blower-2 to be moved away by about 60 meters (about 200ft.) from its existing location. See site plan drawing in Appendix E. The resulting increase in suction length (60 m) was expected to lead to a weak suction, but the increased suction length was also being compensated somewhat by a decrease in the pressure side by the same amount. So the effect on overall performance was difficult to predict, particulary when no information was available. It was anticipated that with some manipulations of the blower-2 rpm, the system would be able to function smoothly.

It should be understood that there were not very many places on the premises where the blower could be shifted. Its best location was probably the one where it was already placed. Anyway, after actual changes were made, the suction pressure drop was considerable leading to frequent choking. An increase in blower rpm was then planned. However, a check with the electrical department revealed that any additional load on the power line was not possible as the system was already close to being overloaded. Attempts to improve the suction by increasing the radius of the bend-3 (Figure 15) did not result in substantial improvement and the system remained prone to choking. Eventually it was decided to decrease the suction length of line-2 and accommodate it by increasing the delivery length for blower-1. Obviously this required a bigger motor for blower-1 to increase its capacity. A decrease of about 40 meters (125 ft.) in the suction for line-2 with a corresponding increase in the pressure side of line-1 led to satisfactory performance.

This project exposed the intern to decision making when little information is available, and the task has to accomplished from experience and by trial and error adjustments. It provided exposure to interaction with various departments to schedule their services, as required, without conflicts or waste of time. The lesson learned was that it always takes more time to do a job than is usually obvious at first. Besides it is easy to overlook a few simple things which can cause undue trouble later. So each possibility should be very carefully checked before giving the go-ahead signal.

CHAPTER III

INDUSTRIAL MANAGEMENT

INTRODUCTION

The internship, in addition to the technical tasks, also provided experience in industrial management. The experience covered technical project management and the supervision of a team of technical personnel. Some exposure to workshops and labor environment was also possible.

TECHNICAL MANAGEMENT

The assignments for the intern were usually routed through the Senior Engineer-Utilities. At times, the user departments would directly contact him for their maintenance, modification or development plans requiring design and development related support. The technical assignments were usually not such that a quick "cook-book" solution could be easily found and communicated to the superior or to the user department. In fact, before different possible solutions could be looked into, the problem at hand and the scope of the desired modification or development had to be properly understood. This required interaction with various departments in the company at different levels to extract relevant information. Many times such information was neither accurate nor easily available. Thus assumptions and guesses had to be relied upon, which of course had to be carefully used, keeping in view the perceptions and limitations of the people providing the information.

Cost Estimation:

The cost estimates for doing modification jobs using the company's manufacturing facilities were often needed. The estimates were prepared by checking material costs from the supply and inventory sections to work out total direct material costs. To this the machine and labor costs needed to do the job were added. This information was available from the workshops planning section which maintained records for different machine and labor rates. The direct material and labor costs were further adjusted for miscellaneous and unforeseen costs to get the total direct cost. Such estimates were usually prepared for virtually all activities to establish the feasibility of doing things internally. In the beginning these estimates were prepared in a rather elaborate fashion by the intern which amounted to an undue time being spent on simple things. It was later found that a few rules of thumb usually result in as good estimates a lot quicker and with a lesser effort. These rules which are based on company experience and expense for different activities were subsequently followed.

Codes and Standards:

During the internship proper codes and standards for some job assignments were not available. This led to a need to rely on experience or to work things completely from ab initio. This approach sometimes forced technical decisions to remain speculative. For instance, the non-availability of proper design codes created difficulty while designing paper and cattle shed trusses. Some effort was, therefore, directed towards collecting technical standards. The intern did some literature search and established proper design procedures. Similarly, the non-availability of performance characteristic curves for the blower fans in the pneumatic paper trim handling system necessitated trial and error adjustments to obtain satisfactory performance. These things, though frustrating, were quite representative of industrial situations.

Workshops Management:

As the design and development department worked in close association with the mechanical workshop facilities, working with this group provided ample exposure to workshop related problems and methods of management, planning and labor relations. The art of "push and pull" in labor management was appreciated. Such tacts can usually be learned only in actual real life situations and would differ in different environments and cultures.

The exposure related to dealing with salespersons supplying materials to the manufacturing departments was also valuable. The magnitude of price variation on different supplies from one supplier to another for the same quality and even on the same product was quite astonishing. One had to be rather well aware of prices to negotiate effectively with salesmen without getting trapped in their wits.

SUPERVISORY RESPONSIBILITIES

This consisted of supervising the technical personnel in the Design and Development department in the capacity of Senior Design Engineer. Apart from purely technical tasks, the responsibilities included assigning and monitoring the work of other members of the department. The intern usually had to break-up technical projects into smaller sections to be assigned to the individual members for detailed design. Such break-up helped in group involvement and promoted a spirit of team work. The supervisory duties provided exposure to the complex nature of managing people. The factors such as motivation, morale, interpersonal rivalaries, intergroup bickerings, status syndrome, promotions, perceptions of recognition, and the value of team work in the organizational management were duely recognized.

An important observation related to the differences in individual abilities in a group. Generally everyone has some unique capabilities and some limitations. For successful management, the executive has to know the strengths and the weaknesses of his team and should try to optimize performance within these constraints. Some development courses should also be planned from time to time to develop individual skills. The company frequently offered supervisory and industrial safety related courses. Suitable technical study circles, however, were non-existent at the local level. This was compensated somewhat by the company sponsored professional development programs abroad for its technical staff.

Another observation was the fear of loss of job which seemed present among most employees even though the company never had any major layoffs. These fears are not easy to remove. It is not clear whether such environment improves performance or simply forces people to work just enough to keep up their jobs. The intern experience, however, suggests that people perform better in a secure environment if responsibility is clearly defined and due credit is given for the job well done. The problems usually arise when the role and responsibility are not clear, or at least the employee and the management perceive them differently.

As the company experienced considerable expansion during 1984, there were some vacancies in the group because some members had been assigned elsewhere. For these, new replacements were planned. This led the intern to conduct some interviews for junior engineers to establish their suitability for placement in the department. The evaluations were communicated to the superiors for further action.

The intern feels the work assignments provided a wholesome experience and adequately fulfilled the internship requirements of the Doctor of Engineering program. This was largely due to the cooperation extended to him at all levels at Packages for which he is grateful.

CHAPTER IV

SUMMARY AND CONCLUSIONS

This report outlines the experience acquired at Packages Limited, to fulfill the internship requirements of the Doctor of Engineering degree. The intern feels that these requirements were adequately met during the internship.

The technical assignments provided exposure to the industrial approach of solving problems. This required finding out relevant information and, sometimes making decisions without complete information. The assignments required interaction with various departments and people at different levels. The results depended on team work. Functioning merely as an individual was just not enough. The internship also provided opportunities for some very useful exposure to the organizational management, human factors and behavior building in organizations. Such experience simply could not be acquired in a classroom environment.

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This Report was typed by the author on Wylbur using SCRIPT BLUELINE.

APPENDICES

APPENDIX-A

The purlins can be designed either as simply supported beams spanning from truss to truss or as continuous beams supported by a number of trusses. In our design arrangement, the 200 ft. purlin was supported at nine points equally spaced at 25 ft each. Thus they can be treated as continuous beams. The following expressions for continuous beams with two equal spans and uniformly distributed load can be used to estimate purlin size:

$$\text{Max. stress, } \sigma = W\ l/8\ Z \tag{1A}$$

$$\text{Max. deflection, } \delta = W\ l^3/185EI \tag{2A}$$

where:

W = Total load on each span

l = Span length

Z = Section modulus

E = Modulus of elasticity

I = Moment of inertia

Total load / purlin = sheet load/span + Wt. of purlin itself/span

$$W = 1.25\times25\times4 + 5\times25$$

$$= 125 + 125 = 250 \text{ lbs.}$$

For steels $E = 30\times10^6$ psi. Also for most civil works, the following thumb rule is used for permissible deflection of beams:

$$\delta = l/350 = 25 \times 12 / 350 = 0.86 \text{ inch.}$$

Using this in equation-2A, the following value is obtained for I:

$$I = W\ l^3/185E\delta$$

$$= 250 \times (25\times12)^3/185\times30\times10^6\times0.86$$

$$= 1.14 \text{ in.}^4$$

From the standard structural steel tables, we find that a channel section C3×4.1 has I about xx-axis of about 1.66 in^4 and weighs 4.1 lbs/ft. It should keep the deflection below the permissible limit of 1 inch.

A check for stress can be obtained from equation-1A, which gives:

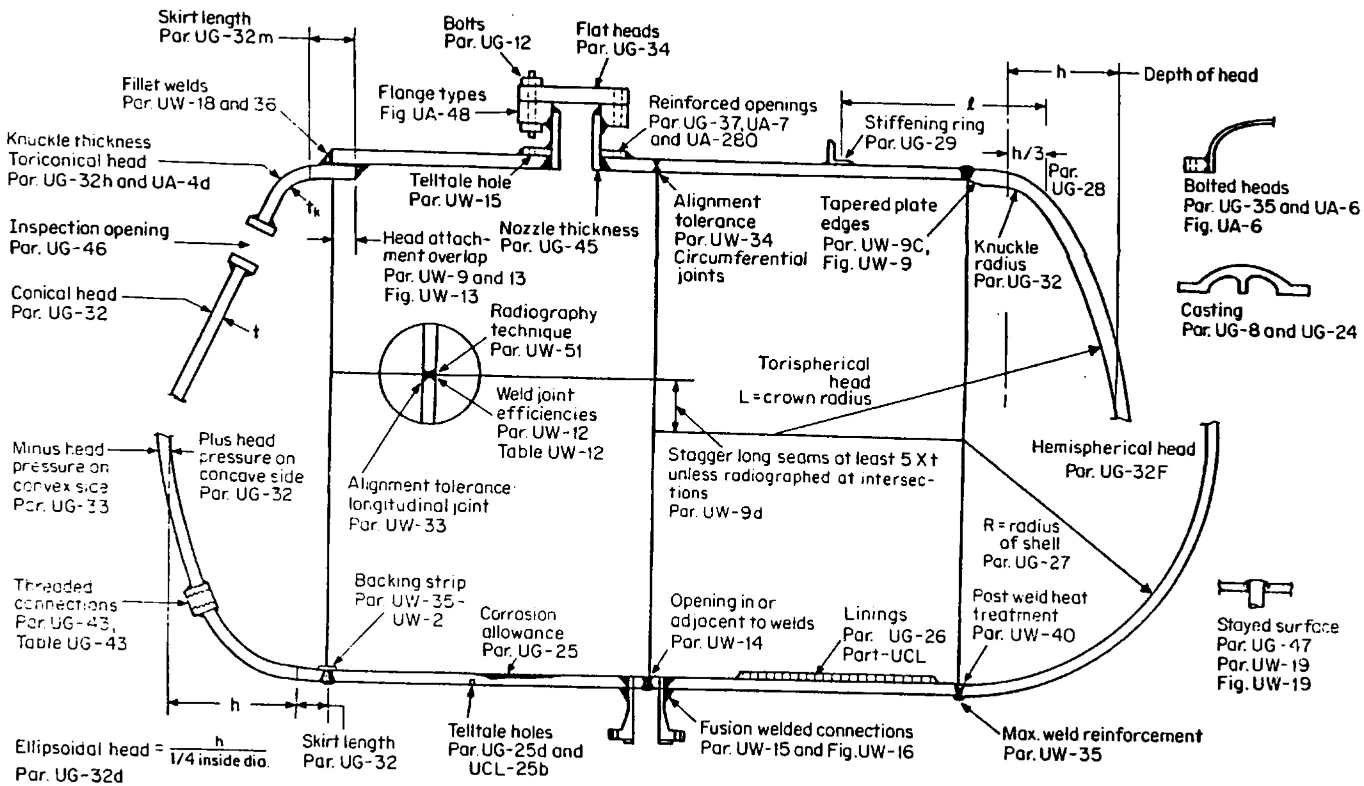
$$\text{Max. stress} = 250\times25\times12/8\times1.10$$

$$= 8{,}523 \text{ psi}$$

which is well below the yield strength of structural steel (about 40,000 psi).

Thus structural steel section C 3×4.1, results in deflection and stress values which are well below their limiting values. Therefore it should be a satisfactory choice for use as purlin.

APPENDIX : B



Reference chart for *ASME Pressure Vessel Code*, Section VIII, Division 1.

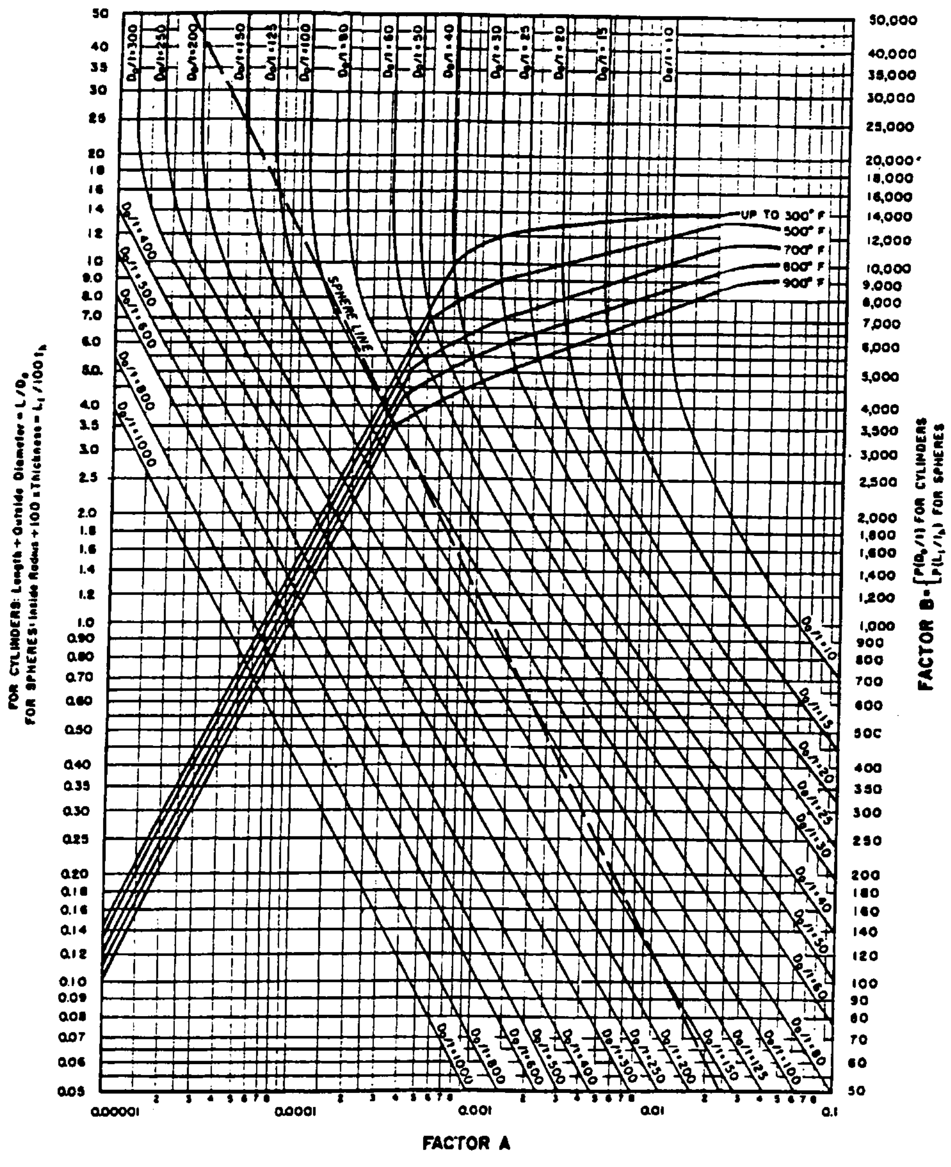


FIG. UCS-28.1 CHART FOR DETERMINING SHELL THICKNESS OF CYLINDRICAL AND SPHERICAL VESSELS UNDER EXTERNAL PRESSURE WHEN CONSTRUCTED OF CARBON STEEL
(Specified Yield Strength 24,000 psi to, but not including, 30,000 psi)

Fig. UCS-28.2

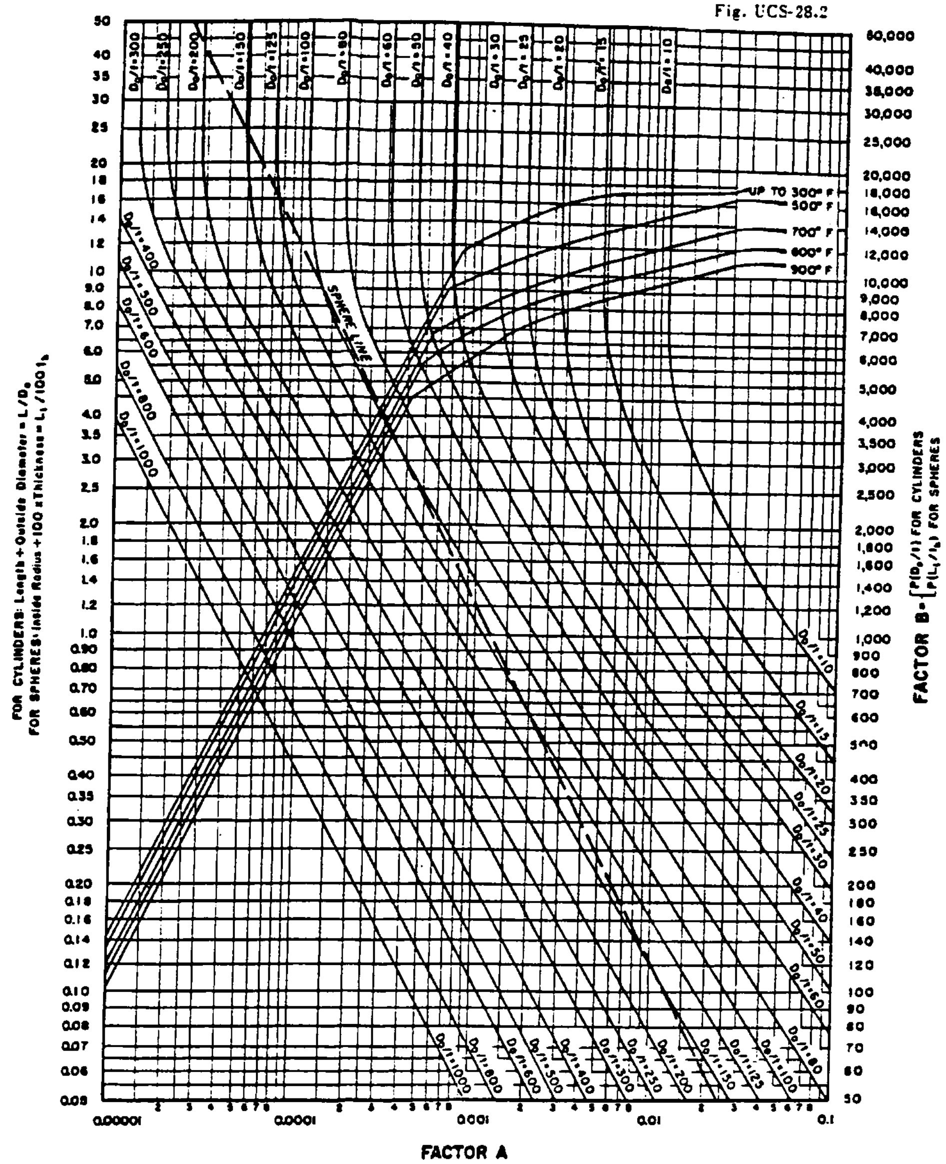


FIG. UCS-28.2 CHART FOR DETERMINING SHELL THICKNESS OF CYLINDRICAL AND SPHERICAL VESSELS UNDER EXTERNAL PRESSURE WHEN CONSTRUCTED OF CARBON STEEL (Specified Yield Strength 30,000 to 38,000 psi inclusive) AND TYPE 405 AND TYPE 410 STAINLESS STEEL

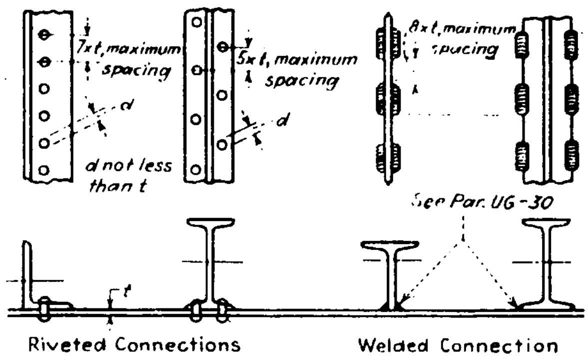
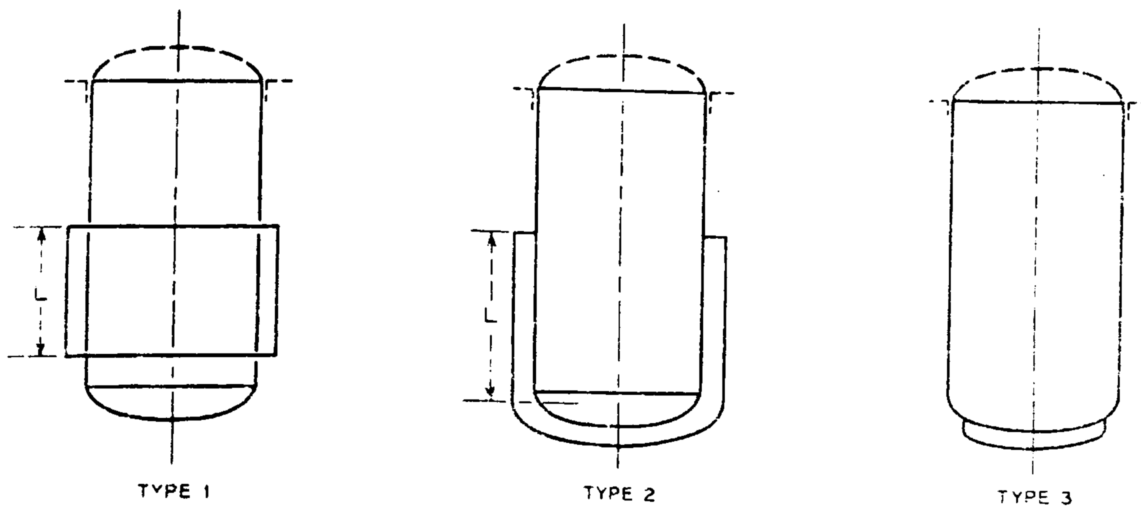


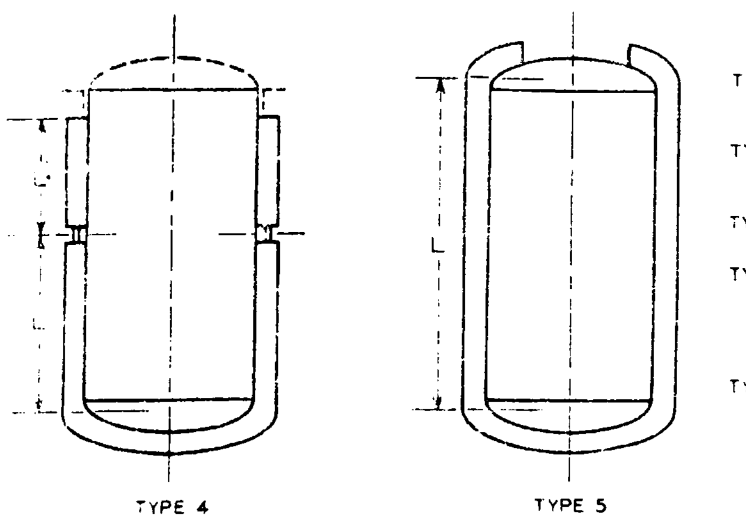
FIG. UG-30 SOME ACCEPTABLE METHODS OF ATTACHING STIFFENING RINGS TO SHELLS OF CYLINDRICAL VESSELS SUBJECTED TO EXTERNAL PRESSURE

TABLE UG-37
VALUES OF SPHERICAL RADIUS FACTOR K_1

(Equivalent Spherical Radius = $K_1 D$; $D/2h$ = axis ratio. For definitions see Par. UA-4 (b). Interpolation permitted for intermediate values.)

$\frac{D}{2h}$		3.0	2.8	2.6	2.4	2.2
K_1		1.36	1.27	1.18	1.08	0.99
$\frac{D}{2h}$	2.0	1.8	1.6	1.4	1.2	1.0
K_1	0.90	0.81	0.73	0.65	0.57	0.50





TYPE 1 – JACKET OF ANY LENGTH CONFINED ENTIRELY TO CYLINDRICAL SHELL.

TYPE 2 – JACKET COVERING A PORTION OF CYLINDRICAL SHELL AND ONE HEAD.

TYPE 3 – JACKET COVERING A PORTION OF HEAD.

TYPE 4 – JACKET WITH ADDITION OF STAY OR EQUALIZER RINGS TO THE CYLINDRICAL SHELL PORTION TO REDUCE EFFECTIVE LENGTH.

TYPE 5 – JACKET COVERING CYLINDRICAL SHELL AND ANY PORTION OF EITHER HEAD.

FIG. UA-101 SOME ACCEPTABLE TYPES OF JACKETED VESSELS

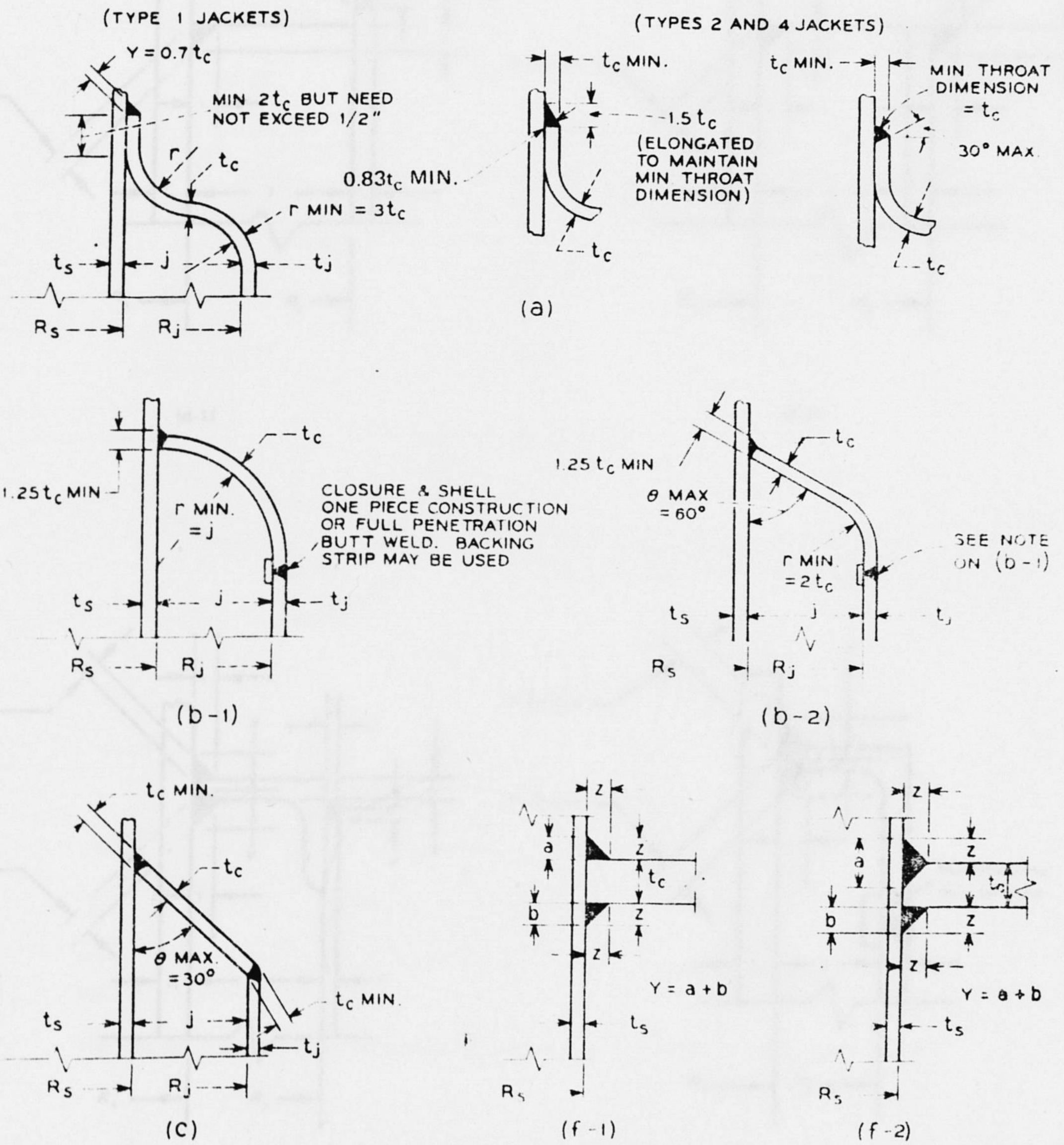


FIG. UA-104 SOME ACCEPTABLE TYPES OF JACKET CLOSURES

[(d-1), (d-2), (e-1), and (e-2) appear on page 319.1]

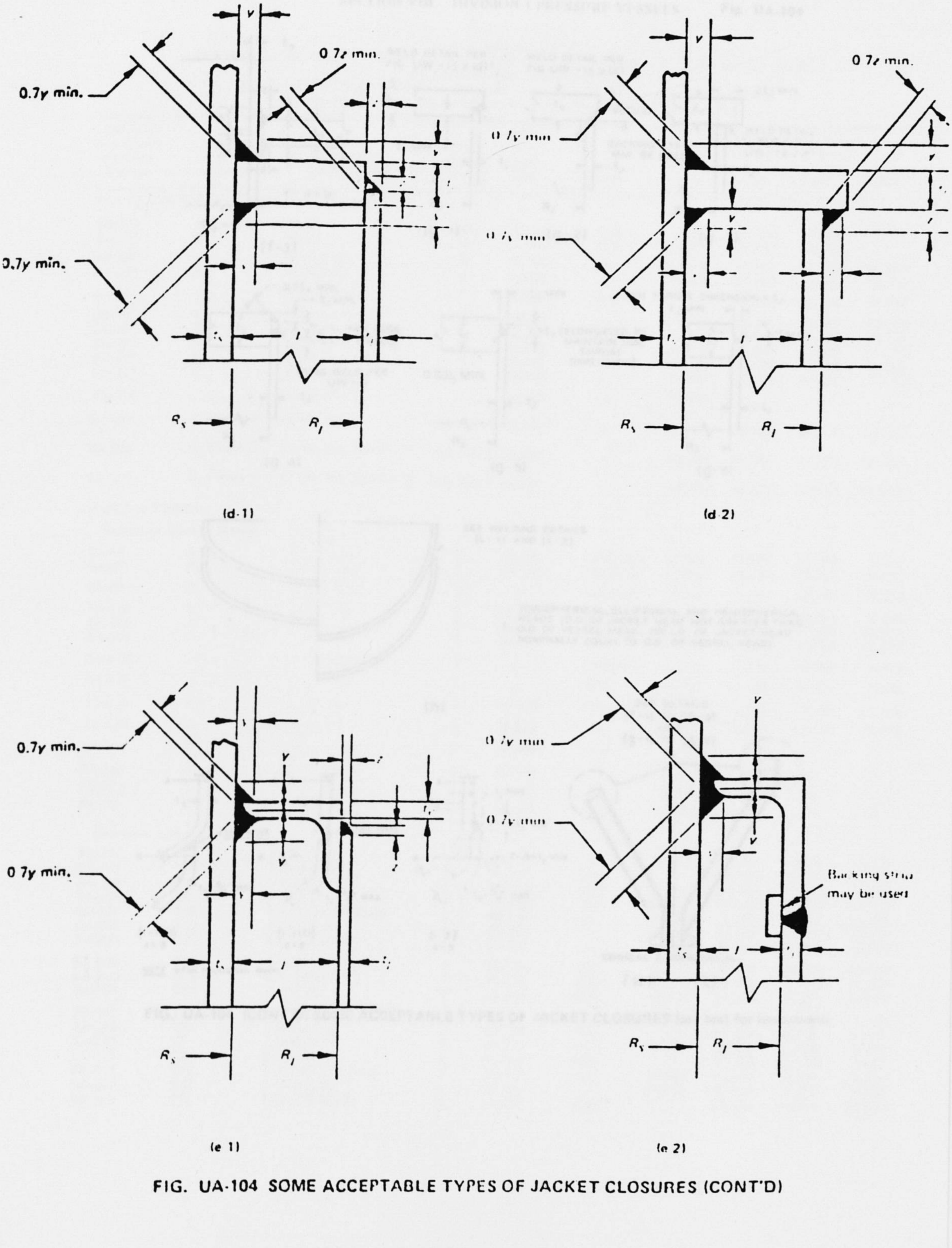


FIG. UA-104 SOME ACCEPTABLE TYPES OF JACKET CLOSURES (CONT'D)

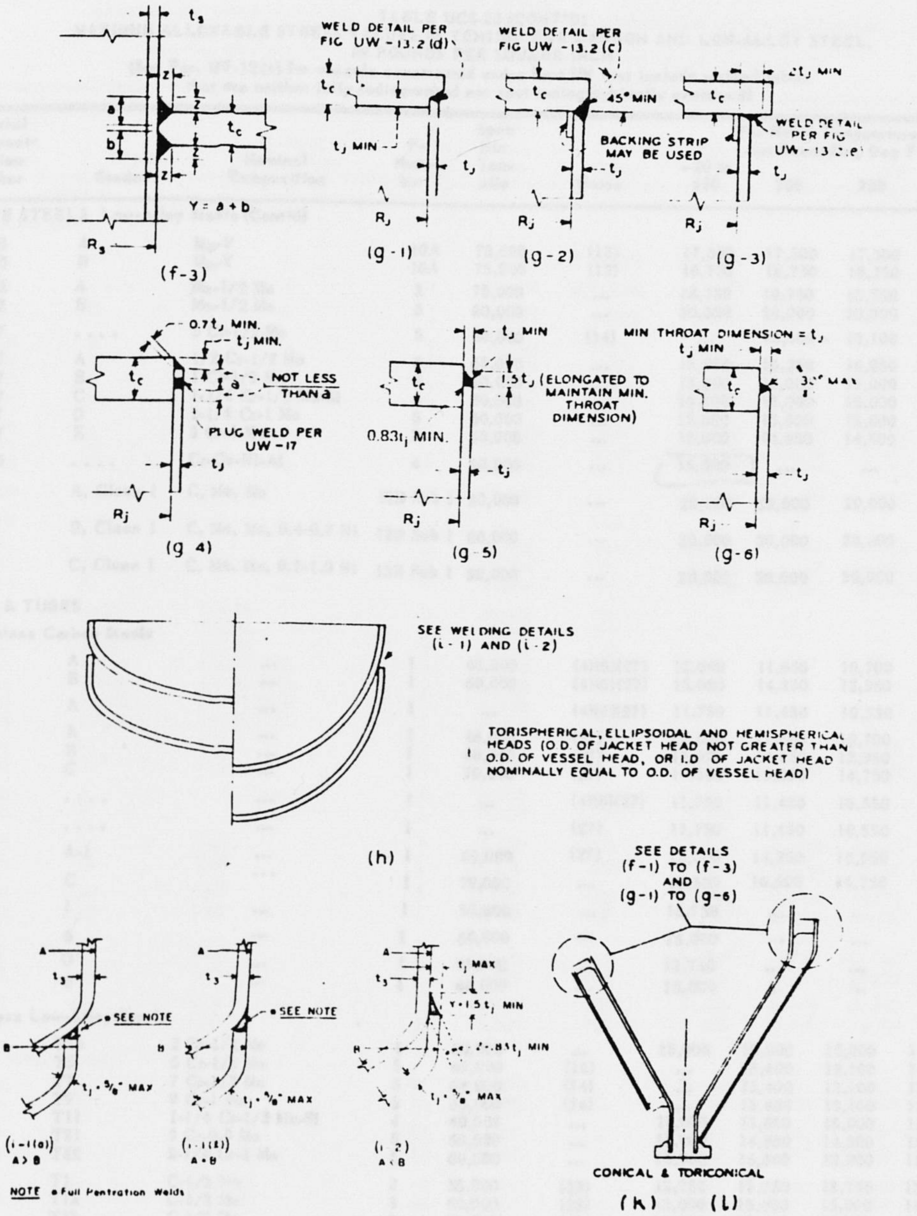


FIG. UA-104 (CONT'D) SOME ACCEPTABLE TYPES OF JACKET CLOSURES (see text for limitations)

TABLE UCS-23 (CONT'D)
MAXIMUM ALLOWABLE STRESS VALUES IN TENSION FOR CARBON AND LOW-ALLOY STEEL, IN POUNDS PER SQUARE INCH
(See Par. UW-12(c) for vessels constructed under Part UW that include welded joints that are neither fully radiographed nor spot radiographically examined)

Material and Specification Number	Grade	Nominal Composition	P-Number	Spec Min Tensile	Notes	For Metal Temperatures Not Exceeding Deg F: −20 to 650	700	750	800
PLATE STEELS Low-Alloy Steels (Cont'd)									
SA-225	A	Mn-V	10A	70,000	(12)	17,500	17,500	17,500	...
SA-225	B	Mn-V	10A	75,000	(12)	18,750	18,750	18,750	...
SA-302	A	Mn-1/2 Mo	3	75,000	...	18,750	18,750	18,750	18,000
SA-302	B	Mn-1/2 Mo	3	80,000	...	20,000	20,000	20,000	19,100
SA-357	....	5 Cr-1/2 Mo	5	60,000	(14)	...	13,400	13,100	12,800
SA-387	A	1/2 Cr-1/2 Mo	3	65,000	...	16,250	16,250	16,250	15,650
SA-387	B	1 Cr-1/2 Mo	4	60,000	...	15,000	15,000	15,000	14,750
SA-387	C	1-1/4 Cr-1/2 Mo-Si	4	60,000	...	15,000	15,000	15,000	15,000
SA-387	D	2-1/4 Cr-1 Mo	5	60,000	...	15,000	15,000	15,000	15,000
SA-387	E	3 Cr-1 Mo	5	60,000	...	15,000	14,800	14,500	13,900
SA-410	....	Cr-Cu-Ni-Al	4	60,000	...	15,000	...	...	...
SA 533	A, Class 1	C, Mn, Mo	12B Sub 1	80,000	...	20,000	20,000	20,000	19,100
SA 533	B, Class 1	C, Mn, Mo, 0.4-0.7 Ni	12B Sub 1	80,000	...	20,000	20,000	20,000	19,100
SA 533	C, Class 1	C, Mn, Mo, 0.7-1.0 Ni	12B Sub 1	80,000	...	20,000	20,000	20,000	19,100
PIPES & TUBES									
Seamless Carbon Steels									
SA-53	A	...	1	48,000	(4)(6)(27)	12,000	11,650	10,700	9,300
SA-53	B	...	1	60,000	(4)(6)(27)	15,000	14,350	12,950	10,800
SA-83	A	...	1	...	(4)(6)(27)	11,750	11,450	10,550	9,200
SA-106	A	...	1	48,000	(27)	12,000	11,650	10,700	9,300
SA-106	B	...	1	60,000	(27)	15,000	14,350	12,950	10,800
SA-106	C	...	1	70,000	(27)	17,500	16,600	14,750	12,000
SA-179	....	...	1		(4)(6)(27)	11,750	11,450	10,550	9,200
SA-192	....	...	1	...	(27)	11,750	11,450	10,550	9,200
SA-210	A-1	...	1	60,000	(27)	15,000	14,350	12,950	10,800
SA-210	C	...	1	70,000	...	17,500	16,600	14,750	12,000
SA-333	1	...	1	55,000	...	13,750	...	...	...
SA-333	6	...	1	60,000	...	15,000	...	...	...
SA-334	0	...	1	55,000	...	13,750	...	...	...
SA-334	6	...	1	60,000	...	15,000	...	...	...
Seamless Low-Alloy Steels									
SA-199	T3b	2 Cr-1/2 Mo	4	60,000	...	15,000	15,000	15,000	14,700
SA-199	T5	5 Cr-1/2 Mo	5	60,000	(14)	...	13,400	13,100	12,800
SA-199	T7	7 Cr-1/2 Mo	5	60,000	(14)	...	13,400	13,100	12,500
SA-199	T9	9 Cr-1 Mo	5	60,000	(14)	...	13,400	13,100	12,800
SA-199	T11	1-1/4 Cr-1/2 Mo-Si	4	60,000	...	15,000	15,000	15,000	15,000
SA-199	T21	3 Cr-0.9 Mo	5	60,000	...	15,000	14,800	14,500	13,900
SA-199	T22	2-1/4 Cr-1 Mo	5	60,000	...	15,000	15,000	15,000	15,000
SA-209	T1	C-1/2 Mo	3	55,000	(28)	13,750	13,750	13,750	13,450
SA-209	T1a	C-1/2 Mo	3	60,000	(28)	15,000	15,000	15,000	14,400
SA-209	T1b	C-1/2 Mo	3	53,000	(28)	13,250	13,250	13,250	13,000
SA-213	T2	1/2 Cr-1/2 Mo	3	60,000	...	15,000	15,000	15,000	14,400
SA-213	T5	5 Cr-1/2 Mo	5	60,000	(14)	...	13,400	13,100	12,800
SA-213	T7	7 Cr-1/2 Mo	5	60,000	(14)	...	13,400	13,100	12,500
SA-213	T9	9 Cr-1 Mo	5	60,000	(14)	...	13,400	13,100	12,800
SA-213	T11	1-1/4 Cr-1/2 Mo-Si	4	60,000	...	15,000	15,000	15,000	15,000
SA-213	T12	1 Cr-1/2 Mo	4	60,000	...	15,000	15,000	15,000	14,750
SA-213	T5b	5 Cr-1/2 Mo-Si	5	60,000	(14)	...	13,400	13,100	12,800

TABLE UCS-23
MAXIMUM ALLOWABLE STRESS VALUES IN TENSION FOR CARBON AND LOW-ALLOY STEEL, IN POUNDS PER SQUARE INCH

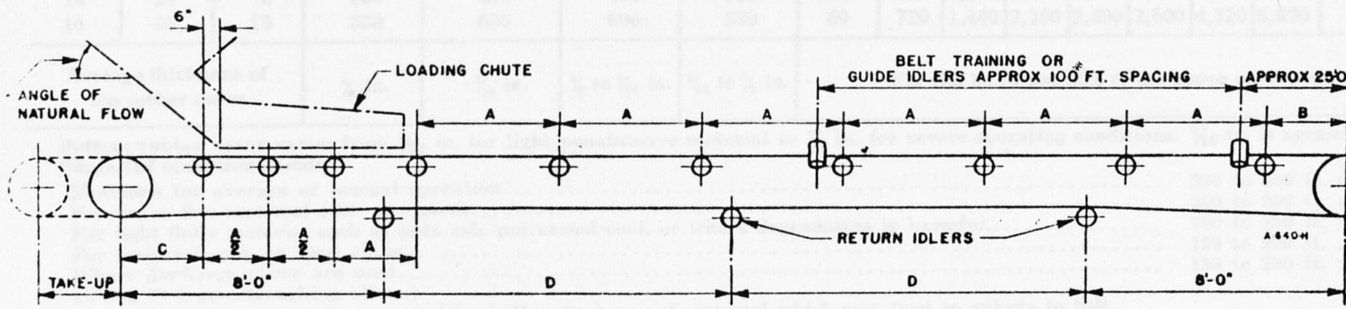
(See Par. UW-12 (c) for vessels constructed under Part UW that include welded joints that are neither fully radiographed nor spot radiographically examined)

Material and Specification Number	Grade	Nominal Composition	P-Number	Spec Min Tensile	Notes	For Metal Temperatures Not Exceeding Deg F -20 to 650	700	750	800
PLATE STEELS									
Carbon Steels									
SA-7	....	...	1	60,000	(1)(3)(19)	12,650	...	...	...
SA-36	....	...	1	58,000	(1)(3)(19)	12,650	...	...	...
SA-113	C	...	1	48,000	(1)(3)	11,050	...	...	...
SA-283	A	...	1	45,000	(1)(3)	10,350	...	...	...
SA-283	B	...	1	50,000	(1)(3)	11,500	...	...	...
SA-283	C	...	1	55,000	(1)(3)	12,650	...	...	...
SA-283	D	...	1	60,000	(1)(3)	12,650	...	...	...
SA-285	A	...	1	45,000	(2)(4)	11,250	11,000	10,250	9,000
SA-285	B	...	1	50,000	(2)(4)	12,500	12,100	11,150	9,600
SA-285	C	...	1	55,000	(2)(4)	13,750	13,250	12,050	10,200
SA-299	....	C-Mn-Si	1	75,000	...	18,750	17,700	15,650	12,600
SA-300	....	...	...	...	(13)	...	...	...	...
SA-414	A	...	1	45,000	(4)	11,250	11,000	10,250	9,000
SA-414	B	...	1	50,000	(4)	12,500	12,100	11,150	9,600
SA-414	C	...	1	55,000	(4)	13,750	13,250	12,050	10,200
SA-433	L-45	1/4 Pb	1	45,000	(25)	...	...	...	...
SA-433	L-50	1/4 Pb	1	50,000	(25)	...	...	...	...
SA-433	L-55	1/4 Pb	1	55,000	(25)	...	...	...	...
SA-433	LK-55	1/4 Pb C-Si	1	55,000	(25)	...	...	...	...
SA-433	LK-60	1/4 Pb C-Si	1	60,000	(25)	...	...	...	...
SA-433	LK-65	1/4 Pb C-Si	1	65,000	(25)	...	...	...	...
SA-433	LK-70	1/4 Pb C-Si	1	70,000	(25)	...	...	...	...
SA-442	55	C-Si	1	55,000	...	13,750	13,250	12,050	10,200
SA-442	60	C-Si	1	60,000	...	15,000	14,350	12,950	10,800
SA-455	A	...	1	75,000	(29)	18,750	...	...	...
SA-455	B	...	1	73,000	(26)	18,250	...	...	...
SA-515	55	C-Si	1	55,000	(27)	13,750	13,250	12,050	10,200
SA-515	60	C-Si	1	60,000	(27)	15,000	14,350	12,950	10,800
SA-515	65	C-Si	1	65,000	(27)	16,250	15,500	13,850	11,400
SA-515	70	C-Si	1	70,000	(27)	17,500	16,600	14,750	12,000
SA-516	55	C-Si	1	55,000	(27)	13,750	13,250	12,050	10,200
SA-516	60	C-Si	1	60,000	(27)	15,000	14,350	12,950	10,800
SA-516	65	C-Si	1	65,000	(27)	16,250	15,500	13,850	11,400
SA-516	70	C-Si	1	70,000	(27)	17,500	16,600	14,750	12,000
Low-Alloy Steels									
SA-202	A	Cr-Mn-Si	4	75,000	...	18,750	17,700	15,650	12,600
SA-202	B	Cr-Mn-Si	4	85,000	...	21,250	19,800	17,700	12,800
SA-203	A	2-1/2 Ni	9A	65,000	...	16,250	15,500	13,850	11,400
SA-203	B	2-1/2 Ni	9A	70,000	...	17,500	16,600	14,750	12,000
SA-203	D	3-1/2 Ni	9B	65,000	...	16,250	15,500	13,850	11,400
SA-203	E	3-1/2 Ni	9B	70,000	...	17,500	16,600	14,750	12,000
SA-204	A	C-1/2 Mo	3	65,000	(28)	16,250	16,250	16,250	15,650
SA-204	B	C-1/2 Mo	3	70,000	(28)	17,500	17,500	17,500	16,900
SA-204	C	C-1/2 Mo	3	75,000	(28)	18,750	18,750	18,750	18,000

APPENDIX : C

Belt width (in.)	A = Spacing for various weights of material						For troughed belt		Returns
	Lb./cu. ft.								
	30	50	75	100	125	150	B (min.)	C (min.)	D
14–16	5½	5½	5	5	4½	4½	2	1½	10
18–20	5	5	5	4½	4½	4	2½	2	10
24	5	5	4½	4½	4	4	3½	2½	10
30	5	4½	4	4	3½	3½	4½	3	10
36	4½	4½	4	4	3½	3½	5	3½	10
42	4½	4½	4	3½	3½	3½	6	4	10
48	4	4	3½	3½	3½	3	5	5	9
54	4	4	3½	3½	3	3	5½	5½	9
60	4	3½	3½	3½	3	3	6	6	8

— Recommended average spacings for belt idlers —



Maximum Size of Material (in.)			Maximum Belt Speeds in Feet per Minute				Width of Belt (in.)	Capacity in Tons (2,000 lb.) per Hour of Material Weighing 100 Lb. per Cu. Ft. Multiply by 0.30 for 30 lb./cu. ft. Material 0.50 for 50 lb./cu. ft. Material 1.50 for 150 lb./cu. ft. Material							Cubic Yards per Hour at 100 Ft./Min. Belt Speed
Coarse Lump		Small Lump, Unsized	Granular Material 1/8 to 3/4 in. (sand, coal, grain, wood chips)	Coarse-Lump Nonabrasive Material (such as coal, earth)	Coarse, Crushed, Abrasive Material (ore, slag, stone, etc.)	Heavy, Large, Abrasive Material (ore, slag, stone, etc.)		Belt Speed in Feet per Minute							
Sized	Unsized							100	200	300	400	500	600	700	
3	4		450	400	400	400	16	42	84	126	168	210			31
3	4		450	400	400	400	18	54	108	162	216	270			40
3	5		550	450	450	400	20	67	135	202	270	337			50
5	8	2	600	450	500	400	24	100	200	300	400	500	600		74
6	10	3	800	500	550	450	30	160	324	486	648	810	972	1,134	120
8	15	4	800	550	600	500	36	235	470	705	940	1,175	1,410	1,645	174
10	18	5	800	550	600	500	42	325	650	975	1,300	1,625	1,950	2,275	240
12	21	6	800	600	600	550	48	440	880	1,320	1,760	2,200	2,640	3,080	326
14	24	8	800	600	600	550	54	570	1,140	1,710	2,280	2,850	3,420	3,990	420
16	26	10	800	600	600	550	60	720	1,440	2,160	2,880	3,600	4,320	5,020	530
Average thickness of top rubber cover			1/8 in.	3/16 in.	1/4 to 5/16 in.	5/16 to 3/8 in.	For flat belts use 50% of the above capacities.								

Bottom rubber cover varies from 1/32 in. for light nonabrasive material to 1/8 in. for severe operating conditions. 1/16 in. is normal.

Suggested operating speeds:

- Minimum for average or normal operation .... 300 to 400 ft. per min.
- For heavy fine material such as cement .... 250 to 300 ft. per min.
- For light fluffy material such as soda ash, pulverized coal, or where degradation is harmful .... 200 to 250 ft. per min.
- For material such as soap chips .... 150 to 200 ft. per min.
- Where discharge plows are used .... 150 to 200 ft. per min.
- (Using 20-degree troughing idlers.)

Speeds of 300 fpm or more are suggested for better discharge of material which may tend to adhere to belt.

Maximum size pieces, maximum belt speeds, and average belt conveyor capacities.

Belt Width	Q	Horizontal Conveyor Centers in Feet											
		100	200	300	400	600	800	1,000	1,200	1,400	1,600	1,800	2,000
		Friction Factor, 0.03; Length Factor, 150 Ft.											
16	14	0.31	0.44	0.57	0.69	0.94	1.20						
18	15	0.34	0.47	0.61	0.75	1.02	1.28						
20	19	0.42	0.59	0.76	0.93	1.27	1.61						
24	23	0.53	0.74	0.95	1.16	1.59	2.00	2.43	2.86				
30	33	0.75	1.05	1.35	1.65	2.25	2.85	3.45	4.05				
36	41	0.93	1.30	1.67	2.04	2.79	3.53	4.27	5.02				
42	51	1.15	1.62	2.08	2.54	3.47	4.39	5.32	6.25				
48	63	1.42	2.00	2.56	3.13	4.27	5.41	6.55	7.69				
54	76	1.72	2.41	3.10	3.79	5.17	6.55	7.93	9.31				
60	86	1.95	2.73	3.51	4.29	5.85	7.41	8.97	10.5				
		Heavy-Duty Idlers											
48	79	1.75	2.45	3.15	3.85	5.25	6.65	8.05	9.45				
54	93	2.12	2.97	3.82	4.67	6.37	8.07	9.77	11.4				
60	102	2.30	3.22	4.14	5.06	6.90	8.74	10.5	12.4				
		Friction Factor, 0.022; Length Factor, 200 Ft.											
16	14	0.27	0.36	0.45	0.54	0.72	0.90						
18	15	0.30	0.40	0.50	0.60	0.80	1.00						
20	19	0.36	0.48	0.60	0.72	0.96	1.20						
24	23	0.45	0.60	0.75	0.90	1.20	1.50	1.80	2.10	2.40	2.70	3.00	3.30
30	33	0.66	0.88	1.10	1.32	1.76	2.20	2.64	3.08	3.52	3.96	4.40	4.84
36	41	0.82	1.09	1.36	1.64	2.18	2.73	3.27	3.82	4.36	4.90	5.46	6.00
42	51	1.02	1.36	1.70	2.04	2.72	3.40	4.08	4.76	5.44	6.12	6.80	7.48
48	63	1.26	1.68	2.10	2.52	3.36	4.20	5.04	5.88	6.72	7.56	8.40	9.24
54	76	1.50	2.00	2.50	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00	11.00
60	86	1.71	2.28	2.85	3.42	4.56	5.70	6.84	7.98	9.12	10.26	11.40	12.54
		Heavy-Duty Idlers											
48	79	1.58	2.11	2.63	3.16	4.22	5.27	6.32	7.37	8.43	9.48	10.54	11.59
54	93	1.86	2.48	3.10	3.72	4.96	6.20	7.44	8.68	9.92	11.16	12.40	13.64
60	102	2.04	2.72	3.40	4.08	5.44	6.80	8.16	9.52	10.88	11.24	13.60	14.96
		100	200	300	400	600	800	1,000	1,200	1,400	1,600	1,800	2,000

$$\text{Hp.} = \frac{CQ(L + L_o)S}{33{,}000} \quad \text{or} \quad \frac{C(L + L_o)\,(0.03\,QS)}{990}$$

Multiply values from this table by $\dfrac{\text{Belt speed in ft. per min.}}{100}$

EXAMPLE: 24-in. belt, 800-ft. ctrs., 350 ft. per min. Friction factor, 0.03; length factor, 150 ft.

$$\text{Hp.} = 2.00 \times \frac{350}{100} = 7$$

Empty-belt horsepower.

C	L_o	Horizontal Conveyor Centers in Feet											
		100	200	300	400	600	800	1,000	1,200	1,400	1,600	1,800	2,000
0.03	150	0.76	1.06	1.36	1.66	2.27	2.88	3.48	4.09				
0.022	200	0.66	0.88	1.10	1.32	1.76	2.20	2.64	3.08	3.52	3.96	4.40	4.84

$$\text{Hp.} = \frac{C(L + L_o)T}{990}$$

Multiply values from this table by $\frac{\text{Tons per hr.}}{100}$

EXAMPLE: Friction factor, 0.03; length factor, 150 ft. Convey 350 tons per hour 800 ft.

$$\text{Hp.} = 2.88 \times \frac{350}{100} = 2.88 \times 3.5 = 10.88$$

Horsepower to convey material horizontally.

Horsepower per 100 Tons per Hour To Raise or Lower

Vertical Height in Feet											
5	10	15	20	30	40	50	60	70	80	90	100
0.51	1.01	1.52	2.02	3.03	4.04	5.05	6.06	7.07	8.08	9.09	10.10

$$\text{Hp.} = \frac{TH}{990}$$

T = Tons per hour (1 ton is 2,000 lb.)
H = Vertical height in feet

Multiply values from this table by $\frac{\text{Tons per hr.}}{100}$

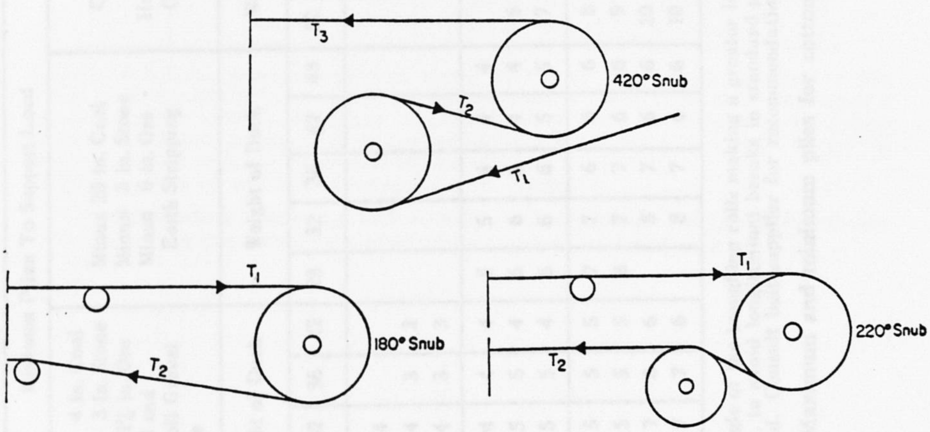
EXAMPLE: Elevate 350 tons per hour through 50 ft.
Hp. = 5.05 × 350/100 = 5.05 × 3.5 = 17.66.

Elevate 175 tons per hour through 25 ft.

Hp. for 100 TPH through 20 ft.	2.02
Hp. for 100 TPH through 5 ft.	0.51
Hp. for 100 TPH through 25 ft.	2.53
Hp. for 175 TPH through 25 ft.: 2.53 × 175/100	4.43

Horsepower due to vertical height (lifting or lowering.)

Degree of Belt Contact	Type of Drive	Operating Belt Tension, T_1		T_2 for Single-Pulley Drive T_3 for Dual-Pulley Drive		T_1/T_2 for Single-Pulley Drive T_1/T_3 for Dual-Pulley Drive	
		Bare Pulley	Lagged Pulley	Bare Pulley	Lagged Pulley	Bare Pulley	Lagged Pulley
180	Plain	1.85E	1.50E	0.85E	0.50E	2.19	3.00
200	Snubbed	1.72E	1.42E	0.72E	0.42E	2.39	3.39
210	Snubbed	1.67E	1.38E	0.67E	0.38E	2.50	3.61
215	Snubbed	1.64E	1.36E	0.64E	0.36E	2.55	3.72
220	Snubbed	1.62E	1.35E	0.62E	0.35E	2.61	3.83
240	Snubbed	1.54E	1.30E	0.54E	0.30E	2.85	4.33
360	Tandem	1.26E	1.13E	0.26E	0.13E	4.80	9.02
380	Tandem	1.23E	1.11E	0.23E	0.11E	5.25	10.19
400	Tandem	1.21E	1.09E	0.21E	0.09E	5.72	11.51
420	Tandem	1.19E	1.08E	0.19E	0.08E	6.25	13.00
450	Tandem	1.16E	1.07E	0.16E	0.07E	7.12	15.27
500	Tandem	1.13E	1.05E	0.13E	0.05E	8.86	21.21



The above values are based on a coefficient of friction between belt and pulley of 0.25 for bare iron or steel pulleys and 0.35 for rubber lagged pulleys.

IMPORTANT NOTE: The ratio T_1/T_2 or T_1/T_3 is very important in determining whether the belt will slip on the drive pulley. The ratio given in the last two columns should never be exceeded for operating belt tensions; for starting tensions only these ratios can be increased 50 percent.

Design data for belt conveyor drives.

Width of Belt (in.)	Minimum Plies To Support Load															Maximum Plies for Troughing									
	Light Materials, (grain, wood, chips)		Minus 4 in. Coal Minus 3 in. Stone Minus 1½ in. Ore Sand and Small Gravel Coke				Minus 20 in. Coal Minus 8 in. Stone Minus 6 in. Ore Earth Stripping					Coarse Ores or Other Heavy Material (large lumps)				20° Troughing Idlers					45° Troughing Idlers*				
	Weight of Duck		Weight of Duck				Weight of Duck					Weight of Duck				Weight of Duck					Weight of Duck				
	28	32	28	32	36	42	28	32	36	42	48	32	36	42	48	28	32	36	42	48	28	32	36	42	48
16	3	3	4	4												4	4								
18	3	3	4	4	3	3										4	4	3	3						
20	4	4	4	4	3	3										5	5	4	3		3				
24	4	4	4	4	4	4	5	5	4	4	4					5	5	5	4	4	4				
30	4	4	5	5	5	4	6	6	5	4	4	6	6	5	5	6	6	6	5	5	5	4			
36	4	4	6	5	5	4	6	6	6	5	5	7	7	6	6	8	8	7	6	6	5	4	4		
42	4	4	6	5	5	5	7	7	6	6	6	8	7	6	6	9	9	8	8	7	6	5	5	4	4
48	4	4	6	5	5	5	8	7	7	6	6	9	8	7	7	10	10	10	9	9	6	6	6	5	5
54				7	6	6		8	7	6	6	10	9	8	8		12	12	11	11					
60				7	7	6		8	7	6	6	10	9	8	8		13	13	12	12					

* Because of the steeper angle of the troughing rolls making a greater bend in the belt to form the deeper trough, thinner belts are necessary for 45° troughing idlers, to avoid longitudinal breaks in standard ply-constructed belts. Where greater strength is required, cord belts or special weaves may be used. Consult belt supplier for recommendations.

Maximum and minimum plies for cotton-fabric ply-constructed rubber conveyor belt.

Permissible Operating Tensions (lb. per in. per ply)			Weight of Fabric	Minimum Pulley Diameters in Inches										Percent Permissible Operating Tension
Type of Belt Splice														
Metal	Vulcanized 2L/S*			Number of Plies in Belt										
	Less than 2	More than 3		3	4	5	6	7	8	9	10	11	12	
26	30	35	28 oz.	18	24	30	36	42	48	48	54	60	66	100
			and	16	20	24	30	36	42	42	48	54	60	80
				12	16	20	24	30	30	36	36	42	48	60
30	35	40	32 oz.	12	16	16	20	24	24	30	30	36	36	40
35	40	45	36 oz.	20	30	36	42	48	54	60	60	66	72	100
			and	18	24	30	36	42	42	48	54	54	60	80
				12	16	20	24	30	36	36	42	48	48	60
40	50	55	42 oz.	12	16	18	20	24	30	30	36	36	42	40
†	60	70	48 oz.		30	36	42	48	60	66	72	72	84	100
					24	30	36	36	48	54	60	60	66	80
					20	24	24	30	36	42	42	48	54	60
					20	20	20	24	30	30	36	36	42	40

* $2L/S = 2 \times$ conveyor centers/Belt speed (F.P.M.); for 450 ft. centers at 300 F.P.M. $2\ L/S = 900/300 = 3$. The 100 percent tension for a 7-ply 32-oz. belt with a vulcanized splice would be 40 lb. per in. per ply, and a pulley 42 in. in diameter should be used. If the belt tension is 80 percent of the maximum permissible, or 32 lb. per in. per ply, a pulley 36 in. in diameter could be used.

† Metal splices are not recommended for 48-oz. fabric.

— Permissible operating belt tensions and minimum pulley diameters for cotton-fabric ply-constructed rubber conveyor belts.

Weight of Fabric	Thick-ness per Ply (In.)	Total Thick-ness of BOTH Rubber Covers (In.)	Weight in Pounds per Inch of Width per Foot of Length — Number of Plies									
			3	4	5	6	7	8	9	10	11	12
28 oz.	0.045	3/16	0.176	0.201	0.225	0.250	0.274	0.299	0.324	0.348		
		1/4	0.210	0.235	0.259	0.284	0.309	0.333	0.358	0.382		
		5/16	0.244	0.269	0.293	0.318	0.343	0.367	0.392	0.416		
32 oz.	0.053	3/16	0.184	0.211	0.238	0.265	0.292	0.319	0.346	0.373	0.400	
		1/4	0.218	0.245	0.272	0.299	0.326	0.353	0.380	0.407	0.434	
		5/16	0.252	0.279	0.306	0.333	0.360	0.387	0.414	0.441	0.468	
36 oz.	0.056	3/16	0.188	0.216	0.245	0.273	0.302	0.330	0.359	0.387	0.416	0.444
		1/4	0.222	0.250	0.279	0.307	0.336	0.364	0.393	0.421	0.450	0.478
		5/16	0.256	0.284	0.313	0.341	0.370	0.398	0.427	0.455	0.484	0.512
		3/8	0.290	0.318	0.347	0.375	0.404	0.432	0.461	0.489	0.518	0.546
42 oz.	0.063	3/16	0.196	0.227	0.259	0.290	0.321	0.353	0.384	0.415	0.447	0.478
		1/4	0.230	0.261	0.293	0.324	0.355	0.387	0.418	0.449	0.481	0.512
		5/16	0.264	0.295	0.327	0.358	0.389	0.421	0.452	0.483	0:515	0.546
		3/8	0.298	0.329	0.361	0.392	0.423	0.455	0.486	0.517	0.549	0.580
48 oz.	0.069	3/16		0.238	0.272	0.306	0.340	0.374	0.408	0.442	0.476	0.510
		1/4		0.272	0.306	0.340	0.374	0.408	0.442	0.476	0.510	0.544
		5/16		0.306	0.340	0.374	0.408	0.442	0.476	0.510	0.544	0.578
		3/8		0.340	0.374	0.408	0.442	0.476	0.510	0.544	0.578	0.612

EXAMPLE: Find weight per ft. and thickness of belt 36 in. wide, 6 ply, 48-oz. duck, with a 1/4-in. top and a 1/8-in. bottom rubber cover.

Total thickness of covers: $\frac{1}{4} + \frac{1}{8}$ in. $= \frac{3}{8}$ in.

Total weight per ft.: $36 \times 0.408 = 14.68$ lb. per ft. of belt.

Total thickness of belt: $6 \times 0.069 + (\frac{1}{4} + \frac{1}{8}) = 0.789$ in.

For weight of belting having other thickness of rubber cover, allow 0.017 lb. for each 1/32-in. thickness of cover.

Weight of cotton-fabric ply-constructed rubber conveyor belt.

APPENDIX : D

Saturation—Vacuum System

Material	Wt per cu ft	100 ft Sat.	100 ft hp/T	150 ft Sat.	150 ft hp/T	250 ft Sat.	250 ft hp/T	400 ft Sat.	400 ft hp/T	Velocity ft/sec
Alum	50	3.6	4.5	3.9	5.0	4.3	5.7	4.7	6.3	110
Alumina	60	2.4	4.0	2.8	4.7	3.4	5.7	4.0	6.4	105
Carbonate, calcium	25–30	3.1	4.2	3.6	5.0	3.9	5.5	4.2	6.0	110
Cellulose acetate	22	3.2	4.7	3.5	5.1	3.8	5.7	4.1	6.0	100
Clay, air floated	30	3.3	4.5	3.5	5.0	3.9	5.5	4.2	6.0	105
Clay, water washed	40–50	3.5	5.0	3.8	5.6	4.2	6.5	4.5	7.2	115
Clay, spray dried	60	3.4	4.7	3.6	5.2	4.0	6.2	4.4	7.1	110
Coffee beans	42	1.2	2.0	1.6	3.0	2.1	3.5	2.4	4.2	75
Corn, shelled	45	1.9	2.5	2.1	2.9	2.4	3.6	2.8	4.3	105
Flour, wheat	40	1.5	3.0	1.7	3.3	2.0	3.7	2.5	4.4	90
Grits, corn	33	1.7	2.5	2.2	3.0	2.9	4.0	3.5	4.8	100
Lime, pebble	56	2.8	3.8	3.0	4.0	3.4	4.7	3.9	5.4	105
Lime, hydrated	30	2.1	3.3	2.4	3.9	2.8	4.7	3.4	6.0	90
Malt	28	1.8	2.5	2.0	2.8	2.3	3.4	2.8	4.2	100
Oats	25	2.3	3.0	2.6	3.5	3.0	4.4	3.4	5.2	100
Phosphate, trisodium	65	3.1	4.2	3.6	5.0	3.9	5.5	4.2	6.0	110
Polyethylene pellets	30	1.2	2.0	1.6	3.0	2.1	3.5	2.4	4.2	80
Rubber pellets	40	2.9	4.2	3.5	5.0	4.0	6.0	4.5	7.2	110
Salt cake	90	4.0	6.5	4.2	6.8	4.6	7.5	5.0	8.5	120
Soda ash, light	35	3.1	4.2	3.6	5.0	3.9	5.5	4.2	6.0	110
Soft feeds	20–40	3.0	4.2	3.4	4.5	3.7	5.0	4.2	5.5	110
Starch, pulverized	40	1.7	3.0	2.0	3.4	2.6	4.0	3.4	5.0	90
Sugar, granulated	50	3.0	3.7	3.2	4.0	3.4	5.2	3.9	6.0	110
Wheat	48	1.9	2.5	2.1	2.9	2.4	3.6	2.8	4.3	105
Wood flour	12–20	2.5	3.5	2.8	4.0	3.4	4.9	4.4	6.5	100

NOTE: The above saturation figures are for 4–, 5–, and 6–in. ID conveying pipes. For larger pipes, use slightly lower saturations and hp/ton. For 6–in conveying pipe, saturations and hp/ton can be reduced upto 15% while for 10–in and 12–in. pipes upto 25% and 35% respectively.

For conveying distances longer than 400 ft., saturations and hp/ton must be increased but on a sliding scale. For a 550–ft conveying distance, increase saturation factor for 400 ft by 17%, a 700 ft conveying distance by 30%. For 850 and 1000 feet conveying distances by 41% and 50% respectively.

Pipe Constants

IPS Pipe Size	Schedule 5	10	30	40
	Pipe Constant			
3 in.	3.6	3.5		3.07
3½ in.	4.8	4.6		4.05
4 in.	6.1	5.9		5.3
5 in.	9.4	9.2		8.4
6 in.	13.5	13.2		12.0
7 in.				16.0
8 in.	23.2	22.7	21.3	
10 in.			34.0	
12 in.			47.8	

Vacuum Slippage Factor for Positive-Pressure Blower

Vacuum (in. Hg)	Equivalent Pressure (psig)	F
6	3	1.105
7	3.5	1.125
8	4	1.145
9	4.5	1.168
10	5	1.190
11	5.5	1.216
12	6	1.241

Atmospheric Pressure and Correction Factor at Various Altitudes

Altitude (feet above sea level)	P. Absolute Pressure psi	H. Absolute Pressure (in. Hg)	R. Correction Factor at that Altitude
0	14.69	29.92	1.00
1,000	14.16	28.86	1.02
2,000	13.66	27.82	1.04
3,000	13.16	26.81	1.055
4,000	12.68	25.84	1.08
5,000	12.22	24.89	1.095
6,000	11.7[illegible]	23.98	1.12
7,000	11.33	23.09	1.14
8,000	10.91	22.22	1.16
9,000	10.50	21.38	1.18
10,000	10.10	20.58	1.20
11,000	9.71	19.75	1.23
12,000	9.34	19.03	1.25
13,000	8.97	18.29	1.28
14,000	8.62	17.57	1.30
15,000	8.28	16.88	1.33

APPENDIX : E

Total land area: 3,600,000 sq. ft.

Existing built-up area : 534,058 sq. ft.

Building area of Tetra Division: 40,295 sq. ft.

Total built-up area: 574, 353 sq. ft.

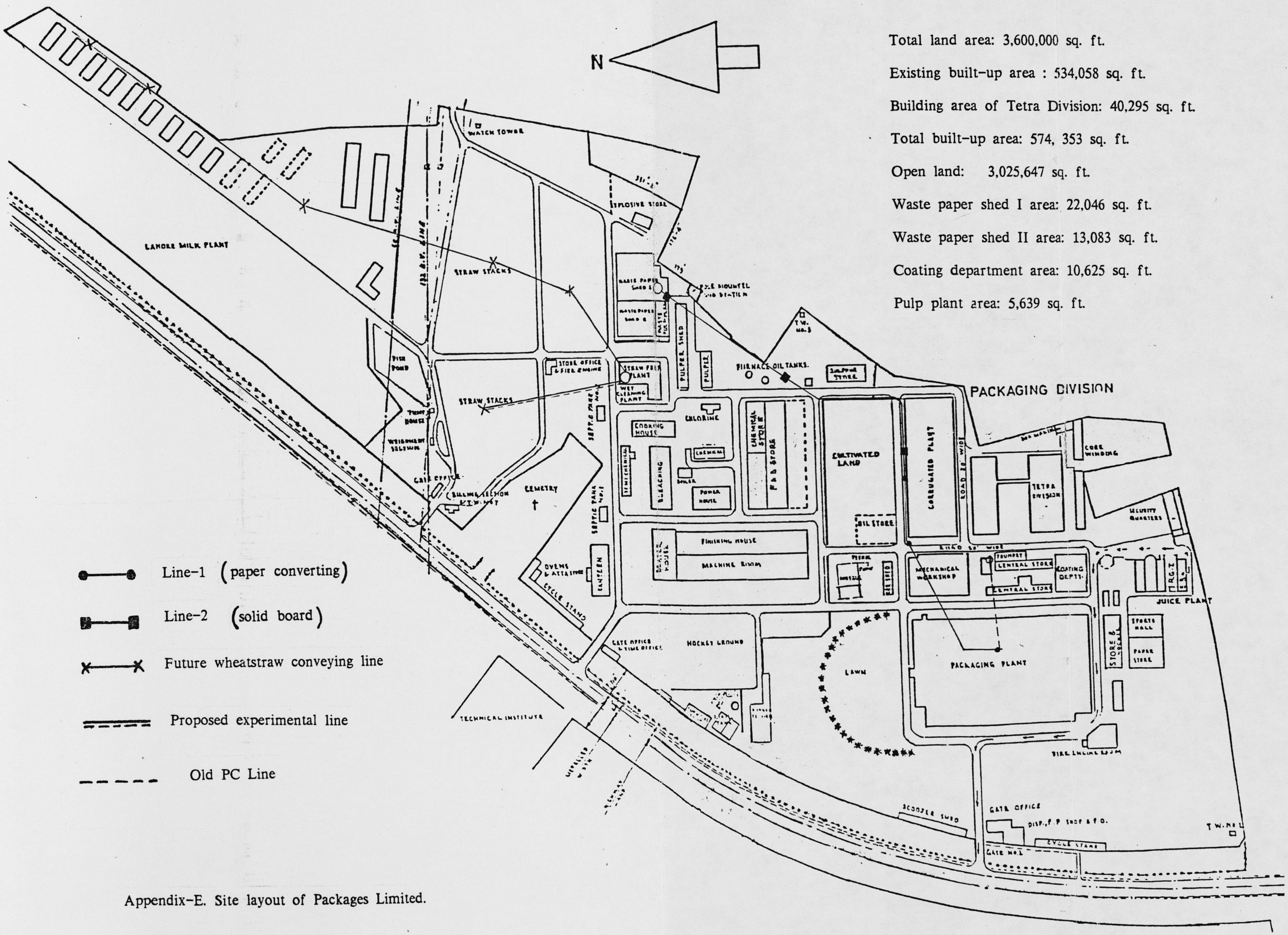
Open land: 3,025,647 sq. ft.

Waste paper shed I area: 22,046 sq. ft.

Waste paper shed II area: 13,083 sq. ft.

Coating department area: 10,625 sq. ft.

Pulp plant area: 5,639 sq. ft.



Appendix-E. Site layout of Packages Limited.