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Backstop Selection Procedure for Multiple and Single Drive Conveyors

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BACKSTOP SELECTION PROCEDURE FOR MULTIPLE AND SINGLE DRIVE CONVEYORS

SYNOPSIS

The safety of inclined conveyor systems relies heavily on the proper selection of backstops. This paper provides an analysis for such systems illustrating the function of the backstop(s) during start-up, steady operation and stopping phases.

A detailed procedure is given for the selection of the appropriate sized backstop in the design stages based on the specifications of the conveyor system. Emphasis is on backstop size selection for load sharing in a multiple drive system for single belt conveyors. The reliability of the selection procedure is confirmed by field test load measurements on a multiple drive conveyor system.

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INTRODUCTION

The development of the belt conveyor system for moving bulk materials has resulted in capacity increases which require constant re-assessement of system components. One of the more serious considerations of the conveyor systems, particularly the inclined loaded conveyor, is the elimination of belt roll back when forward power is removed deliberately or accidentally. Whereas a friction brake mounted on the drive pulley would have been adequate to hold back a loaded belt on a small conveyor, the large conveyor system requires an instantaneous hold back response with a device of reliable holding capacity. Reliability of the hold back device (hereafter referred to as a backstop) is paramount when the potential for catastrophy is apparent. Personnel injury, torn belts, mangled conveyor support structures and material spill at transfer points are possible when an uncontrolled roll back occurs.

The backstop should be considered part of the conveyor system, subject to stresses and deflections generally not considered by current backstop selection methods. This paper will review not only the existing methods, but also will introduce selection considerations not typically discussed by the backstop manufacturer and the potential user.

BACKSTOP APPLICATION

Typical backstop installations on belt conveyor systems are shown in Figs. 1 and 2. Fig. 1 shows the backstop mounted on the belt pulley shaft. Fig. 2 shows the backstop mounted on the outer end of a double extended low speed shaft of the speed reducer driving the belt pulley.

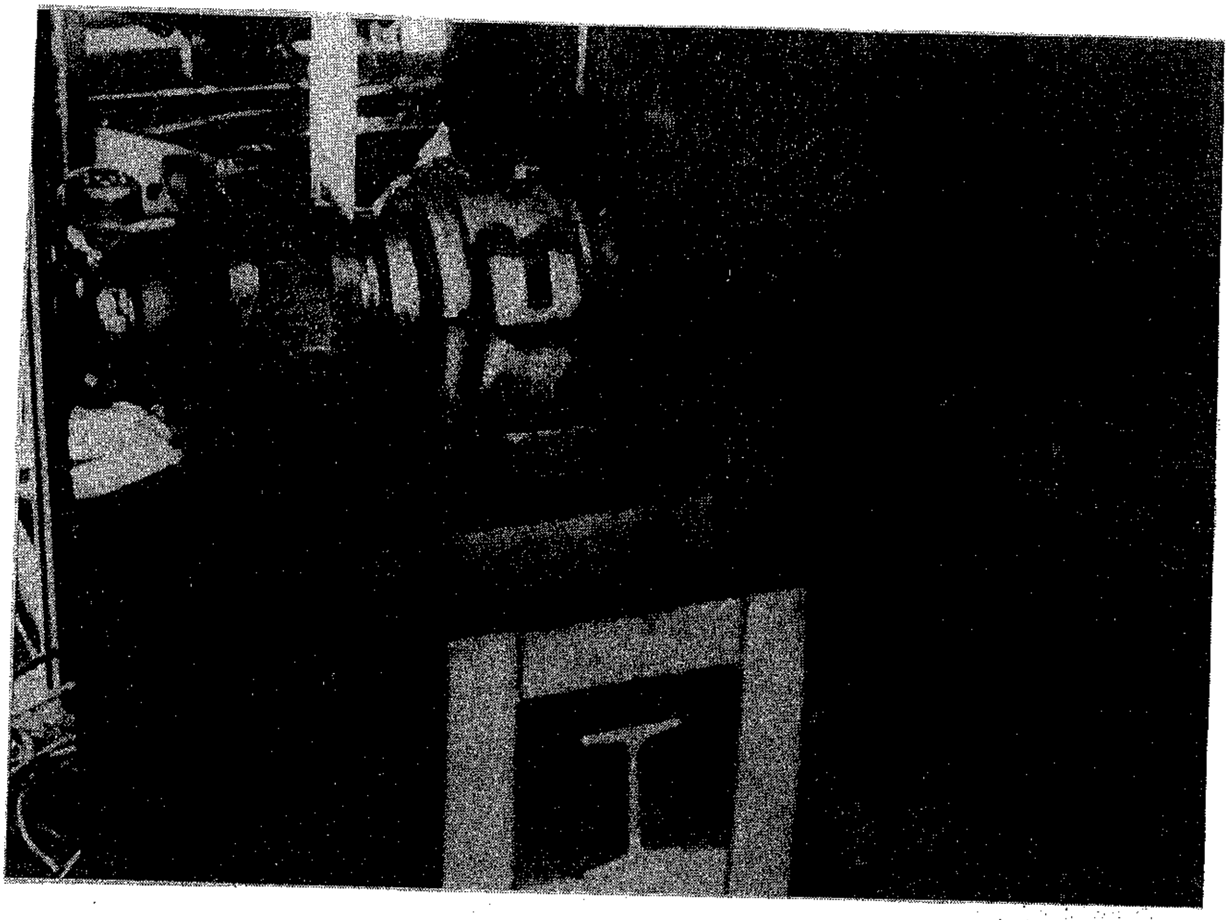

FIGURE 1

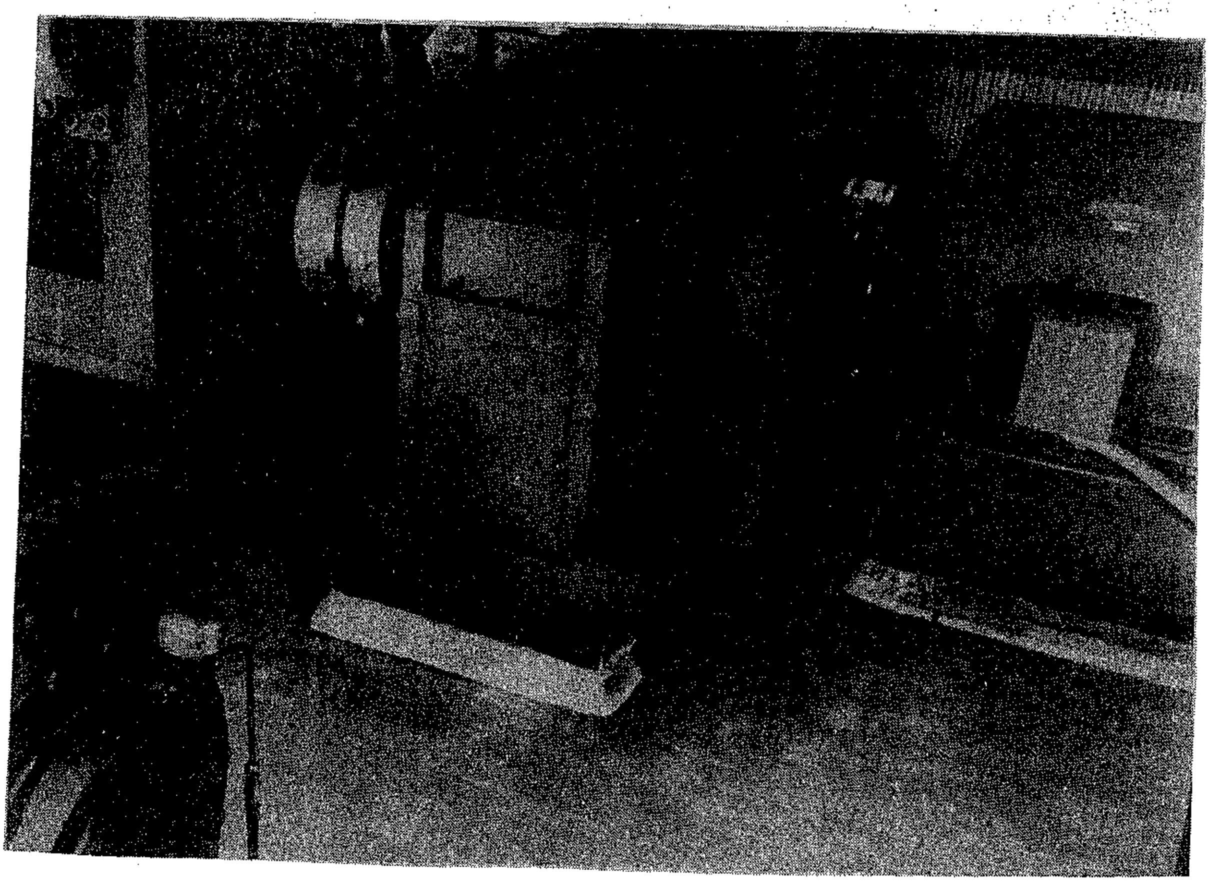

FIGURE 2

BACKSTOP DESIGN

There are several types of backstops available. The principle ones are the friction and interference (cog) types. The cog type backstop does not lend itself to load sharing analysis which is the main feature of this paper.

An example of the friction type backstop is shown in Fig. #3.

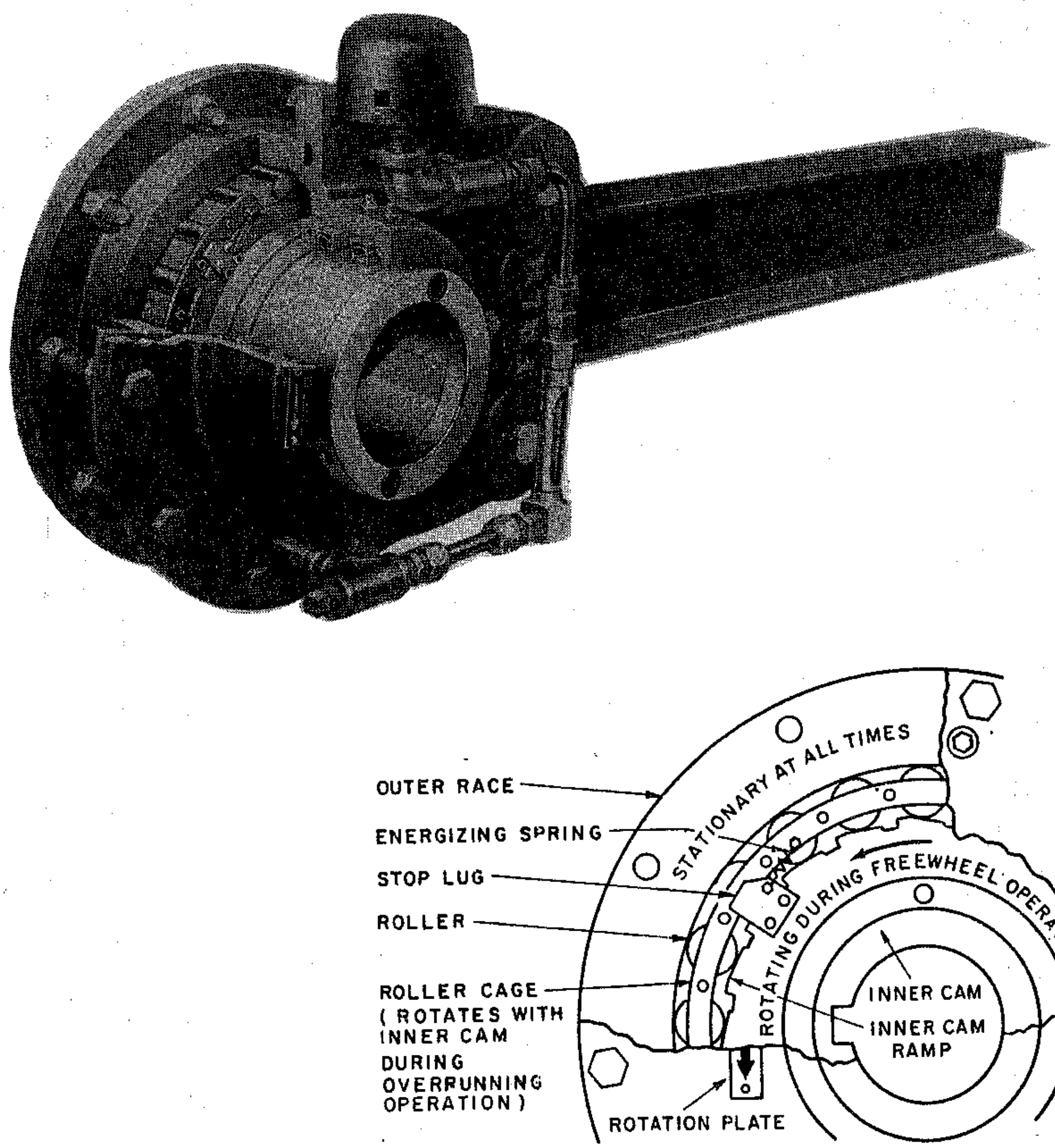

FIGURE 3

This type of backstop operates by wedging the rollers between the inner cam ramps and the outer race when in the backstop mode. The energizing springs are under tension at all times even when the backstop is free wheeling. The spring force keeps the rollers in constant contact with the ramp and outer race surfaces for instantaneous response when free wheeling ceases and backstopping commences. The backstop manufacturer is expected to provide prudent engineering judgement in the selection of materials, specifications for allowable stresses, strains, deflections and manufacturing accuracy. All element functions must interact smoothly whether in the free wheeling or backstop mode. Since most backstop failures occur when overrunning, the fits and clearances must be carefully analyzed for all conditions of operation. The interaction of the loaded and deflected backstop components is of primary importance when considering the sharing of load between rollers. The simultaneous engagement of all the rollers in the backstop design shown on Fig. #3 is probably not attainable in practice even with the most stringent requirements for manufacturing accuracy. However, machining of parts to nominal tolerances is acceptable because as the backstop load cycle develops, stressed and deflected parts make elastic accommodation to one another which diminish stress. Example: The ratio of the first roller load to the last roller load at the start of engagement may be very high, however, with increasing load the difference declines rapidly so that at approximately 20% of rating the difference is negligible. Fig. #4 shows the point at which rate of elastic deflection and the rate of elastic plus manufacturing tolerance deflection (inelastic deflection) are the same on a torsional deflection test of an individual backstop. The importance of this elastic deflection will become apparent later. In any case, it is evident that manufacturing accuracy significantly affects the backstop response.

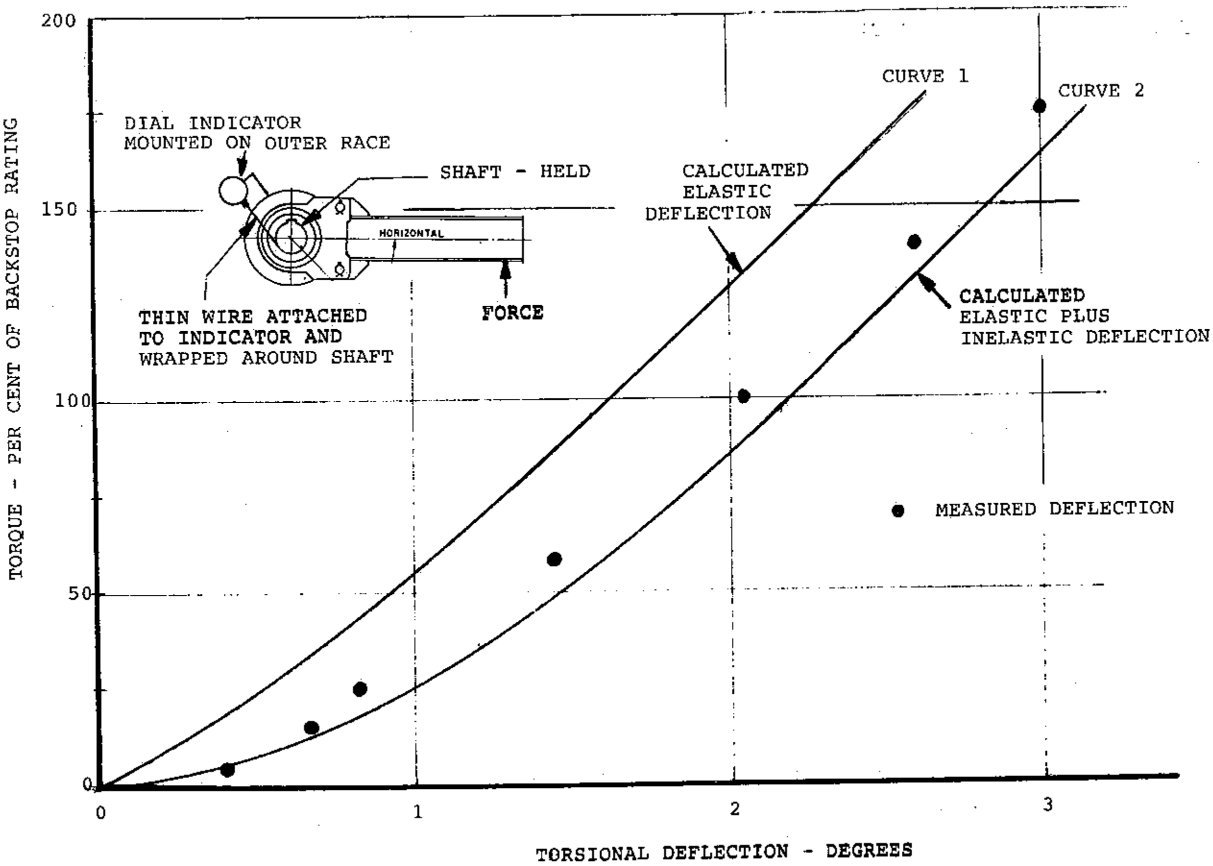

FIGURE 4

Important features of any design are those that reduce stress concentration and load distribution of contacting loaded members. In the design shown, Fig.#5 , note the outer race flange not only strengthens against a "bell" effect at the outer race ends, but also removes the side plate fastening holes from the stressed outer race mid-section as is done on some designs. Localized stress, 2 to 3 times greater than nominal can be created by the hole stress riser effect.

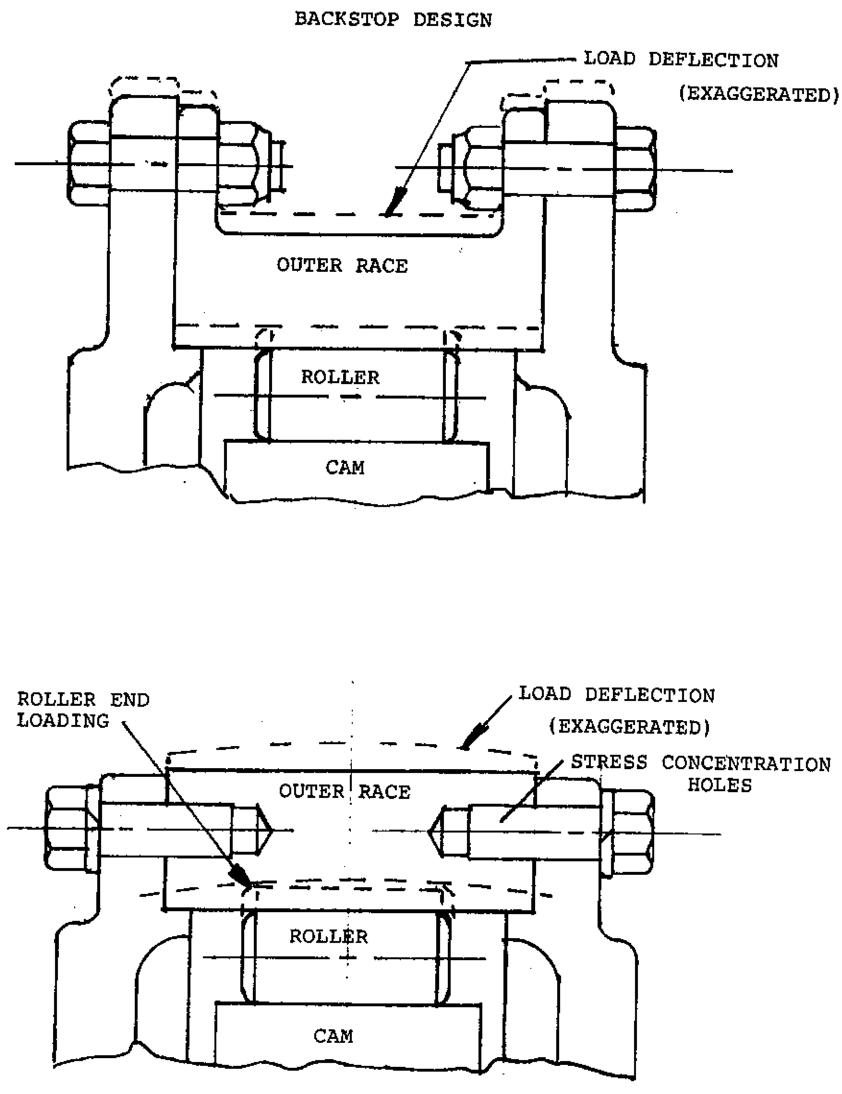

FIGURE 5

The materials selected must be able to withstand substantial roller to ramp and roller to outer race contact stress. Nominal contact stresses (Hertz) are in the 450,000 to 550,000 psi range with surface indentation (Brinelling) starting at approximately 650,000 psi up to 750,000 psi. These parts are normally case carburized or flame hardened to a depth which depends on load and dimension of contacting elements. Typically, hardness is 56 to 65 R_c.

BACKSTOP SELECTION PROCEDURES

Several methods for selecting backstop(s) for a belt conveyor system are available.

METHOD 1

Drive motor(s) Stall Torque. This method requires motor stall torque rating which implies that the belt conveyor system will not be subjected to a greater rollback load than the drive motor can develop.

METHOD 2

When the force required to lift the load vertically is greater than one-half the force required to move the belt and load horizontally, a backstop is required. A variation to this method would be to ignore one-half the force required to move the belt (which is the friction).

METHOD 3

Utilize a service factor against normal motor horsepower, lift horsepower, and lift horsepower minus one-half the frictional horsepower.

The initial response, with regard to a backstop, is to provide for any load contingency and as a result an oversized backstop is selected. The prudent conveyor designer will determine the potential for failure in the conveyor system and make engineering justified backstop selections.

In the absence of information, other than the motor horsepower, the backstop manufacturer will usually select a backstop sized for an estimated motor stall torque. If the actual stall torque for the motor is known, a more reliable backstop selection can be made.

Generally the conveyor designer has carefully analyzed the system and provided for the use of standard conveyor components, idlers, pulleys, belts, etc. The final item selected is the motor horsepower required to power the system. It is the distribution of power in the system that should be considered when a backstop is selected. On an inclined conveyor this power is generally consumed in lifting the load, with friction in the system being secondary. Under these circumstances the selection of the backstop need only concern the lift horsepower. Unfortunately, it is not that simple. Other factors come into play, such as overloading, jamming, and load distribution in a multi-drive, multi-backstop conveyor system.

The current method for selecting backstops in a multi-drive conveyor system assumes a load distribution of extremes between the driving elements in which the motor stall torque is the base consideration. A rational distribution of loading, by the conveyor designer, should be determined through analysis of the torsional and bending deflections of those elements directly loaded when the backstop is engaged. Examples are the backstop itself, the torque arm attached to the backstop, the stirrup at the reaction end of the torque arm, the belt, belt pulley shaft, coupling and any intermediate shafting driving the backstop. All of these elements deflect in sequence with the result that torque load to a backstop in a multi-backstop system is modified by the elasticity of the path. If the elastic deflection deviates considerably, the load sharing is proportionately affected. Studies of motion of loaded machine members show it usually consists of combination of elastic deflection and inelastic clearance. The elastic deflections are associated with stress/strain relationships of materials, whereas, inelastic motions are related to manufacturing tolerances of clearance fits. Fig. #4 shows the calculated torque deflection curve for a backstop. Curve #1 is the torsional deflection of a backstop with the elastic deflection summarized. Curve #2 shows the deflection when the extremes of clearance resulting from manufacturing tolerances are added. Somewhere between these two curves is the actual torque deflection curve for an individual backstop. The figure shows such a plot. Examination of the curves show that for a given deflection the torque response can vary significantly, which leads to the obvious conclusion that, for supposedly identical backstops, the torque deflection response is not identical. Extrapolating this relationship to the other conveyor system elements reveals that an unequal load sharing condition can result. It is in the best interest of design that the engineer recognize this condition and make appropriate corrections to minimize the differences.

Support for the analysis of the belt conveyor as an elastic system when making backstop selections is demonstrated by the following field test data. The multiple drive arrangement is shown on Fig. #6.

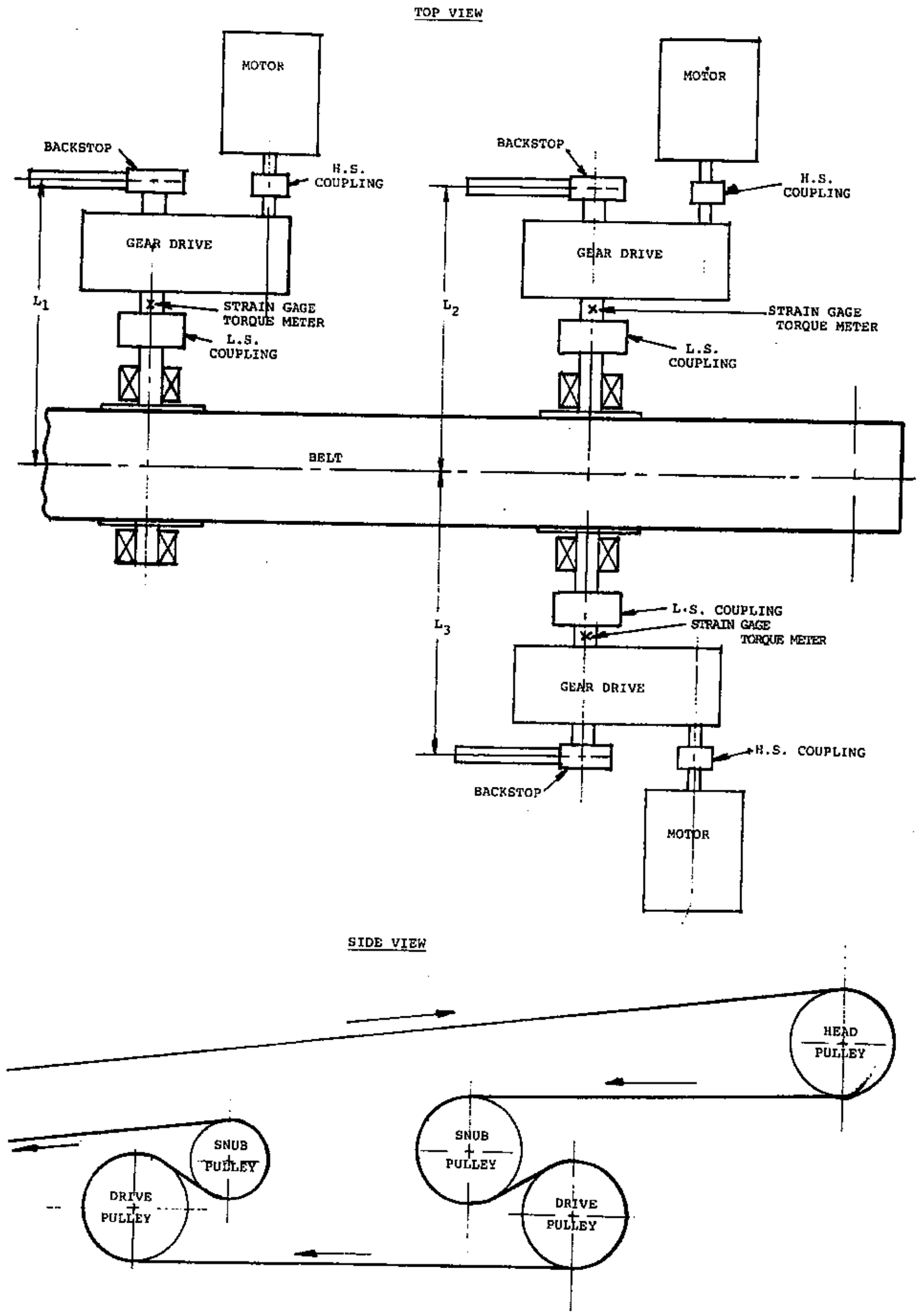

FIGURE 6

The L.S. shafts of the three gear units were fitted with bonded strain gages to record torque. The torque responses were recorded simultaneously by a tape for play back and analysis. Starting, running, stopping and static torque measurements were made with the belt loaded and empty. The data recorded indicated the following:

1. Load sharing at each drive varied from 27% to 40% of the total torque with the average 33%. Each drive would alternately carry the greatest proportion of the total torque load in a random varying manner.

2. The total backstop load was approximately 82% to 85% of the running torque load. Again, load sharing was random with alternating loads.

Conclusions drawn from the test data indicate that:

1. Fixed load sharing, in the driving mode, does not occur in a multi-drive belt conveyor system. The variations appear to be the result of constantly changing traction at each pulley.

2. The load sharing between backstops in the stopped mode follows the random load sharing experienced in the driving mode, however, it is compounded by the elastic response of the torque reaction.

3. The load sharing ratio between drives of the same horsepower to the same pulley was determined to vary between a minimum of 40% to a maximum of 60% of total load.

4. The load sharing <u>between</u> pulleys driven by the same horsepower vary between a minimum of 40% to a maximum of 60% of total load. The above distributions were taken in a matched drive system in which the system torsional and bending elasticity between the pulleys and the reaction point are dimensionally identical.

Under these circumstances, the maximum load carried by a single backstop in a 4 drive, 2 pulley system would be 36% of the total load carried where each drive is of the same horsepower.

An example of dual drive, single pulley belt conveyor load sharing analysis is as follows: Refer to Fig. #7 which shows two variations of the same capacity conveyor drive.

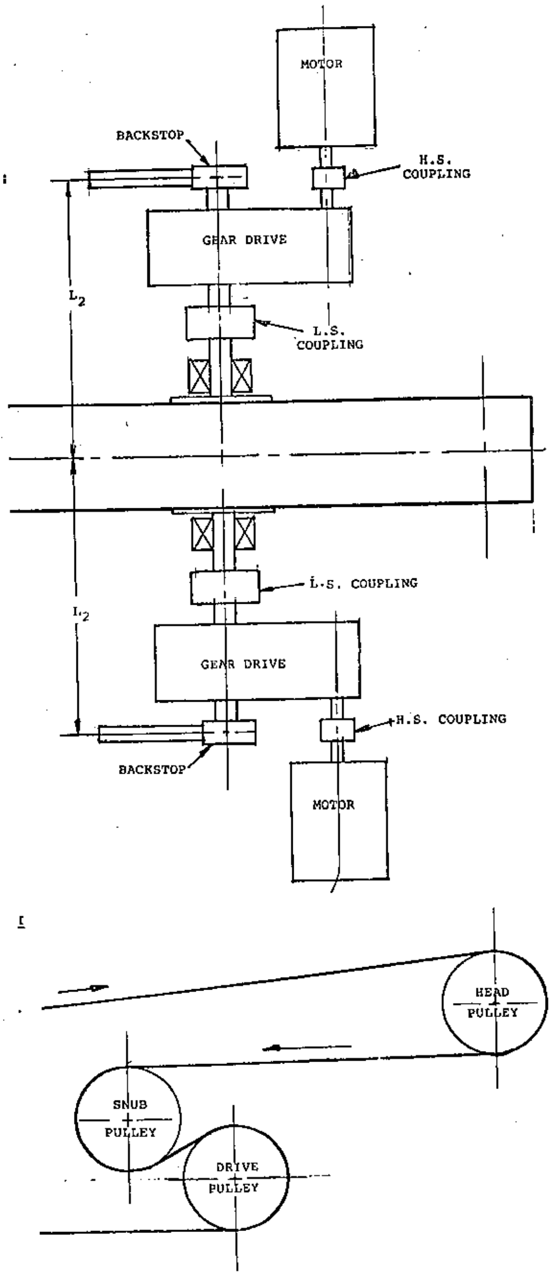

FIGURE 7A

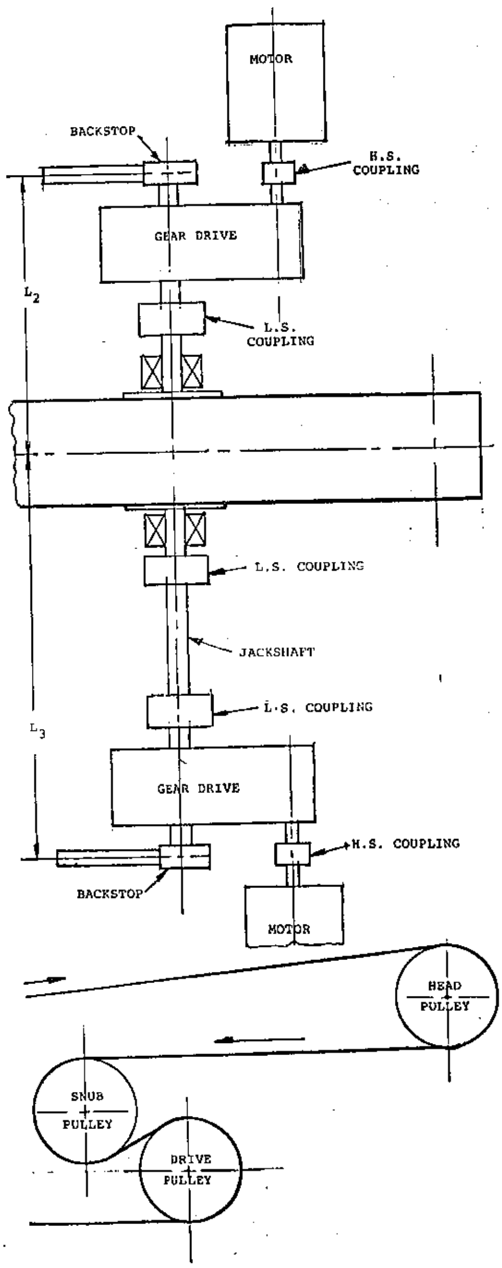

FIGURE 7B

FIGURE 7

In Fig.#7a the torque paths are dimensionally identical. In Fig.#7b the torque path on one side is longer than the other because of space requirements around the head shaft. A jackshaft and additional flexible coupling were added. Fig. #8 shows the summarization of the calculated average deflections converted to torsional deflection for each path.

AVERAGE TORSIONAL DEFLECTION BETWEEN BELT PULLEY AND BACKSTOP
REACTION POINT IN A CONVEYOR SYSTEM

1. Stirrup
2. Torque Arm
3. L.S. Coupling (Steelflex)
4. Backstop
5. Short Path - Shaft
6. Long Path - Shaft
7. Short Path - Summary
8. Long Path - Summary

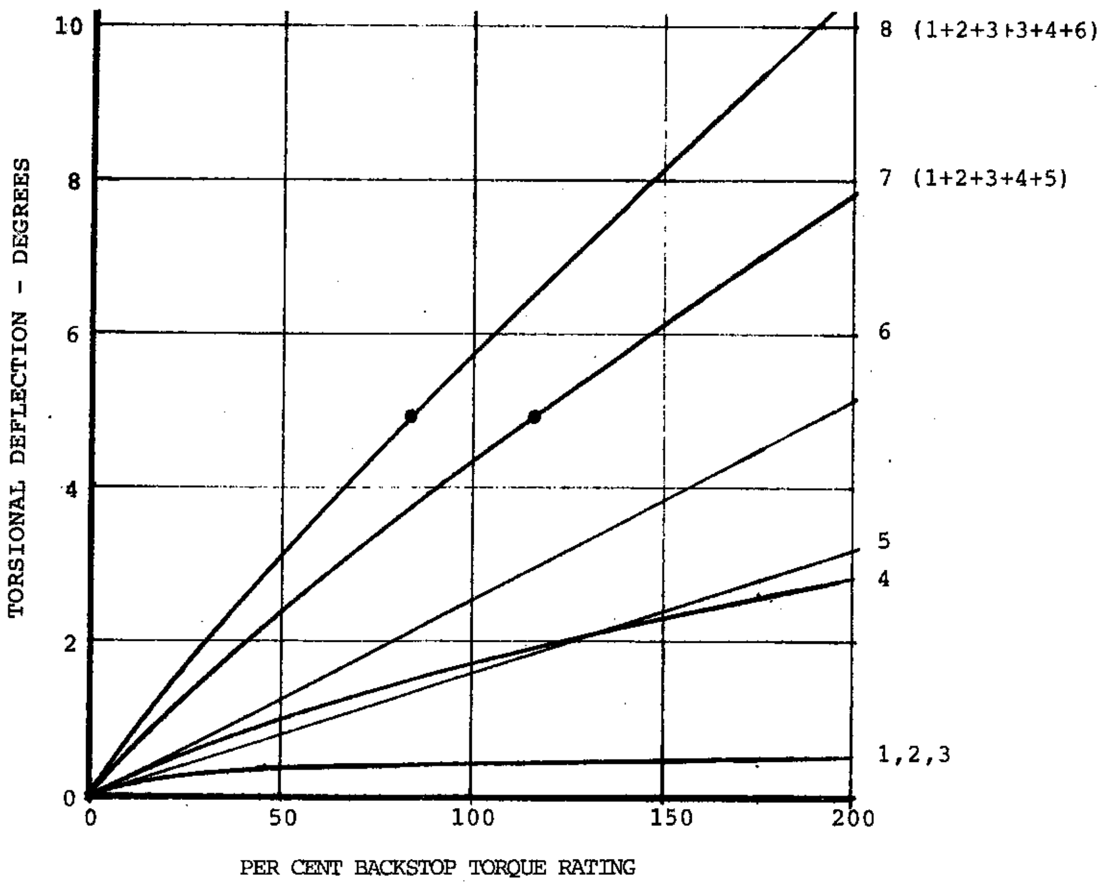

FIGURE 8

Notice the difference of deflection between Curve 7 and 8. For equal deflections the long path backstop carries approximately 42% of the total load while the short path backstop carries 58%. Note that the proceeding referred to the "average deflection"; Fig. #9 shows the long and short path deflections modified by the extremes of plus and minus deflection variations that must be considered in the absence of actual information.

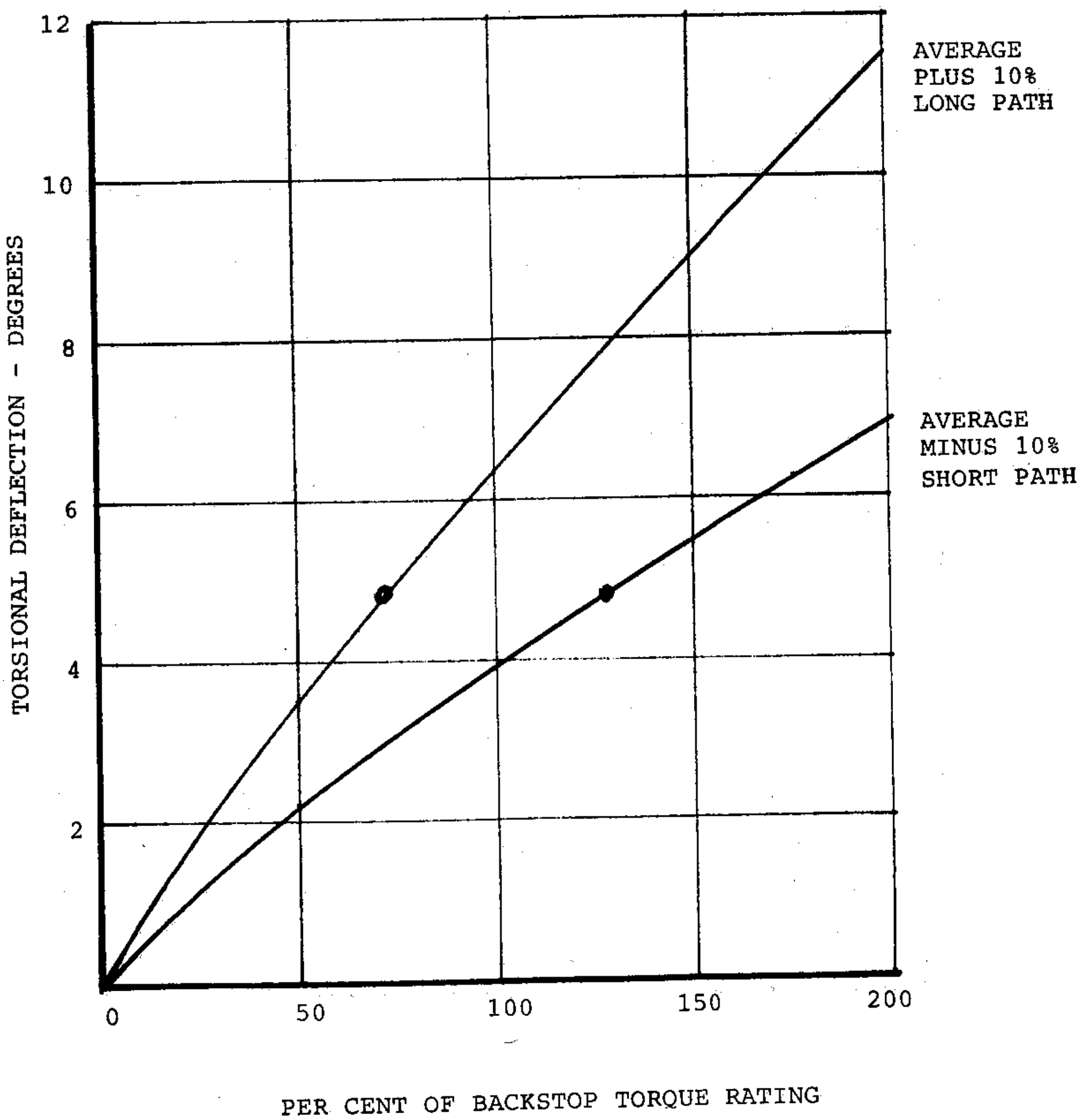

FIGURE 9

For equal deflections the long path backstop carries 36% while the short path carries 64% of the total load. In this case, it would be advantageous to try to use a single backstop on the short path side if the torque transmitting components are of sufficient strength to carry the total load.

In conclusion, the designer should make a torsional motion analysis to determine load sharing in any system containing two or more backstops, and make appropriate backstop size selections. If possible, the designer should try to equalize the torsional deflections in a multi-backstop arrangement by selecting proper sized elements to either increase or decrease torsional stiffness. The designer must also remember that the other torque transmitting components such as shafts, couplings, gears, etc. in an unequal elastic system will be subjected to the same loads as the backstop.