

# Design of Conveyor Drums for an Iron Ore Terminal

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**ABSTRACT:** Materials are evolving today faster than at any time in history. Industrial nations regard the development of new and improved materials as an "underpinning technology" one which can stimulate innovation in all branches of engineering, making possible new designs for structures, appliances, engines, electrical and electronic devices, processing and energy conservation equipment, and much more. Many of these nations have promoted government-backed initiatives to promote the development and exploitation of new materials: their lists generally include "high-performance" composites, new engineering ceramics, high-strength polymers, glassy metals, and new high-temperature alloys for gas turbines. These initiatives are now being felt throughout engineering, and have already stimulated design of a new and innovative range of consumer products.

**KEYWORDS:** Polymer, Conveyor, Tensile stress, Compressive stress

## I. INTRODUCTION

The engineer must be more aware of materials and their potential than ever before. Innovation, often, takes the form of replacing a component made of one material (a metal, say) with one made of another (a polymer, perhaps), and then redesigning the product to exploit, to the maximum, the potential offered by the change. The engineer must compare and weigh the properties of competing materials with precision: the balance, often, is a delicate one<sup>[1]</sup>. It involves an understanding of the basic properties of materials; of how these are controlled by processing; of how materials are formed, joined. The conveyor belt is one of the most efficient devices available for moving goods over short distances. Billions of tons of minerals, foodstuffs and consumer goods are handled in this way every year<sup>[3]</sup>. Fig.1 shows the essentials of a typical conveyor system. The following data are typical of the largest conveyors, which are used for handling coal, iron ore and other heavy minerals<sup>[6]</sup>.

|              |   |                            |
|--------------|---|----------------------------|
| Capacity     | : | 5000 tonne h <sup>-1</sup> |
| Belt speed   | : | 4 m s <sup>-1</sup>        |
| Belt tension | : | 5 tonne                    |
| Motor rating | : | 250 kW                     |
| Belt section | : | 15m wide×11mm thick        |

Distance between centres of tail drum and drive drum: 200 m

It is important that conveyor systems of this size are designed to operate continuously for long periods with minimum "down-time" for routine maintenance: the unscheduled breakdown of a single unit in an integrated plant could lead to a total loss of production<sup>[5]</sup>. Large conveyors include a number of critical components which are designed and built essentially as "one-offs" for a particular installation: it is doubly important to check these at the design stage because a failure here could lead to a damagingly long down-time while a harassed technical manager phones the length of the country looking for fabrication shops with manoeuvrable capacity, and steel merchants with the right sections in stock<sup>[4]</sup>.

## II. TAIL DRUM DESIGN

The tail drum (Fig.1) is a good example of a critical component. Fig.2 shows the general arrangement of the drum in its working environment and Fig.3 shows a detailed design proposal. We begin our design check by looking at the stresses in the shaft. The maximum stress comes at the surface rollers it must be kept under a constant tension T. This is done by hanging a large weight on the tension drum. The drive is supplied by coupling a large electric motor to the shaft of the drive drum via a suitable gearbox and overload clutch of the shaft next to the shaft-plate weld (Fig.4).

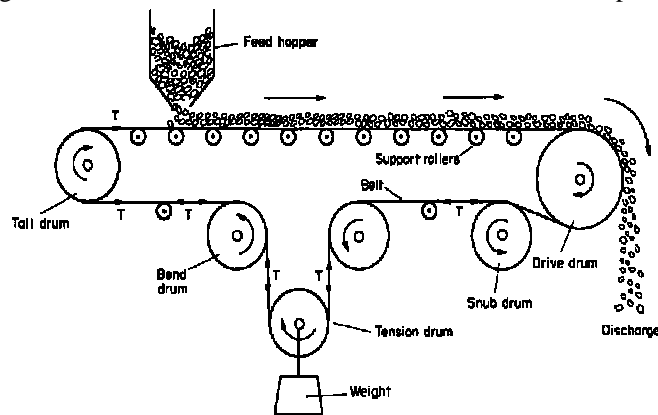


Fig.1 Schematic of a typical conveyor system. Because the belt tends to sag between the support

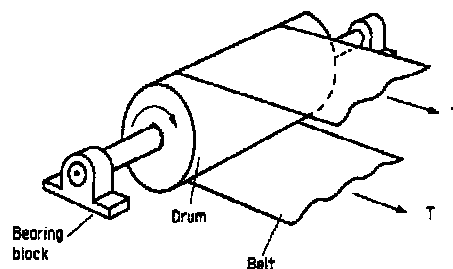


Fig.2 Close-up of the tail drum. The belt tension applies a uniformly distributed sideways loading to the drum

We can calculate the maximum stress from the standard formula  $[Mc]_{\max} = I$

Where the bending moment M is given by  $M = F_x$  and the second moment of area of the shaft is given by

$$I = c^4 / 4$$

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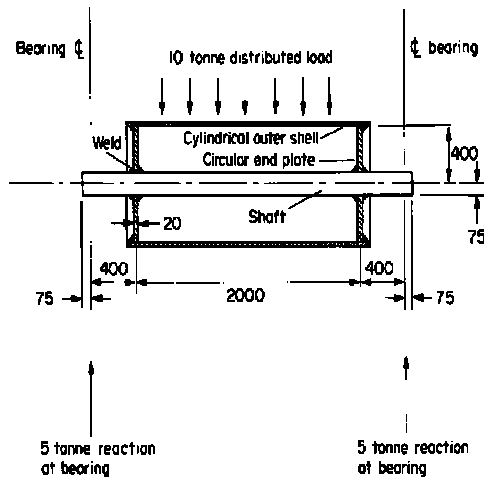


Fig.3 Cross-section through the tail drum.

All dimensions are in mm. We have assumed a belt tension of 5 tonnes, giving a total loading of 10 tonnes

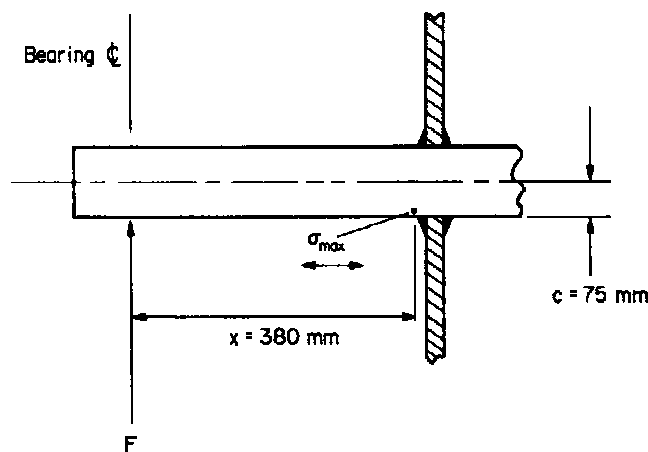


Fig.4 Shaft-plate detail

Using values of  $F = 5000 \times 981 \text{N} \times 380 \text{mm}$ , and  $c = 75 \text{mm}$ , we get a value for  $\sigma_{\max}$  of 56 MPa. This stress is only a quarter of the yield stress of a typical structural steel, and the shaft therefore has an ample factor of safety against failure by plastic overload. The second failure mode to consider is fatigue. The drum will revolve about once every second, and each part of the shaft surface will go alternately into tension and compression. The maximum fatigue stress range of  $2 \times 56 = 112 \text{MPa}$  is, however, only a quarter of the fatigue limit for structural steel (Fig.5); and the shaft should therefore last indefinitely. But what about the welds? There are in fact a number of reasons for expecting them to have fatigue properties that are poorer than those of the parent steel (see Table 1).

Fig.6 shows the fatigue properties of structural steel welds. The fatigue limit stress range of 120 MPa for the best class of weld is a good deal less than the limiting range of 440 MPa for the parent steel (Fig.5). And the worst class of weld has a limiting range of only 32 MPa. The shaft-plate weld can be identified as a class E/F weld with a limiting stress

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range of 69 to 55 MPa. This is a good deal less than the stress range of 112 MPa experienced by the shaft. We thus have the curious situation where a weld which is merely an attachment to the shaft has weakened it so much that it will only last for about  $2 \times 10^6$  cycles – or 1 month of operation. The obvious way of solving this problem is to remove the attachment weld from the surface of the shaft. Fig.7 shows one way of doing this. We have seen that welds can be very weak in a fatigue situation, and we would be wise to check that the modified weld in Fig.7 is adequate for the job.

| Characteristic   | Comments   |
|--|--|
| Change in section at weld bead   | Gives stress concentration. In the case of butt welds this can be removed by grinding back the weld until flush with the parent plates. Grinding marks must be parallel to loading direction otherwise they can initiate fatigue cracks. |
| Poor surface finish of weld bead.  | Helps initiate fatigue cracks. Improve finish by grinding  |
| Contain tensile residual stresses which are usually as large as the yield stress           | Weld liable to fatigue even when applied stress cycle is wholly compressive. Reduce residual stresses by stress relieving, hammering or shot peening.  |
| Often contain defects (hydrogen cracks, slag inclusions, stop-start marks).                | Help initiate fatigue cracks. Critical welds must be tested non-destructively and defects must be gouged out.  |
| Large differences in microstructure between parent metal, heat-affected zone and weld bead | Sharp changes in mechanical properties give local stress concentrations  |

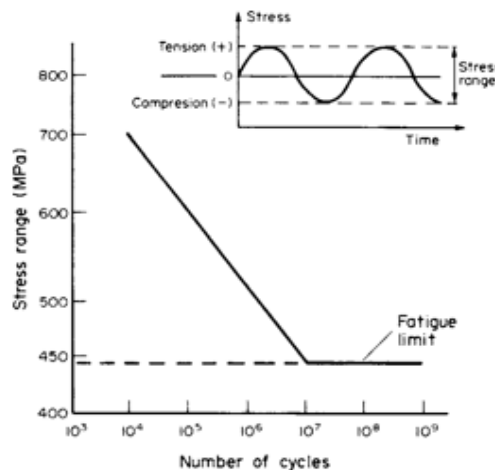


Fig.5 Fatigue data for a typical structural steel in dry air.

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Note that if the fatigue stress range is less than 440 MPa (the fatigue limit) the component should last indefinitely. The data relate to a fatigue stress cycle with a zero mean stress, which is what we have in the case of our tail drum.

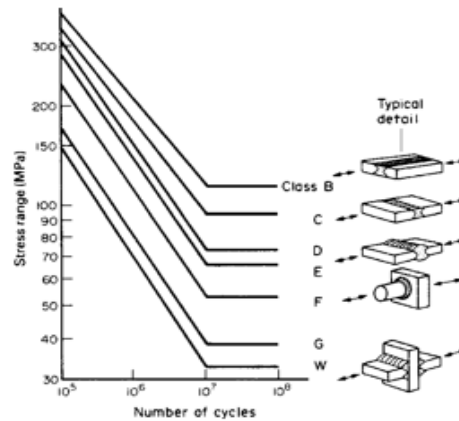


Fig.6 Fatigue data for welded joints in clean air.

The class given to a weld depends critically on the weld detail and the loading direction. Classes B and C must be free from cracks and must be ground flush with the surface to remove stress concentrations. These conditions are rarely met in practice, and most welds used in general construction have comparatively poor fatigue properties.

Table.1 Weld characteristics giving adverse fatigue properties. Although it is difficult to spot immediately, the weld is in fact subjected to large bending stresses (Fig.8). These can be calculated as follows. We begin by equating moments in Fig.8 to give

$$F \times 4m = M_A + M_B$$

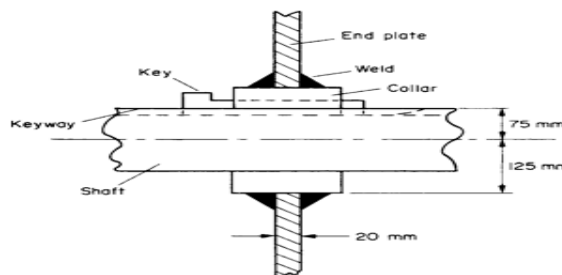


Fig.7 Modification to remove attachment weld from surface of shaft.

The collar can be pressed on to the shaft and secured with a feather key, but we must remember that the keyway will weaken the shaft.

Although it is difficult to spot immediately, the weld is in fact subjected to large bending stresses (Fig.8). These can be calculated as follows. We begin by equating moments in Fig.8 to give

$$F \times 4m = M_A + M_B$$

Here  $M_A$  is the couple which must be applied at the mid span of the shaft to make  $dy/dx=0 = 0$ ; and  $M_B$  is the couple that is needed to cause the out-of-plane deflection of the end discs. Using the standard beam-bending formula of  $D^2y/dx^2 = M / EI$

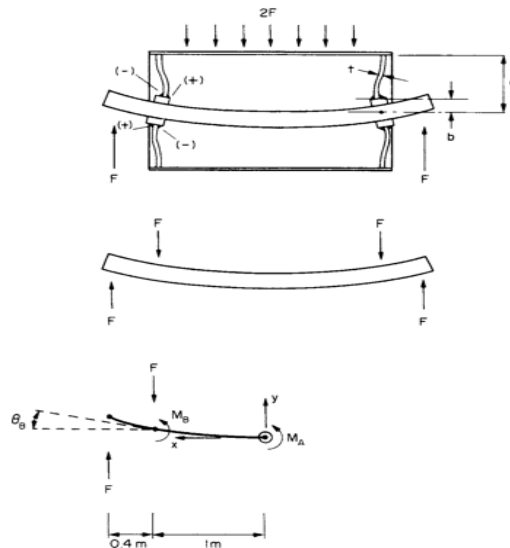


Fig.8 Exaggerated drawing of the deflections that occur in the loaded drum.

The shaft deflects under four-point loading. This in turn causes the end plates to deflect out of plane, creating tensile + and compressive – stresses in the weld.

We can now turn our attention to the end plate. The standard formulae are

$$[M^B]_{\max} = at^2 \quad (1)$$

$$[M^B]_B = Et^3 \quad (2)$$

where  $\sigma_{\max}$  is the maximum stress at the inner circumference of the plate,  $a$  is the outer radius of the plate and  $t$  is the plate thickness and  $b/a$  are dimensionless constants whose value depends on the ratio of the inner to the outer radius,  $b/a$ . This stress gives a fatigue range of 284 MPa which is about five times the limiting range for an E/F weld. This is obviously a case of going back to the drawing board! In fact, as Table 2 shows, the solution to this design problem is far from straightforward.

### III. RESULTS AND DISCUSSIONS

Table 2: Design modifications to overcome over-stressing of collar-plate weld

| Modification  | Comments   |
|---|--|
| Make end plate thicker (increase $t$ )                    | Plate would have to be 70 mm thick: heavy; difficult to weld to collar; might overstress plate-shell weld; collar would become irrelevant. |
| Make end plate thinner (decrease $t$ )                    | Plate would have to be 2 mm thick: too thin to carry normal loading without risk of buckling.  |
| Increase shaft diameter and decrease end plate thickness. | (Increase $t$ & (decrease $t$ ) Would need 250 mm diameter shaft + 15 mm thick end plate: very heavy and expensive. Would still            |

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|  |   |
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|  | need to check plate-shell weld.   |
| Remove weld altogether. Make plate a running fit on shaft and drive it using splines.        | Allows shaft to flex without bending plate. Continual flexing will wear the contacting surfaces and may lead to fretting fatigue. |
| Make shaft stiffer by supporting it with intermediate plates welded to both shaft and shell. | Alignment and assembly problems and higher cost.  |
| Decrease shaft overhang – move bearings next to end plates.                                  | Requires major modifications to support framing or over-long shells.  |
| Use fixed (non-rotating) shaft and mount bearings in end plates.                             | Probably the best solution, but no good for drive drums.  |

**IV. CONCLUSION**

Structural steelwork from small items like our conveyor drums to large structures like ships and bridges is usually fabricated by welding. It is relatively easy to design structures so that the parent material has an adequate resistance to elastic deformation, plastic collapse, fast fracture or fatigue. But the welds with their stress concentrations, residual stresses, microstructural variations and hidden defects are the Achilles’ heel of any design. It has long been known that welds can be potent initiators for fast fracture. But it is less commonly known that most welds have abysmal fatigue strengths. Designers beware.

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