AN OVERVIEW OF THE USE OF FLYWHEELS ON TROUGHED CONVEYORS

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1. INTRODUCTION

In an ever more competitive industry, conveyor designers are constantly in search of innovative and cost effective ways to ensure both technical and economically optimal designs.

In recent years, variable frequency drives (VFDs) have become a cost effective conveyor control system which allows the conveyor designer to specify complex starting and stopping characteristics in order to reduce transient stresses.

However, due to power failures and emergency stops, the design of the conveyor must still be such that undesirable transient stresses are limited to acceptable levels during a non-powered stop.

In order to minimise the negative effect of a power interruption or non-controlled stop, flywheels are often specified as a means of extending the stopping time and reducing low belt tensions and eliminating dynamic effects which is normally achieved by a VFD during an operational stop.

Adding a large flywheel to the drive system raises safety concerns and the designer must take cognisance of the fact that this type of energy storing device may require special protection in the event of a catastrophic failure.

This paper investigates the reasons for selecting flywheels, economic benefits, safety aspects and preventive safety measures.

2. DISCUSSION

Elastic waves, sometimes referred to as transient stress waves, are generated in conveyor belts during starting or stopping [3]. In order to minimise the magnitude of these waves during the starting period, a prolonged start period with a velocity ramp in the shape of an S-curve is typically specified.

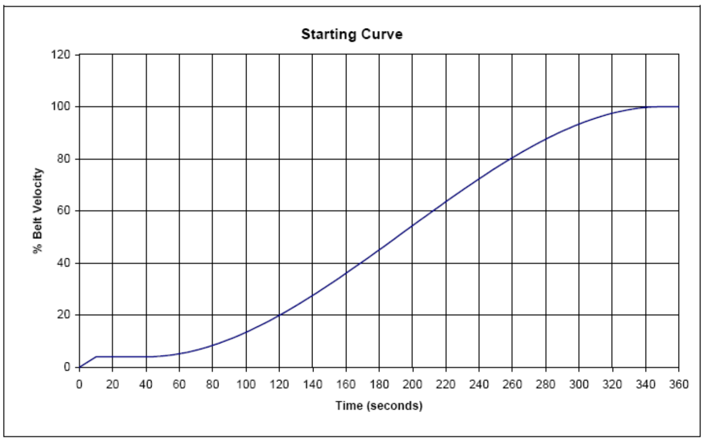

Figure 1. Typical Hermite S-curve for starting of long overland conveyors

However, a prolonged soft start is of little or no use if the other stresses, developed during the shut-down period of the conveyor, are not similarly contained.

With VFD control, an operational conveyor stop could also be specified as a predetermined graph or in some cases as a linear reduction in applied torque, say 1,5 - 2% per second, thus achieving a smooth transition period.

During an emergency stop with no power available to ensure a smooth transition period, some conveyors develop transient stresses accompanied by low belt tensions in certain regions of the conveyor.

The abovementioned problem can be solved by adding a flywheel to the drive system thereby increasing the stopping or coasting time of the conveyor to achieve a smooth ramp-down period.[1]

There are, of course, also other options available to address the abovementioned problem and the following case study investigates the various design considerations available to achieve an optimal design. It should be noted that there are factors such as conveyor over-run and the associated chute capacity which will also influence the final selection of the particular design option. This is considered outside the scope of this paper.

3. CASE STUDY – SHAFT INCLINE CONVEYOR

Conveyor Data:	
Capacity	2500 t/h
Length	890 m
Inclination	17 º
Lift	213 m
Belt Width	1500 mm
Belt Speed	3,1 m/s
Installed Power	2 x 1000 kW Dual Drive
Take-up system	Horizontal Gravity

Table 1. Conveyor parameters

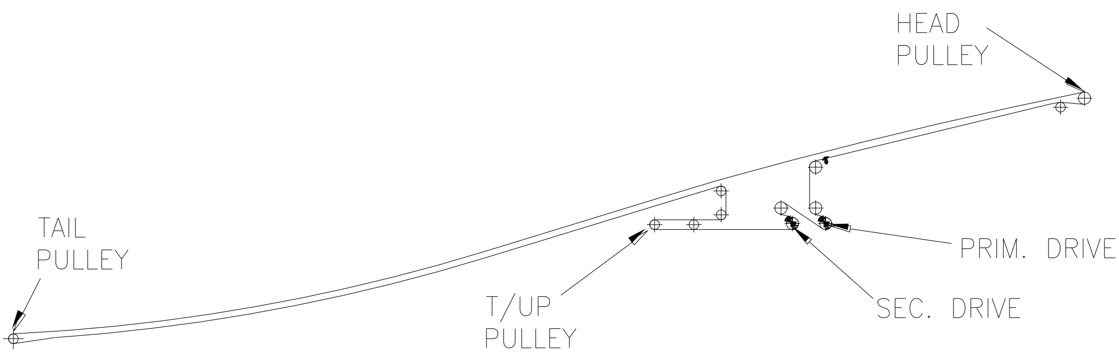

Figure 2. Layout of the head and drive end of the incline conveyor

4. DESIGN CONSIDERATIONS

A dynamic simulation, which predicts the elasto-dynamic behaviour of a conveyor belt, was conducted on the incline conveyor. The dynamic simulation indicated very low belt tensions with unacceptable levels of typically 12% sag at the lower end of the conveyor, and also dynamic shockwaves due to the very fast stopping time of the 17 degree incline conveyor. Increasing the take-up tension assisted in reducing the low belt tensions but the dynamic shockwaves were still present just prior to the conveyor coming to standstill. The dynamic instability is evident from the difference between head and tail speed of up to 30% during the stopping period.

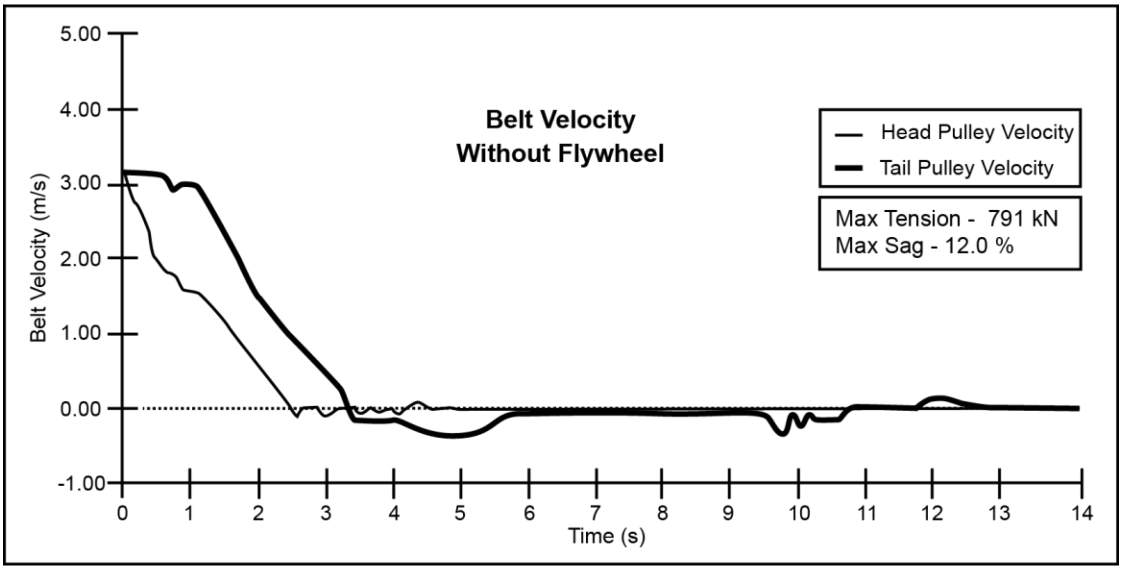

Figure 3. Belt speed without flywheels (emergency stop) [5]

The following design concepts were considered in order to eliminate the low belt tensions and the dynamic instability of the conveyor.

- Adding flywheels to the drive system
- Automatic take-up winch in lieu of the gravity take-up
- Capstan brake on the gravity take-up system
- Change the location of gravity take-up from the head end to the tail end

4.1 Flywheels

The dynamic analysis shows that, by adding a flywheel with an inertia of 225 kgm² to each of the two 1 000 kW drives, the stopping time of the conveyor is extended from three to eight seconds.

The required take-up tension reduced from 255 kN to 155 kN and the maximum belt sag reduced from 12% to 2,6%. Similarly, the maximum tension reduced from 791 kN to 705 kN for all the load case scenarios with the introduction of the flywheels.

In addition, the dynamic shockwave is eliminated and there is virtually no difference in belt speed at the tail and head during the deceleration period.

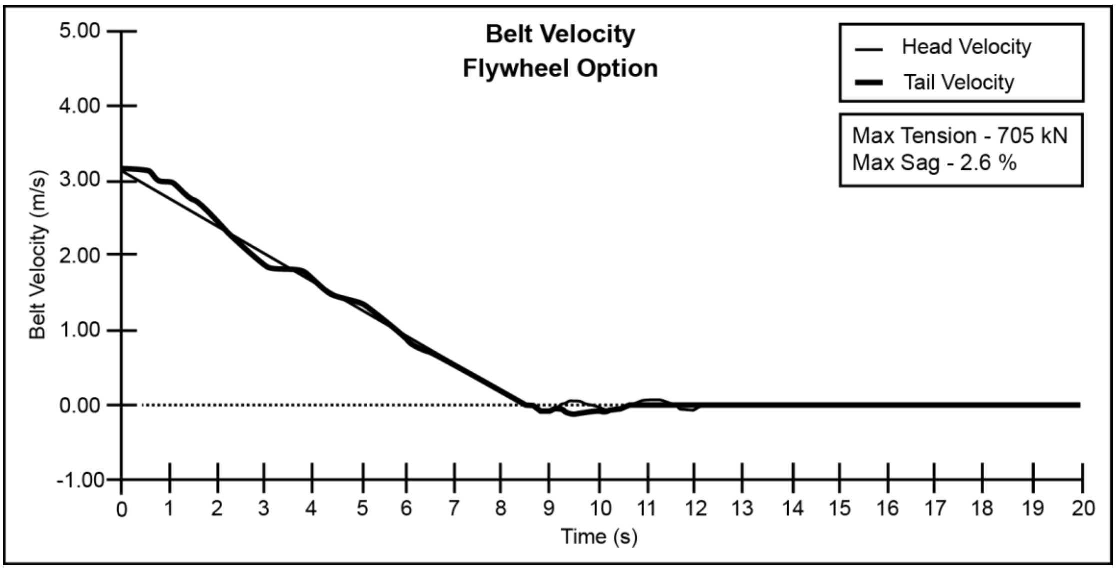

Figure 4. Belt speed with flywheels (emergency stop) [5]

4.2 Automatic Take-up Winch

An automatic winch would maintain the nominal 155 kN constant take-up tension during starting and steady state running, however it would be fixed during an emergency stop. The winch would have a worm gear and a brake on it to prevent any movement except during tension adjustments.[5]

The analysis proves to be very similar to the analysis done without flywheels. There is up to a 30% difference in the belt speed at the tail and the head with associated stress wave formations and low belt tensions, although the belt sag levels are well contained with the maximum sag level at 2,5 %.

The figure below shows the dynamic behaviour during an emergency stop. The maximum belt tension increases significantly during an emergency stop. This is due to the dynamics of the system and the backstop engagement shockwaves that occur without the flywheels. The peak belt tension is 847 kN. Furthermore the take-up tensions increase from the steady state value of 155 kN, to over 420 kN.[5]

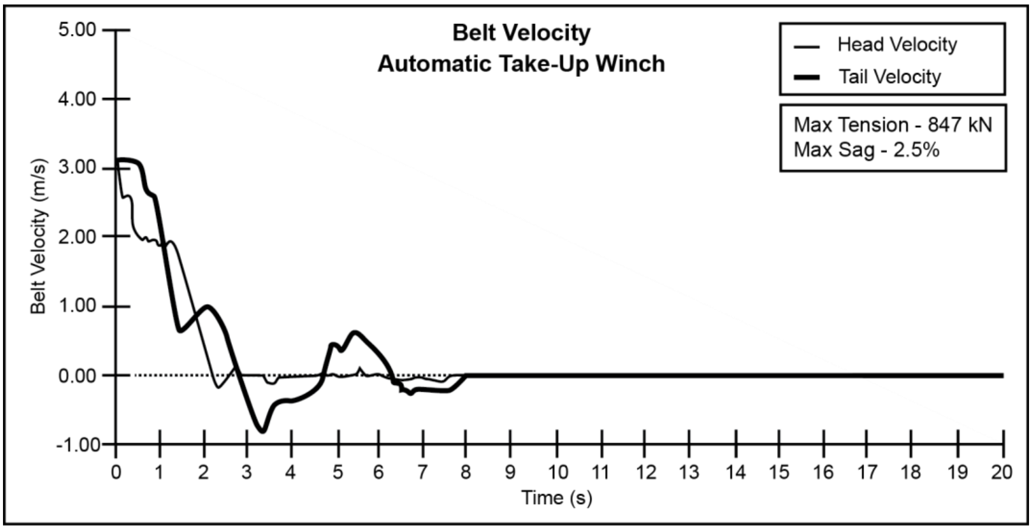

Figure 5. Belt speed with an automatic take-up winch (emergency stop) [5]

4.3 Capstan Brake

The capstan brake is essentially a brake system connected to the take-up counterweight that 'locks up' and prevents the take-up system from reacting to the sudden change in tension during a non-powered stop. [1]

The figure below shows the results of the capstan brake option. The belt sag of 3,5% is more than the fixed take-up option due to the capstan application which is not instantaneous. The maximum tension is lower at 705 kN for the normal friction load case. It is important to note that with a capstan, this conveyor is very sensitive to small changes in the design parameters. For example, under the low friction case (shown below) the maximum tension is 803 kN.[5] The low friction case needs to be included in the analysis since this scenario can be expected to occur after some time of operation.

The brake application and lag times can also have a very large effect on the peak belt tensions due to the very fast stopping time and dynamic shock wave that occurs during the emergency stop. [5]

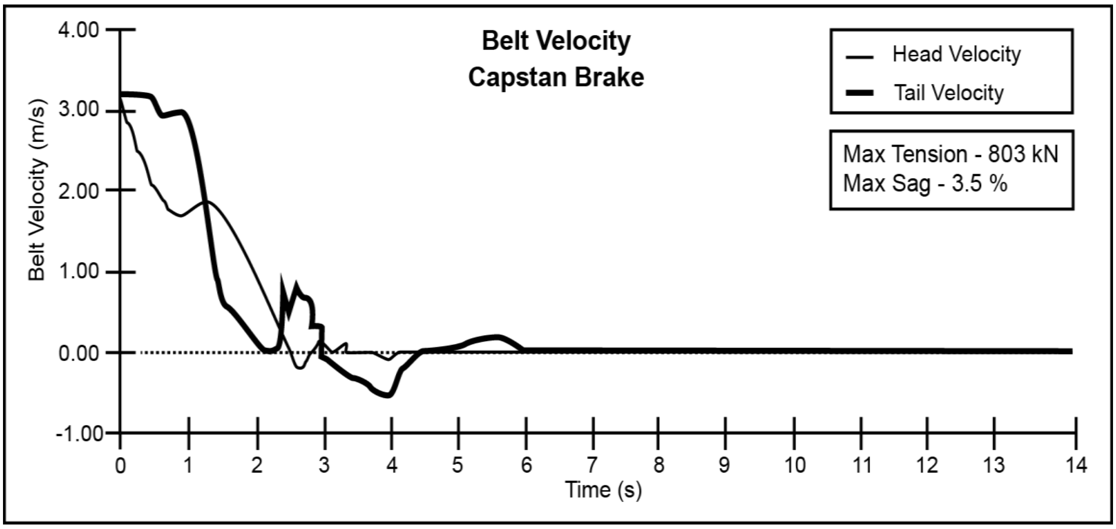

Figure 6. Belt speed with a capstan brake (emergency stop) [5]

4.4 Location of Take-up

Locating the take-up at the tail end of an incline conveyor should always be the first consideration for the designer. The take-up in the tail position allows for lower belt tensions and improved dynamic stability in the conveyor system as the belt self-weight provides the required slack side tension at the drives. This is, however, not always possible due to restrictions in terms of headroom and many incline conveyors are progressively extended down the incline shaft, therefore favouring the concept of having the take-up on surface. For this case study, a dynamic simulation with a gravity take-up situated at the tail-end of the conveyor was conducted to evaluate the effect this position would have on the dynamic stability of the conveyor.

In this case the take-up tension can be reduced from 155 kN to 50 kN. This is due to the gravitational effect of the mass of the belt on the return side. (50 kN tail tension in addition to the mass of belting results in the same required T2 tension of 155 kN)

During a power failure the tail take-up absorbs the belt slack and the maximum belt sag is 4,5%. However, the same high tension wave still occurs when the belt comes to rest and the backstops engage. Again this is due to the fast stopping time and dynamic shockwave. The maximum belt tension is 860 kN. [5]

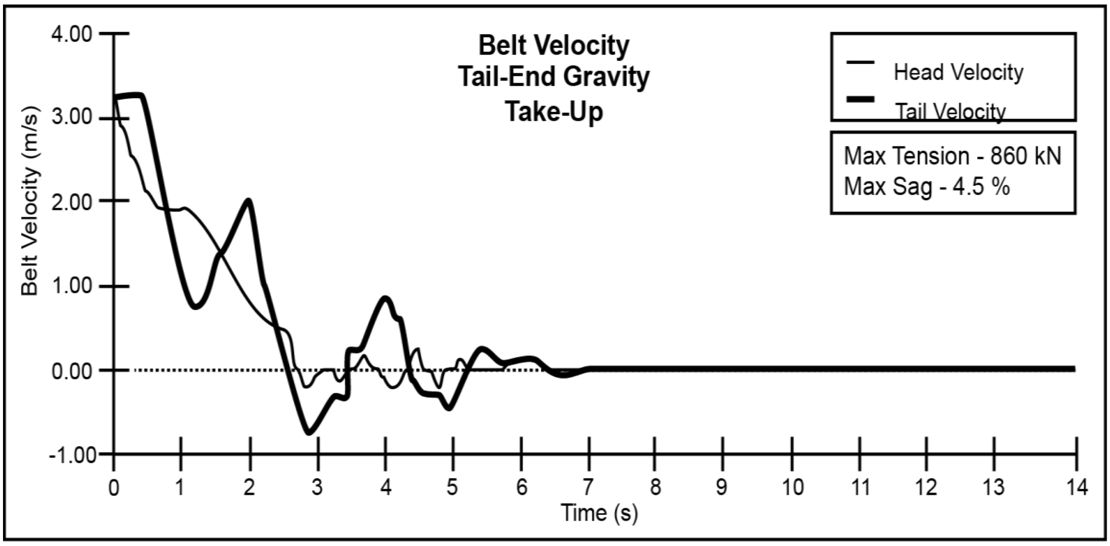

Figure 7. Belt speed with tail gravity take-up (emergency stop) [5]

5. DYNAMIC ANALYSIS RESULTS AND CONCLUSIONS

In this case study, the option of using flywheels fitted to the drive units to increase the total inertia of the conveyor resulted in the lowest peak tensions for all the load case scenarios and is also the most dynamically stable design during emergency stop conditions.

The lower belt tensions result in lower cost for the belting, pulleys, bearings and associated structural steelwork.

It can therefore be concluded that for this particular design case, using flywheels to increase the drive inertia and therefore the emergency stopping time of the conveyor, provided the optimal technical and also the most economical design.

It is also fair to say that the designer cannot only consider the technical and economic aspects and that the end user will require additional considerations to be taken into account such as the safety aspects and an evaluation of the additional maintenance associated with the flywheels.

For this application the 225 kgm² flywheel would be a 400 mm wide forged mild steel disk with a diameter of 900 mm. Each flywheel would weigh 2 400 kg. The sheer size of such a flywheel rotating at near 1 500 r/min raises safety concerns which must be adequately addressed during the design stage.

Disadvantages and safety risks:

- Safety – catastrophic failure, although the probability is low, the level of consequence is high
- High degree of secondary damage due to increased drive inertia
- Additional bearings and associated monitoring and maintenance

6. ADDRESSING THE SAFETY RISKS AND MAINTENANCE REQUIREMENTS OF THE FLYWHEEL

In order to ensure a safe flywheel design, a better understanding of the design process is required. The following will give a brief overview of the design methodology followed as well as the assumptions made in order to have a safe flywheel system incorporated into the drive of the conveyor.

6.1 Failure Modes

There are three main failure modes that can be identified when considering a flywheel failure:

- Bearing failure resulting in eccentric shaft movement which in turn results in eccentric flywheel movement.
- Shaft failure resulting in eccentric flywheel movement.
- Flywheel failure resulting in pieces of the wheel shooting out and the flywheel becoming eccentrically loaded.

The failure modes yielded the following two items that need to be addressed, considering the bearings and shaft are designed not to fail under maximum load conditions:

- Eccentric Flywheel Movement due to failure
- Protection from material flung from flywheel in event of failure

6.2 Primary Design Safety

The first protection in the flywheel is the design of all the components to ensure that there will be no failure due to poor bearing life, fatigue or instability.

The following design safeguards need to be implemented:

6.2.1 Flywheel

To achieve the desired inertia value the flywheel dimensions are modified until the desired Inertia value is obtained. The radius is limited up to the centre line height of the gearbox so the thickness is modified in order to obtain the inertia value. This causes the weight of the flywheel to increase significantly. The hoop stress and radial stress generated within the spinning flywheel needs to be verified in order to ensure that the flywheel will not fail due to rotational stresses. This is generally not a concern when the flywheel diameter is less than 1 000 mm and the speed does not exceed 1 500 r/min.

The flywheel itself needs to be balanced to G2.5 at 1 500 r/min in order to ensure proper balancing. Material certificates need to be given as well as ultrasonic testing needs to be done to ensure the material is homogenous. The material consideration is not critical as the maximum stresses during operation will not cause stress close to the yield strength of even mild steel. For larger diameter or higher speed flywheels this may be a concern.

For this design, forged mild steel was used. During normal operating conditions the maximum stress in the flywheel is 32 MPa. This is the inside hoop stress.

6.2.2 Flywheel shaft

The shaft diameter needs to be sufficient to support the flywheel assembly during normal operating conditions and the stress must be low enough to ensure the fatigue limit of the shaft is not exceeded under maximum running conditions. These conditions are the following:

- The flywheel weight shear force is considered (under maximum vibration conditions of 12 mm/s),
- along with the bending moment generated by the bearing supports,
- the torsion on the shaft is also taken into account under motor start-up torque conditions and braked stopping.

Once these forces are defined the shaft stress must not exceed 80 MPa on EN19-T shaft. This will allow infinite life of the shaft. For this design the shaft stress is 35 MPa during the maximum loading of the drive and under maximum vibration conditions.

6.2.3 Support bearings

The bearing life to be obtained should exceed the design life of the drive components. Even though the bearings will not have infinite life they must last long enough to be replaced along with the rest of the drive items. These bearings can be either grease or oil lubricated. Depending on the speed and load relationship, the bearing lubrication will change. The lubrication type should be selected by the bearing manufacturer. For this case study the bearing type used is oil lubricated pillow blocks for high speed flywheels as per the recommendations of various bearing suppliers.

It is also important that bearings be checked for minimum loads to prevent skidding since the bearings are overdesigned due to supporting shaft diameter. The oil viscosity needs to be considered for the application as well, and VG 220 should be sufficient for most applications.

For this case study the L10h bearing life is roughly 2,3 million hours due to the large diameter shaft.

6.3 Secondary Safety

If one of the primary design safety mechanisms has been bypassed by damage to the components or by non-standard operating conditions it is critical to contain the flywheel in the event of catastrophic failure.

When considering the failure modes it can be seen that the eccentric force will be the largest to contend with during a failure. In order to address this, the allowable radial movement of the flywheel has to be limited. This can be done using a ‘catcher’ which wraps around the flywheel in order to prevent the eccentricity from escalating upon a shaft or bearing failure as well as to contain any loose debris from a failed flywheel.

During a failure this catcher will be destroyed, but the flywheel will be contained. Further considerations that need to be taken into account in the catcher design is heat dissipation during failure as well as deflection of the structure.

Other considerations also need to be taken into account in the design of the catcher such as foundation strength of the base plate, material spillage protection, installation of catcher onto the drive and the ease of removal of the catcher.

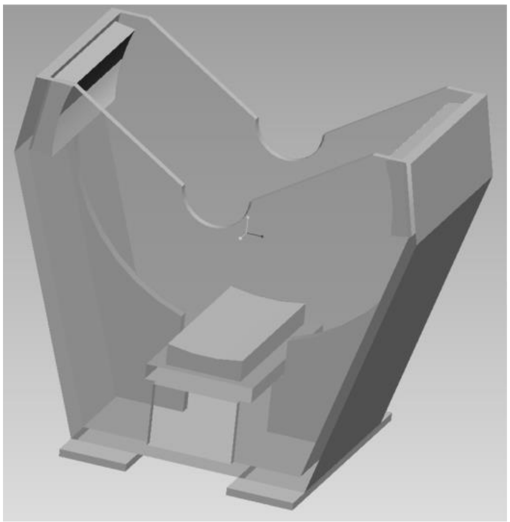

Figure 8. CAD model of catcher with transparent sides in order to view internal structure.

6.4 Flywheel Design Conclusion

The design of the flywheel and its ancillary components is not complicated once all the considerations have been taken into account. With primary and secondary safety features the flywheel is one of the safest components on the drive and no failure should occur if all design rules are followed. With the secondary safety device there should be absolutely no risk of loss or injury due to a catastrophic failure.

7. CONCLUSION

Flywheels are simple in design, cost effective and reliable energy storing devices and will most probably always be a consideration during the design stages of complex conveyor systems.

Due consideration must however, be given to the safety aspects especially with the larger type flywheels. It has been shown that, similar to the advances of technology for new conveyors, ancillary components such as flywheels can also be developed to new technological standards with adequate protection in terms of encapsulated guarding.

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