Cross Belt Sampler: Mechanical Design of the World's Largest Hammer Sampler for Bauxite Export Contractual Requirements

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Automated, mechanical cross belt (hammer) samplers remain popular because they are easy to retrofit into brown fields applications or green fields projects when cross stream samplers are not always designed into the plant layout from initiation. Cross belt samplers require less headroom and are easy to retrofit onto existing conveyors. Despite disputes about possible delimitation and extraction errors resulting from hammer samplers, they are not excluded for use from ISO sampling standards (13909 - Coal and Coke and 8685 - Bauxite) but are excluded from others (3082 - Iron ore). Possible errors can be mitigated by applying "know-how" into the bespoke design of a hammer sampler for installation on a specific conveyor belt system. This paper discusses the design details of a primary stage hammer sampler for a Bauxite ship-loading sampling plant. The design requirements of 10 000 metric ton per hour ship loading rate, 100mm particle top size, 1800mm wide conveyor travelling at 5.4m/s, results in a hammer sampler that takes up to a 260kg increment with each cut. The application requires more torque and at faster responses than that delivered by 10 Bugatti Chiron's combined and is (to my knowledge) the largest of its kind requiring a unique high-torque-at-low-rotational-speed drive system where conventional geared motors could not achieve the necessary output performance. Even though the sampler is the primary stage to a complete operational sampling plant, the emphasis is on the power requirement calculations, the mechanical design, components, materials and features of this unit that makes it not only mechanically operational but also intended at high sampling precision levels prescribed by the Theory of Sampling (TOS). Despite a small statistical bias detected for the sampling system (not the hammer sampler only), the sampling plant performed within the maximum tolerable bias specified for the commercial trade application and is fit for purpose.

Keywords: cross belt sampler, hammer sampler, TOS, mechanical design.

Credits: The Multotec Samplers design team who contributed to the design in their respective disciplines.

Introduction

The comprehensive application of the Theory of Sampling (TOS) into minerals sampling applications is still limited globally despite ongoing, active, and dedicated contributions and learnings from free publications by (amongst others) the International Pierre Gy Sampling Association (IPGSA). Some believe that the various related textbooks, technical publications, papers, best practices, and learning are too far removed from practical implementation to be understood by the industry (Steinhaus and Minnitt, 2014[1]), possibly because the theory is perceived as ambiguous (Pitard, 2005[2]). Pitard (2005[2]) goes on to explain that sampling standards are lacking in detailed prescriptions towards correct mechanical equipment designs that can reduce or eliminate selective materialization errors that known for large magnitude sampling error contribution. Without detailed prescriptive standards, the responsibility is transferred to knowledgeable stakeholders, engineering companies, consultants, and Sampling Equipment Manufacturers (SEMs) to ensure sampling projects achieve their intent. Sellers and buyers of mineral commodities are equally at risk of bias and/or poor precision level sampling where inaccurately reported grades occur. Sampling results (lack of) confidence levels can lead to remuneration losses/gains, let alone reputational damage towards quality assurance and control of large mining companies. The responsibility of the experienced SEM and their inputs into reputable sampling projects is explained by Steinhaus and Minnitt (2014[1]). SEMs have invaluable experience with sampling of mineral ores (Steinhaus and Minnitt, 2014[1]) and can often find themselves not only as preferred suppliers to the project, but at the same time consultants/advisors who contribute towards and promote correct sampling and uphold their responsibility where they form part of the client's decision making panel.

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This paper explains the SEM's key involvement and hence the panel's consideration of influential project parameters that concluded the design of potentially the world's largest hammer sampler over a theoretically "more correct" belt end crosscut sampler.

Hammer cross belt sampler history

Hammer samplers first appeared in the 1960's originating from German mineral bulk solids processing equipment supplier, Siebtechnik GmbH. The technology is still commonly used as primary belt samplers to sample periodic material increments from a production conveyor belt by scooping perpendicularly to the direction of conveyor travel across the belt width. Hammer samplers remain popular because they are easy to retrofit into brown fields applications or green fields projects where cross stream samplers require more headroom and are also not always designed in or space allowed for in the plant layout at project initiation. Hammer samplers take smaller increments than cross stream equivalents, especially for high-speed conveyors and this will often result in smaller downstream sampling plant with consequent reduced footprint and substantial cost savings. Despite purist TOS work (Pitard, 2005[2]; Robinson, Sinnott, and Cleary, 2008[3]) about possible delimitation and extraction errors that could result from automated mechanical hammer samplers – they are not excluded for use from ISO sampling standards (13909 - Coal and Coke and 8685 - Bauxite) but are excluded from others (3082 – Iron ore). It is my view that the possible errors can be mitigated by applying "know-how" into the bespoke design of a hammer sampler for installation on a specific conveyor application.

Project Background

The client understood the importance of correct sampling as an important contributor to quality assurance of bauxite grade between port-and-buyer. Commercial payment terms were structured around ISO 8685: 1992[4] with specification of Aluminium and Silica grade maximum tolerable bias (MTB) levels. Compliance to ISO 8685 and sampling plant performance to sampling variance within the MTB levels are therefore important to defend commercial trade risks. The client identified the need for correct sampling and ensured that commercial trade conditions were written around the applicable commodity ISO sampling requirement. They did their part well, only to be let down by the executing engineering company and a Sampling Equipment Manufacturer (SEM) who proposed a single stage hammer sampler. Apart from mechanical design inadequacy and evident under designed parts, this single stage sampling system would result in enormous composite sample masses forcing the operation to take too little sample cuts required for an ISO compliant sampling scheme. Thankfully, informed client technical adjudication role-players compared the single stage, primary sampler proposal against their experience with multi-stage sampling plants on iron ore, coal and manganese and probed the compliance of the offer with a different SEM. This SEM consulted and guided project stakeholders on: 1) sampling correctness, 2) adequate automated mechanical sampler design and 3) understanding the requirements for ISO compliance,

A further project constraint included production conveyors which were under construction. SEM advice on sampling correctness of a cross stream primary sampler was explored but concluded that existing conveyor designs and transfer tower heights did not allow for belt end crosscut samplers to be fitted at the head pulley. A feasibility study was conducted, with conceptual equipment designs from the SEM, to determine the techno-commercial project implications to install cross stream samplers. The anticipated project costs and reworks on existing infrastructure rendered this option to be unfeasible despite recommendations that quality sampling should be independent of economic aspects (DS 3077:2013[5]). Suitable positions were identified for installation of a primary hammer (cross belt) sampler, despite structural engineering challenges on existing conveyor stringer load allowances that would have to be overcome. Despite other ISO sampling standards prohibiting the use of cross belt samplers, ISO 8685[4] does not exclude hammer samplers from use in Bauxite sampling applications. The run of mine (ROM) material is crushed down to -100 mm export size using Mineral Sizers which are promoted to generate less fines and more regular shaped product both of which are suitable characteristics for representative cross belt sampler applications.

The client's mandate to be recognized as a reliable bauxite supplier overcame the technical and commercial hurdles to install a multi-stage sampling system with cross belt primary sampler as a "compromise" to cross stream samplers which are better regarded by sampling purists.

Literature Review

Cross stream versus cross belt sampling

The advantages, disadvantages, precision levels, reliability, and comparisons between cross stream (crosscut) and cross belt (hammer) samplers are discussed in detail by Steinhaus and Minnitt (2014[1]). They conclude (referencing the work of Rose, 2012[6]) that superior practical precision levels are achievable by cross stream samplers but not easily attained in poorly installed, inadequately maintained and neglected sampling plants where cross belt samplers are easier to install. The simplified installation is not only appealing commercially on project outcomes of cost and time, but the simplicity often results in better integration of the sampler with the production plant infrastructure and ultimately more likely to pass “bias tests” - still used as validation criteria of sampling plants for commercial mineral commodity trading contracts. Where hammer samplers are allowed by mineral ISO standards, they remain viable alternatives for practical implementation, but the responsibility lies with all stakeholders and particularly the SEM expertise and inputs to ensure successful sampling projects.

Mass versus time-based sampling and the weighting error

The Weighting Error is defined by TOS as the error that results when a sample increment weight is inconsistent despite consistent lot sizes (production weight intervals) it is supposed to represent - where the sampling rule of proportionality is not conserved. In the case of hammer samplers, they can only sample the material loaded on the belt at the instance of increment extraction. Even if the cutter speed set point is changed in between consecutive increments, the sampling action will not result in a different sample increment weight because the hammer sampler will still sample the material burden on the belt (now only at different velocity through the stream). The increment mass extracted by a hammer sampler will only change if the velocity of the production conveyor it is sampling from changes and results in a lower loading (ton per meter) on the belt. If each hammer sample increment is intended to represent a constant production weight, where the sample cut is prompted by a cumulative weightometer input in a mass-based scheme, there could be a discrepancy in the ratio of sample increment weight over constant production weight, between consecutive increments. For this reason, hammer samplers can only be used on a time basis.

Cross stream samplers, however, can be controlled over a defined range by varying frequency drives to change their cutter speed before a cut is initiated and then cut through the stream at that instantaneous speed set point. This possibility allows a cross stream sampler, running on a mass-based sampling scheme, to adapt its cutter speed to spend more or less time through the stream and result in a constant ratio of sample increment weight over constant production weight. Cross stream samplers can be used in mass- or time-based sampling regimes. For use in time-based regimes, the cutter speed must be consistent between cuts at a fixed set point irrespective of instantaneous throughput.

Cross stream hammer sampler design

System parameters: design inputs for this application

Table I below lists the client conveyor and material properties design inputs to the SEM for equipment mechanical design for this bauxite sampling application. The system design parameters resulted in a hammer cross belt sampler larger than previously built by SEM Multotec Process Equipment and affiliated Licensor Siebtechnik GmbH and to our knowledge, is the largest of its kind in the world.

Table 1. Conveyor and material design input parameters.

Conveyor parameters		Material parameters & properties	
Input	**Value**	**Input**	**Value**
Production throughput	10 000 ton/h – max. 7 000 ton/h – nom.	Particle nominal top size	100 mm**
Conveyor width	1 800 mm	Bulk density for volume	1.4 ton/m^3
Fixed belt speed	5.4 m/s	Bulk density for weight	2.3 ton/m^3
Idler angle	45° onset parameter	Nature	Abrasive, sticky
Idler angle *	Sampler design dictated 35° *	Transportable Moisture Limit	14%

*The sampler design dictated a 35° idler angle requirement. Constraints on belt curvature profile, cutter to belt gap tolerance and sample increment discharge trajectory are discussed in more detail in applicable sections to follow. The production belt profile was transitioned from 45° to 35° (and back) in 2.5° intervals with SEM supplied idler sets.

** Low fines fraction and regular, cubic-shaped particles expected from ROM crusher technology: mineral sizer.

Main components of the hammer sampler

The main components of the hammer sampler are shown in Figures 1 and 2 below. The design of these components' critical variables, parameters and tolerances are discussed in the following subsections.

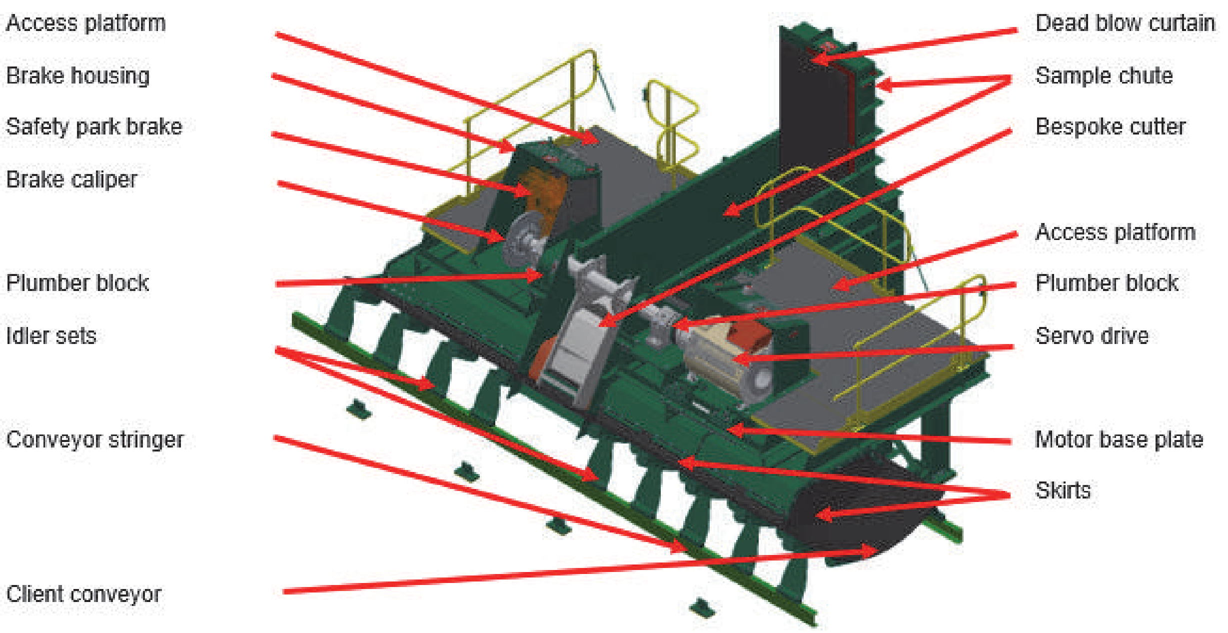


Figure 1. Main components of the hammer sampler seen from the drive end – sectional view without support frame. Handrails appear incomplete resulting from the sections made on the model to show the operational parts.

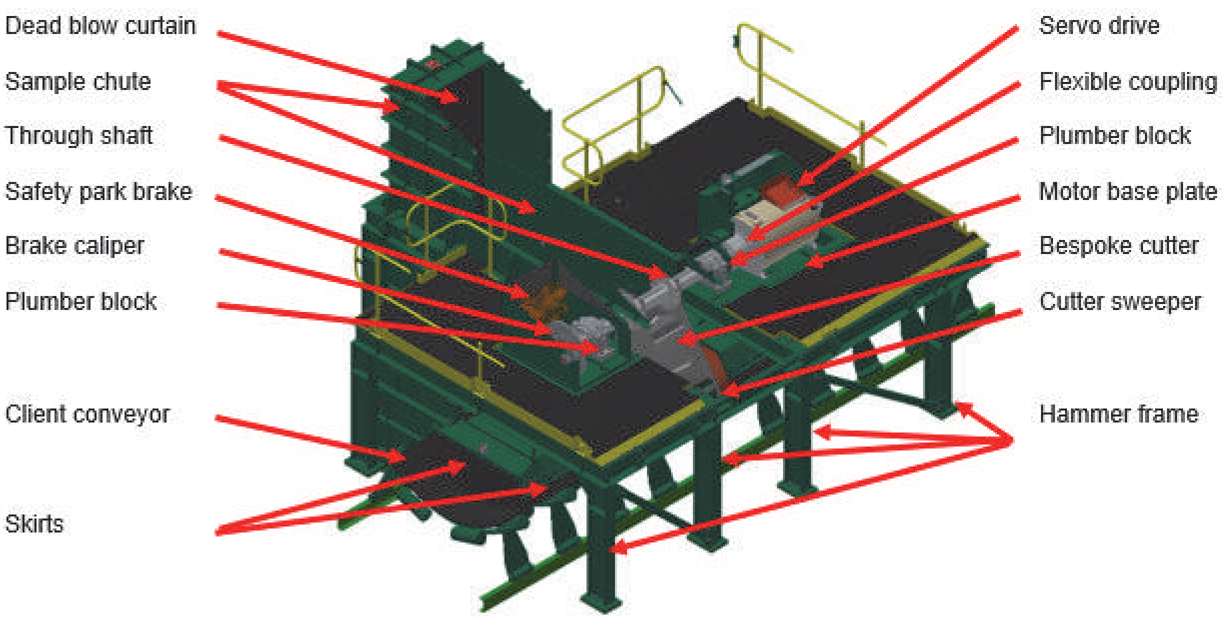


Figure 2. Main components of a hammer sampler seen from the park brake end –sectional view now including hammer integral support frame.

Cutter

The cutter is the heart of the machine and must be designed to minimize error generating mechanisms as set out by Pitard (2005[2]) and Robinson, Sinnott and Cleary (2008[3]). The resultant design outcomes of cutter mass, it's center of gravity and weight of increment that will be extracted by the cutter geometry feeds into the design power requirements of the hammer sampler. All the SEM cutters are bespoke design to the installations to result in accurate sampling with minimized materialization errors. The sections below elaborate on the additional detailed design considerations for this demanding application.

Straight versus skew cutter

Through their modelling simulations on coal Robinson, Sinnott and Cleary (2008[3]) quantified the extent to which both skew and straight cut hammer samplers were prone to delimitation and extraction errors when "over-sampling" material in their respective mechanisms. Their findings are summarized below:

- Straight cut hammer samplers: the leading cutter blade side wall (upstream) is subject to material pushing up against the outside of this cutter wall, which under friction from the approaching conveyor, is also thrown into the sample chute erroneously when not designated to be delimitated between the cutter blades.
- Skew cut hammer samplers: the combination of 1) angle of the cutter blades and 2) speed of cutter relative to conveyor speed could result in material not delimitated to be part of the sample, to be bulldozed off the belt by the trailing (downstream) cutter edge.

Both designs are not without possible bias if not carefully designed for the application. However, straight design cutters can be supplemented with sturdy rubber curtains mounted to the sample discharge chute on the outside of either cutter blade side wall. These curtains can deflect unintended material, built up on the side wall of the leading blade, back onto the production conveyor belt and prevent the oversampled material from reaching the sample chute and minimize or even eliminate particle misplacement. The straight cutter design is selected for this application and suitable rubber curtains designed with maintenance access to frequently inspect their condition.

Cutter opening (width/aperture)

Robinson, Sinnott and Cleary (2008[3]) explains a mechanism of a bow wave in front of the cutter movement through the stream that "blocks" the cutter opening and result in a delineation error where oversampled material is incorrectly thrown into the sample chute. The mechanism is applicable to straight- and skew cut hammer samplers. These extraction and delimitation errors result from not only direct interaction between sample-designated particles and the cutter blades, but also propagated interactions between designated particles and the indirect momentum transfer they effect to neighboring particles through a bridging effect. The proportional mass of oversampled material decreases with increasing cutter width where the effect of bridging is less pronounced. The authors recommend that cutter width (opening/aperture) be at least four times the particle top size being sampled - the maximum tested in their simulations. Naturally, the cutter blade leading edges must be sharpened in their design (and maintained) and the blade wall thickness as thin as possible to allow a "clean" cut while also maintaining structural integrity in its application.

A cutter width of 500mm with sharpened leading cutter edges, safe maintenance access hatches to regularly inspect edge condition and abrasion resistant stainless steel material of construction with suitably thin wall thickness of 20mm (a finite element analysis confirmed design) is designed for this application. The cutter walls (Figure I & II above) are designed to be fabricated from the same sheet of continuous steel to eliminate possible inward steps that can contribute to bridging and soft edges close to the belt surface, both of which can result in delimitation and extraction errors (design allowances on recommendation of Pitard, 2005[2]).

The 500mm cutter results in a linear proportional 1.67 times larger sample increment, calculated using equation [1]. This larger increment may result in larger and more downstream sampling plant infrastructure (crushers, sub-sampling equipment and interlinking solids handling conveyors and chutes) but through a case study had no significant cost addition to processing a three times nominal top size, 300mm cutter increment.

$$\boldsymbol{M_{increment}} = \frac{\boldsymbol{W_{cutter}} \times \dot{\boldsymbol{M}}}{\boldsymbol{v_{conveyor}} \times \mathbf{3600}} \qquad [1]$$

$M_{increment}$ is the increment mass in kg

W_{cutter} is the cutter width in mm

$\dot{M}$ is the production throughput in ton/h

$v_{conveyor}$ is the conveyor belt speed in m/s

Conveyor belt profile and gap between cutter and belt

Robinson, Sinnott and Cleary (2008[3]) reports a delimitation error through selective material loss of the -12mm fraction through the gap between back of the sampler cutter and the production conveyor belt. Their results were recorded for system parameters of cutter-to-belt gap of 25mm for a PSD of -50mm top size coal also containing <12mm particles. A standard industry practice and necessity (Pitard, 2005[2]) is to install a rubber sweeper at the back of the cutter (Figure 3) to sweep the sample-intended fines from the belt and prevent the potential delimitation error. Pitard goes on to explain that even though a rubber sweeper might be installed, a large cutter-to-belt gap will still result in losses when fines build up against the sweeper and are deflected sideways underneath the side walls of the cutter.

For this application the cutter gap is designed to 5mm. The client conveyor idler angle of 45 degrees did not allow a smooth radius of curvature but rather segmented trapezoidal steps between the 5 idlers which created 4 dead zones that cannot be sampled clean off the belt by a cutter with fixed radius of rotation. The cutter gap could only be reasonably maintained over the 1800mm conveyor width using custom design idler set designs using 5 roll, offset idlers, at 35° idler angle (Figure 4). The client conveyor had to be transitioned over SEM supplied idler sets from 45° to 35° and back to 45° in 2.5° intervals along the length of the conveyor. Conveyor idler transition should not be done without carefully considering belt tension, deflection points, stress points and belt loading to determine the length of transition and spacing between idler sets.

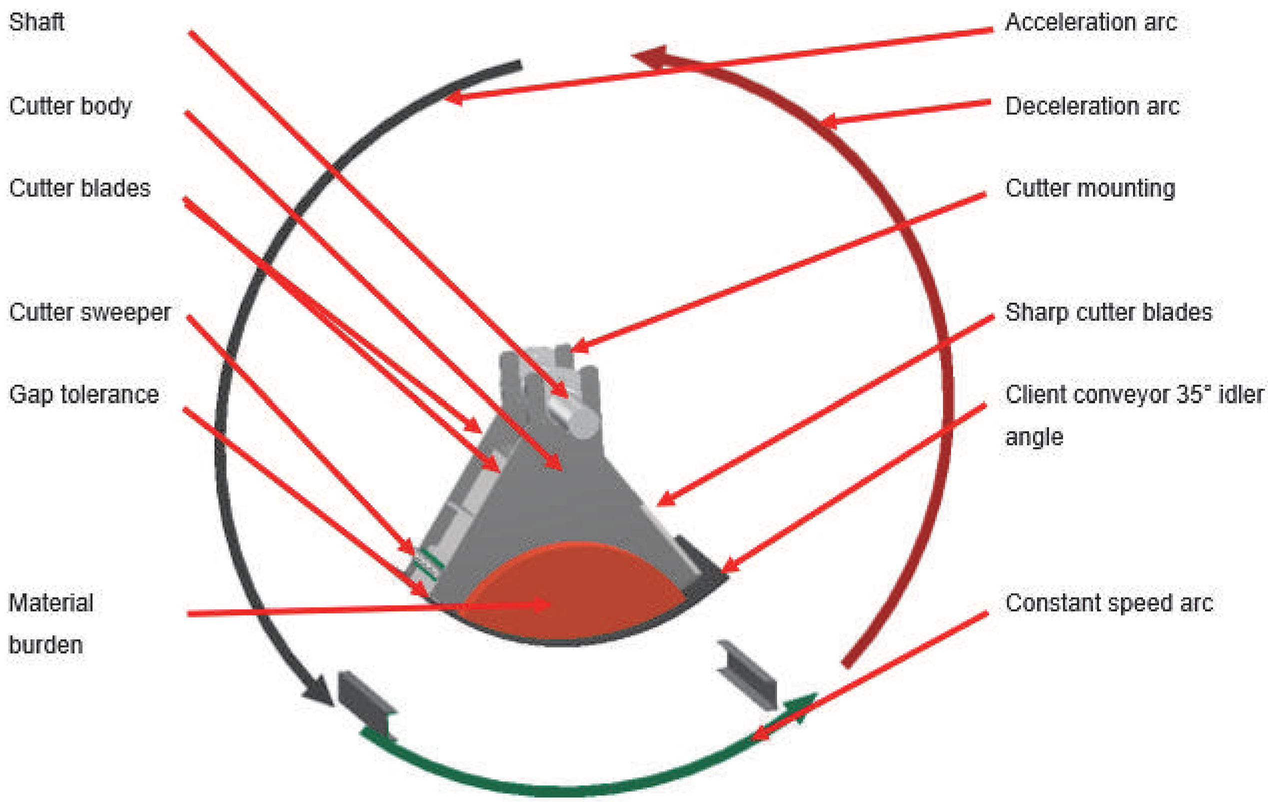


Figure 3. Cutter capacity relative to material burden and rotational arcs available for acceleration (black), constant speed (green) and deceleration (red).

Pitard (2005[2]) explains: "The last thing that should happen is for the cutter to lift or damage the belt as it enters the stream. Therefore, a gap between the bottom of the belt and the lower part of the cutter is necessary, especially as the cutter enters the stream". He recommends a pronounced gap between cutter and belt at the leading edge of the cutter but at the same time comments that this clearance can result in sampling errors. To prevent possible damage to the client conveyor, the 5mm cutter gap specified for this design across the full cutter arc therefore must be maintained with a tolerance of +-0.5mm. (This design consideration was paramount in sampler installation onto the conveyor because it meant that the 15 ton hammer sampler, capable of 16 kN.m of torque in 0.852 seconds, installed on an inclined belt, 14m above ground level, on its own support structure straddling the production conveyor has to be aligned overall to a tolerance of +-0.5mm. This was quite an engineering constraint, but a feat that was achieved, even though more costly than conventional hammer sampler installations.)

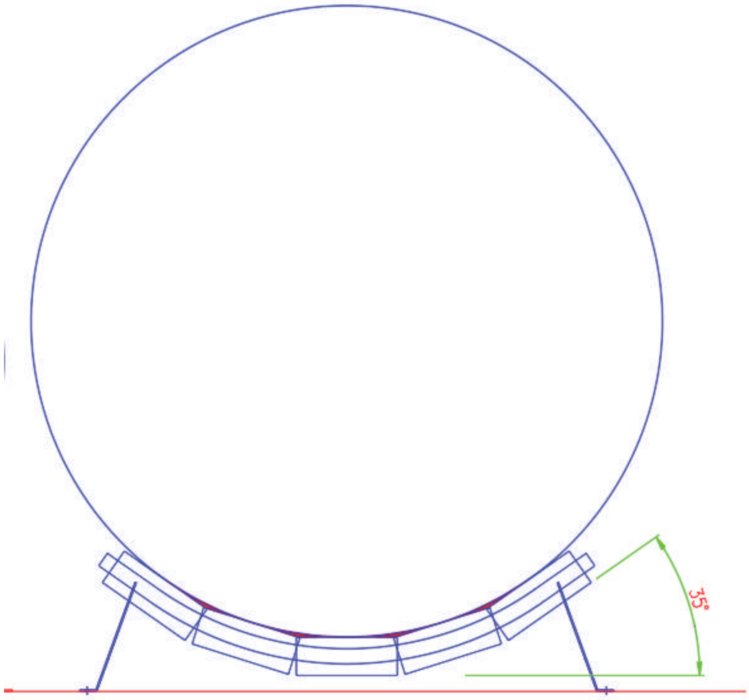


Figure 4. Cutter rotational diameter showing design tolerance between cutter and belt given the SEM supplied 5-roll, offset, 35-degree idlers to the 1800mm conveyor. The proportionately small dead zones remaining in purple can confidently be cleaned off with a sweeper minimizing delimitation errors.

Cutter geometry and capacity

As recommended by Pitard (2005[2]) the cutter geometry is designed to generously cover the calculated material profile on the belt (Figure 3). The depicted burden profile is calculated using the conservative bulk density parameter for volume design. Furthermore, the total cutter capacity is designed at 1.5 times the increment (extracted burden profile) volume calculation to prevent sample increment reflux that would result in delimitation error.

Cutter speed

Cutter speed is designed according to recommendation by Robinson, Sinnott and Cleary (2008[3]) who concluded that for a straight cut hammer sampler, a cutter speed of 1.5 times belt speed minimizes delimitation and extraction errors evident from the minimized amount of "oversampled" material thrown from the production conveyor. Given the production conveyor velocity of 5.4m/s, the cutter tip speed is designed at 8.1 m/s. The SEM confirmed with the client that after a soft start, the production conveyor is a fixed speed design. A process interlock to prevent sampler action until design belt speed is reached is employed.

Considering error generating mechanisms by Pitard (2005[2]) from insufficient sample discharge chute geometry: the design cutter speed will discharge the increment over the original client conveyor idler angle of 45 degree almost vertically upward from where it could erroneously report back onto the belt. The idler angle must change to 35° below the hammer sampler cutter for material trajectory to be flatter but requires a sample discharge chute with a high ceiling (Figure I & II). Considering the same mechanisms, the sample chute is also elongated in the direction of increment trajectory (Figure I & II) to prevent delimitation error resulting when material bounces back off the sample chute. As a final safeguard against this error mechanism, a durable, weighted, free hanging, natural rubber "dead blow" curtain is supported from the ceiling some centimeters before the vertical receiving wall of the sample chute (Figure 1 & 2) to break the material velocity down sufficiently from 8.1m/s and allow the increment to drop down.

Cutter coupling onto shaft

The cutter is designed to be a replaceable wear part. The cutter is flanged and bolts to the shaft's machined flanges. The assembly is more expensive to manufacture as opposed to complete cutter/shaft/bearing combination replacement but worthwhile considering future ease of maintenance, replacement time and replacement costs of a single part. The design is deemed necessary given bespoke, expensive shaft design as well as shaft alignment to servo drive motor via the flexible coupling which requires laser alignment tolerances.

Should a "cutter stuck in stream" scenario occur for uncontrolled reasons, the cutter is designed as the sacrificial part over other components of the machine because the cutter cost, ease of replacement and isolation of damage to other components rendered it the lowest impact part. The cutter is designed to warp at its neck below the shaft mounting flange. After controlled failure, the incoming material burden (approaching at 2.7 ton/s design throughput) is allowed to pass without major stream interference that minimizes spillages off the belt.

Cutter center of gravity

Discrete modelling software from a suitable design package is used to determine the center of gravity and its radius from the shaft center point for the cutter design. These values are used as input into the power requirement calculations of the hammer sampler.

Power requirements

Arc available for dynamic phases through cutter rotation

A hammer sampler cutter is typically stopped at (or near) the 12 o'clock position from where gravity contributes to acceleration downward towards the material stream and conversely with deceleration upwards away from the stream (Figure 3). Each sampler rotation has four zones that the cutter passes through: the arc of the cutter width in park position, cutter acceleration, constant speed sampling and deceleration.

The arc available for each zone is determined by the cutter design geometry, belt width and belt radius of curvature and obtained from drawing models. For this design, the available arcs are: 1) cutter arc 78°, 2) acceleration 102°, 3) constant speed 82° and 4) deceleration 98°. The cutter radius of rotation is designed as 1.22764 m. The angular velocity of the cutter (ω) can be calculated using equation [4] and for this design is 8.1/1.22764 = 6.598 rad/s. The angular velocity can be converted to revolutions per second (rps) by dividing through 2π resulting in 1.050 rps (or 63 revolutions per minute). This means that one complete cutter rotation takes 0.95228 seconds. For this design, the time of the cutter in each of the 3 dynamic zones were: (102°/360°) × 0.95228 = 0.26981 s acceleration, similarly 0.21619 s constant speed and 0.26188 s deceleration (the cutter arc in park position is not included in the time determination).

General power requirement

The apparent power required for each of the three hammer sampler dynamic phases can be calculated by the equation for power requirement of a rotating body: $P = 2\pi nFr$ [2]

P is power (Watt = kg.m^2/s^3)

n is revolutions per second (Rev/s)

F is force (N = kg.m/s2)

r is radius of applied force F (m)

Then, n is calculated by: $n = \frac{\omega}{2\pi}$ [3]

And in turn ω is calculated by: $\omega = \frac{v_{cutter}}{R_{cutter}}$ [4]

ω is angular velocity (radians/s)

v_{cutter} is cutter tip velocity (m/s)

R_{cutter} is cutter tip radius (m)

Writing [4] into [3]: $n = \frac{\omega}{2\pi} = \frac{1}{2\pi}\frac{v_{cutter}}{R_{cutter}}$ [5]

Writing [5] into [2]: $P = 2\pi nFr = 2\pi Fr\frac{1}{2\pi}\frac{v_{cutter}}{R_{cutter}} = Fr\frac{v_{cutter}}{R_{cutter}}$ [6]

F is calculated by one of two equations: 1) for the constant speed arc where the force requirement is that required to remove the increment mass from the belt and 2) the force required to accelerate the cutter body mass to the required angular velocity.

Power to remove the increment from the belt

By Newton's third law, and assuming negligible friction factor and assuming perfect collision conditions, the force to remove the sample increment from the belt is equal in magnitude and opposite in direction to the force required to drive the cutter through the stream at constant velocity, and can be calculated by: $F_{inc} = m_{inc}\, a_{inc}$ [7]

F_{inc} is the force to remove increment from belt (N)

m_{inc} -s the mass of increment (kg)

a_{inc} is the required acceleration of increment (m/s^2)

The increment mass m_{inc} is calculated by equation [1] listed earlier. By substituting [1] into [7]:

$$F_{inc} = m_{inc}\, a_{inc} = \frac{W_{cutter}\dot{M}}{3600\, v_{belt}}\, a_{inc} \quad [8]$$

Increment acceleration is calculated by: $a_{inc} = \frac{\Delta v}{\Delta t} = \frac{v_2 - v_1}{t_2 - t_1}$ [9]

Δv is the change in increment velocity (perpendicular to conveyor travel direction) (m/s)

Δt is the time taken for velocity change to occur (s)

The initial increment velocity (v_1) is 0 m/s, then $\Delta v \approx V_2$. For the same reasons $\Delta t \approx t_2$. Then [9] becomes:

$$a_{inc} \approx \frac{v_2}{t_2} \quad [10]$$

From the design parameters, we know that v_2 will be equal to 1.5 times belt speed and therefore:

$$a_{inc} = \frac{1.5\, v_{belt}}{t_2} \quad [11]$$

We know that t_2 = 0.21691 s and a_{inc} is calculated to be 1.5*5.4 / 0.21691 = 37.3 m/s^2. From [8], F_{inc} is then calculated to be 257.2 × 37.3 = 9.593 kN. With the assumption that the displaced sample increment's center of gravity is at 1.00 m from the shaft center, the power requirement can be calculated with [6] to be 63.3 kW.

Power to accelerate the cutter from park position to required cut-velocity

Ignoring the contributions of gravitational acceleration as a reasonable design assumption considering the gravitation will assist cutter acceleration downwards towards the stream and reduce force requirement, the torque required for acceleration can be calculated by: $\tau_{acc} = I \propto_{acc}$ [12]

τ_{acc} is the torque required to accelerate rotating components (N)

I is the inertia of rotating components (kg.m^2)

α_{acc} = angular acceleration of rotating components (m/s^2)

Then, simplifying the rotating components to be a single component and not a compounded object, the inertia can be calculated by [13]: $I = m\, r^2$ [13]

m is the mass of the cutter (kg)

r is the distance of the center of gravity from shaft rotation axis (m)

The simplified cutter inertia works out to be (with a design cutter mass of) 265 × (1.22764)^2 = 399.4 kg.m^2. Next, angular acceleration can be calculated by:

$\propto_{acc} = \frac{\omega_2 - \omega_1}{t_2 - t_1}$ but with ω_1 and t_1 equal to 0 (starting from park position), $\propto_{acc} \approx \frac{\omega_2}{t_2}$ [14]

The angular acceleration is calculated to be 6.598/0.26981 = 25.454 rad/s^2.

Now the torque required for acceleration can be calculated with [12] to be 399.4 × 25.454 = 10.2 kN.m.

Next the apparent power is calculated with:

$$P = Fr\frac{v_{cutter}}{R_{cutter}} = \tau_{acc}\frac{v_{cutter}}{R_{cutter}} \quad [15]$$

With resulting 10.2 * 8.1 / 1.22764 = 67.0 kW of apparent power required for cutter acceleration.

Power to accelerate the cutter from park position to required cut-velocity

Following equations 12 through 15 for deceleration, the apparent power is calculated as 66.4 kW.

Drive system

When the conventional geared motor will not do…

Three reputable global suppliers of geared motors were approached with above power requirements for proposals on a conventional 4 pole motor coupled to a suitable reduction gearbox. The feedback was unanimous, even after special consideration by head offices in Germany and Brazil: that geared motors cannot deliver the torque in the required times and that mechanical reliability of gearbox internal gears is not guaranteed because gear materials can possibly not withstand the shock loads of the application.

The servo drive and its engineered operational system

Recommendations by a consulting engineer lead to a solution in a high-torque DST2, multi-pole, direct current, electrical, gearless drive from Baűmuller Group. Over its operating range of 0-80 rpm, the selected servo drive model delivers up to 16 kN.m of torque in even faster response times than required without the need for a gearbox. But tis high precision drive comes with its own design considerations and constraints.

A direct current (dc) power supply is required for acceleration and maintaining constant speed through increment extraction. In the detailed design of the project, the power requirements were firmed up (including the effect of gravity and considering the cutter as a compound object) and the cutter acceleration control amended to a logarithmic function (not linear) from the drive's control system to efficiently power it through the 3 dynamic zones. The drive has a factory fitted motor resolver that mechanically monitors the drive output shaft rotation to within fractions of a degree; this monitoring is used by the drive's dc-control system to apply torque control throughout the shaft rotation.

The direct current technology also means that electricity will be generated in the deceleration zone when the hammer cutter acted as a dynamo; a brake resistor bank is employed to dissipate the generated electrical energy to atmosphere as heat energy. Considerations to apply phase control and synchronization to allow the generated electricity to be put back into the grid was considered but in this case is unfeasible given the hammer sampler's low duty cycle of a sample cut once every 5 minutes.

The drive's operational air quality requirement is equivalent to European Union, office environment air quality which is far removed from bauxite ship loading conditions in an equatorial climate. The drive is enclosed with an IP65 enclosure (Figures 1 & 2) and this enclosure fitted with a particulate filtration and moisture absorbing silica gel crystal breather that changes colour upon water saturation. An online relative humidity (and ambient temperature) analyzer is installed to monitor operational conditions within the enclosure.

The servo drive's inherent heat transfer mechanism is designed for continuous running. The hot and humid climatic conditions, high instantaneous ampere drawn coupled with a low duty cycle of the hammer sampler operation determine that the generated heat must be removed from the drive and it's dc- drive system by a forced (pumped) fluid-cooling system. The servo drive's factory fitted thermocouples were used as control loop inputs to control the cooling system. The cooling system has its own process and safety instrumentation and interlocks. The heat transfer design considers the dew point temperatures of the equatorial conditions to prevent moisture condensation in the servo drive internals. Also, the brake resistor heat dissipation is carefully engineered for the hot ambient temperatures and sampler duty cycle.

The precision tolerances between the drive's rotor and stator allows no linear force transfer (other than the intended motor output torque) or misalignment of the servo drive's shaft to within tolerances of 0.5mm vertical, 0.5mm horizontal and 0.25 degree offset. A flexible coupling between hammer sampler through shaft and servo drive output shaft allows torque transfer while decoupling deflection and offset transfer to the drive output shaft. Laser alignment between the two shafts ensures that drive tolerances are met.

Motor Base

An adjustable motor base allows precision laser alignment between drive stub shaft and hammer through shaft. Simultaneously the motor base frame transfers the torque generated by the servo drive to the hammer sampler main frame. Finite Element Analysis is conducted through design iterations to ensure mechanical rigidity to within +-0.5mm deflection tolerance of the hammer sampler's cutter to belt gap. All mounting interface surfaces for the servo drive to the motor base and in turn the motor base to the support frame are machined to design tolerances to allow force transfer through the full design area.

Cutter park brake

The servo drive is not equipped with mechanical park function / brake. The size and inertia of the cutter in free fall would result in fatality if accidentally released from park position when motor loses power or is isolated. Even though the servo drive and it's dc-drive allows functionality and control of balancing the cutter in a pendulum park position, this would generate heat which is not dissipated through servo drive rotation (while stationary in park position) and increasing the forced cooling system's duty unnecessarily. A mechanical safety park brake is used in-between 5-minute sample cut intervals. The brake need not be designed to serve as a dynamic stop brake in case of operational emergency because the dedicated dc-drive system controls also the servo drive emergency stop conditions.

Shaft

An extended, through shaft is designed to rotate the cutter while transferring the necessary drive power to the cutter and mount the park brake on the non-drive end. A suitable shaft material and heat treatment condition is engineered to allow for normal mechanical duties, but further rigidity to within the tolerance of +-0.5mm over the span of the shaft, as well as withstand the shock load over iterative analysis.

Bearings

Installation of a hammer sampler onto a sloped conveyor by means of a "shaky suspended bridge" support structure is not recommended (Pitard, 2005[2]). This statement is elaborated here considering the force vector down the conveyor slope that will be generated by a hammer sampler rotation where the sampler is not installed horizontally.

Over and above the normal bearing loads (perpendicular to shaft axis) of this heavy duty application and alignment tolerances, selected thrust bearings are required to withstand the force vector generated down the plane of the inclined conveyor that the sampler is mounted on (parallel to the shaft). These bearings are packed in plumber blocks. The bottom mounting surfaces of the plumber blocks must be designed to undergo additional manufacturing to be machined flat, post reliable OEM factory supply. The machined plumber blocks are bolted into plumber block seats which must also be machined to tolerance to ensure proper plumber block seating in the hammer sampler's support frame. Without machine surface interfaces, the force transfer available area might be compromised outside of design limits.

Housing

Pitard (2005[2]) acknowledges the importance of the (seemingly simple plate work) housing design as an integral part to the working and potential accuracy of a hammer sampler by recommending:

- Generous and safe inspection hatches (Figure 5) to allow maintenance inspection of wear parts to ensure design conditions are maintained,
- Eliminate delimitation and extraction errors by isolating and containing flung material from high velocity vector cutter-with-stream interaction through (refer Figures 1, 2 and 5):
- Generous ceiling design to contain material trajectory.
- Sufficient sample chute length to prevent sample increment bouncing back onto the conveyor.
- The addition of skirts and curtains that isolate sample material form unintended material sampling.
- In this case, the addition of a dead blow curtain to break sample increment velocity.

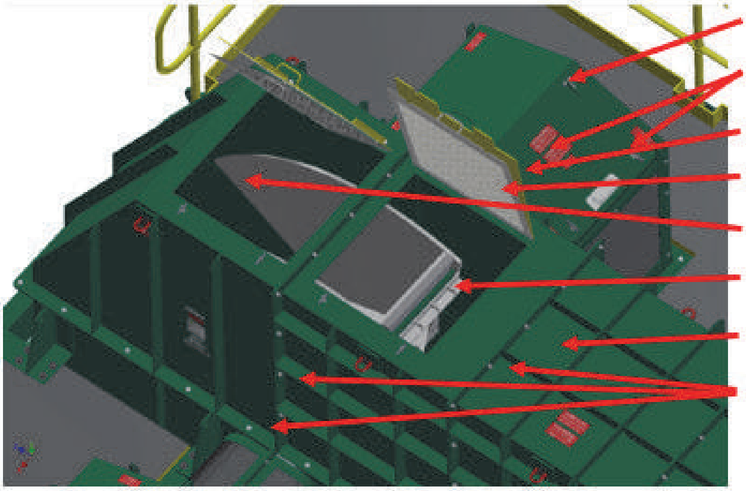


Figure 5. Housing design including multiple maintenance inspection hatches that are accessible to inspection, too small to climb through and safe: with double lockout possibility of solid door and/or expanded metal mesh screens.

Other outcomes of the housing are protecting operators against the rotating components (also display warning labels in English and operator first language option) and to protect machine components from external damage. All the above is included in the hammer sampler design. The housing of this design did contribute to overall machine static loads but did not contribute to stiffening of the structure which is all designed into the support frame. A client design request is to add additional low friction Tivar 88, abrasion resistant,3% w/w glass bead laced liners into the sample chute to assist with sticky material discharge.

Frame

The frame transfers the torque induced forces from the motor, through the motor base, to the hammer support structure and then into concrete pilons into the ground.

Critical parts of the drive train were bolted to the frame where both interface surfaces of the frame and the components are machined to tolerances to allow full area for force transfer. Seating surface areas are used in FEA design iterations to ensure deflection tolerances of +-0.5mm overall were met.

The resulting force transfer through the sampler frame's 10 feet mounting positions cannot be accommodated into the existing client conveyor's stringers. The engineering requirement is to build a suitable, independently grounded, support structure that straddles the client conveyor. This structure must support the hammer sampler 14 meter above ground level, must successfully transfer the generated machine force into concrete pilons driven into the swamp-like soil conditions – all while stiff enough to comply with the overall 0.5mm tolerance and last through cyclic operation.

The decision is taken to mount the ten hammer sampler feet onto two girders with 5 receiving pads each. These pads are machined across the length of the girder in a single milling action to ensure the alignment between hammer sampler shaft center and conveyor center line fell within the project design tolerance. The girders were in turn mounted to the hammer support structure.

Safety features

The mechanical safety features of the machine design are:

- The housing which guards operators against rotating components and machine parts against external damage.
- Individually lockable, solid and expanded metal mesh maintenance inspection hatches.
- Mechanical park brake.
- Audiovisual alarm before sampling plant start up.
- Hardwired emergency stop

Safety interlocks are:

- Emergency Stop activated or engaged.
- dc-Drive system ready
- dc-Drive Fault
- Cooling fluid heat sink chiller fault
- Internal motor temperature

Process interlocks are:

- Profile detector that prevents large lumps / abnormally high material burdens to be sampled outside of design intent.
- Cutter park position proximity switch flags a fault if the cutter is delayed or never reaches the park position
- Sample chute blocked chute detector delays / prevents the sampler from taking another sample cut
- Moisture analyser to delay / prevent sampling material with moisture higher than the TML that will choke up the downstream sampling plant
- Production conveyor running to prevent a sample cut until client conveyor is operational
- Additional dc-Drive process interlocks are considered as proprietary information
- A metal detector interlock is recommended to prevent damage from sampling rigid steel tramp

Control and instrumentation

The entire sampling plant infrastructure, drives, instruments, interlocks and sequencing is controlled via a Siemens Programmable Logic Controller (PLC). The hammer sampler however, required such fast response times that its own proprietary logic controller is used as a dedicated fast response processer located within the dc-drive unit.

At ISO 8685:1992 compliant primary sampling intervals of 5 minutes, the PLC would evaluate system healthy conditions and if confirmed, will prompt the dc-drive controller to take a sample cut. The dedicated controller would take control of hammer sampler rotation and once successfully completed, would relay a signal back to the PLC that a successful cut is taken.

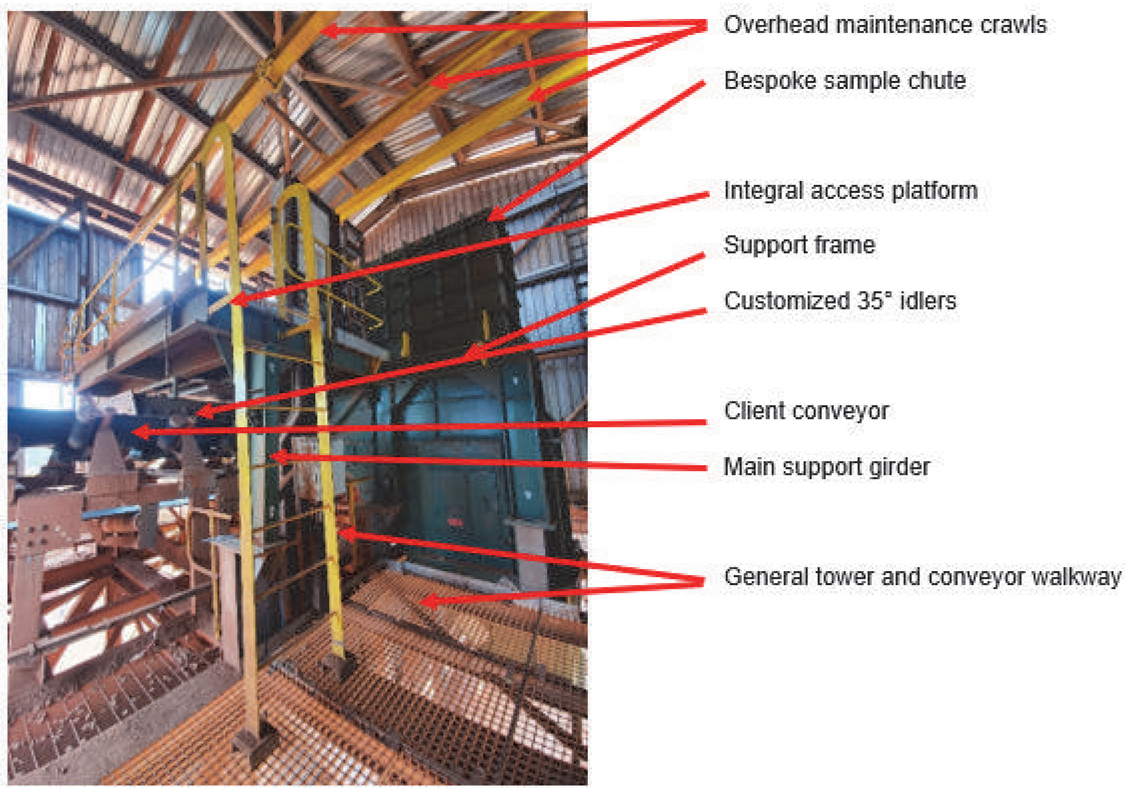


Figure 6. A photo of the installed and operational, world's largest hammer sampler showing some featured components and integration over the client conveyor. The technical design team nicknamed the machine Mjölnir, according to Norse mythology: the name of Thor's trusted hammer.

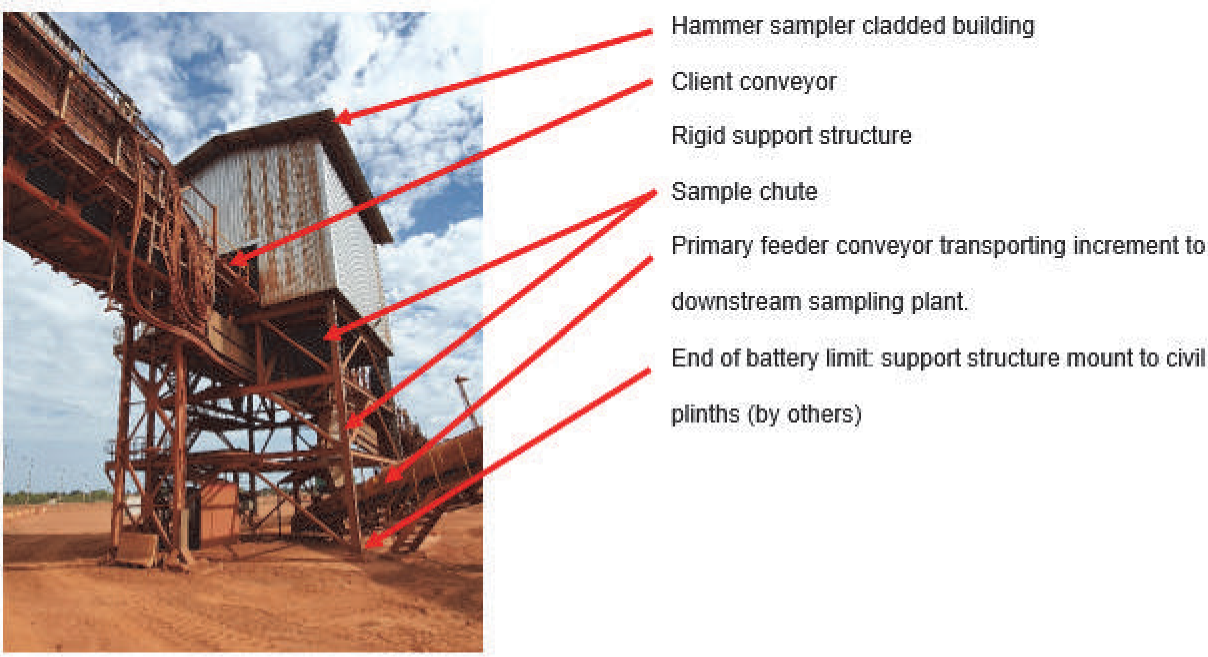
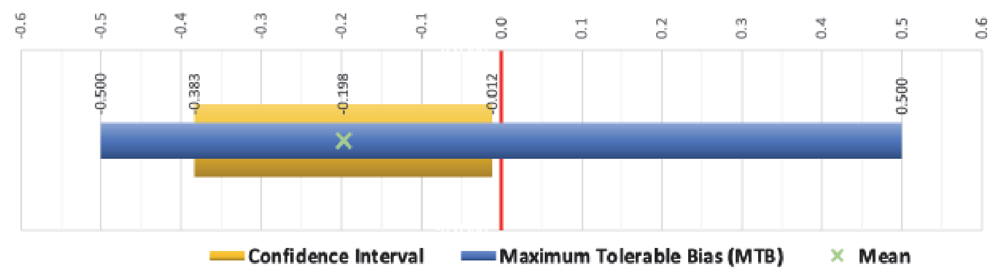


Figure 7. A photo of the installed hammer sampler's 14-meter-high tower, i.e. Mjölnir's home.

Performance – bias testing

The sampling plant, sample preparation and analytical procedures were bias tested 6 months after commissioning, by an independent, experienced third party. The bias test results can be seen in Figure 8 below. The reference samples were extracted from the stopped production conveyor as per standard industry practice and in compliance to ISO – 10226 Aluminium ores – experimental methods for checking the bias of sampling[7]. The composite chemical sample results were compared to the profile plate reference belts cuts over 60 sample sets (one set comprising the composite system sample and reference samples A and B). No separate bias testing of the primary sampler was conducted – only the entire sampling plant performance was tested.

Parameter	Significance of Bias		MTB Limit (%)	Confidence Interval (CI)
	Statistical Perspective	Practical Perspective		
Alumina (Al_2O_3) (%)	Significant	Insignificant	± 0.5	-0.198 ± 0.186



Parameter	Statistical Perspective	Practical Perspective	MTB Limit (%)	Confidence Interval (CI)
Silica (SiO_2) (%)	Significant	Insignificant	+ 0.2	0.124 ± 0.049

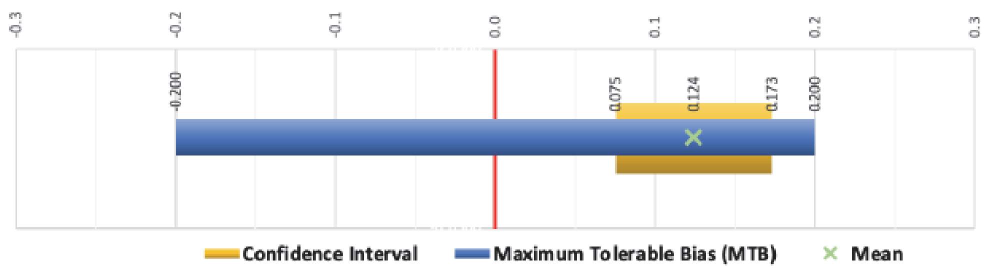


Figure 8. Bias testing results of the sampling scheme showing client specified Maximum Tolerable Bias Level (MTB) with the blue line, bias test results with the yellow line.

Bias detection was done in accordance to ISO – 10226[7]. A bias from a statistical perspective was observed where the average and standard deviation (position and span of the yellow line) does not intercept zero. However, from a practical point of view the data renders the bias irrelevant since the confidence interval falls within the MTB limits set out by the client as applicable to their commercial trade parameters.

Silica grade is overreported against the reference samples. The client informed that the Silica grade reports in the fine fraction. This implies that fines are oversampled in their proportion in relation to the Aluminium bearing coarser material. This finding is in direct contradiction to claims that hammer samplers tend to leave fines on the belt. An isolated bias test on the primary hammer sampler only (not the entire sampling plant) would be a more conclusive basis to support this claim.

Aluminium grade was slightly underreported while Silica content slightly overreported against the “truth” (reference samples). If the seller of bauxite is faced with a contractual claim for quality (low Aluminium content or high Silica content), the under and over reporting of respective elements would act in the bauxite seller’s favor to defend the claim when explaining that the sampling plant values are conservative on each quality parameter.

Conclusions

Revenue risk and reputational damage to mineral trading parties demands key and correct, representative, automated mechanical samplers compliant to TOS as essential. The absence of detailed mechanical design criteria in most international standards (that commercial transactions are based on) requires client, engineering houses and the SEM to uphold their responsibilities to sampling projects that successfully meet their quality assurance intent. These projects will often result in multi-discipline engineering projects which must not be underestimated in complexity and cost during budgeting, detailed design and execution. Where cross stream samplers are not practical given independent projects’ circumstances, hammer samplers are proven to be a viable alternative if designed correctly.

The accuracy and precision levels (representativeness) required for this metal accounting application required revisited consideration of all published error generating mechanisms and minimizing their effect through component-design and hammer sampler integration into the plant infrastructure. The cutter angle (straight versus skew), geometry, capacity, width and speed are influential design parameters. However, the selected 5mm gap between cutter and belt was paramount and dictated not only that the client conveyor parameters had to be changed to a smooth profile, 35° idler angle curvature– but also that the installed hammer sampler and all its components had to be designed to a tolerance of +-0.5mm dynamic deflection. This tolerance concluded that a 15 ton hammer sampler, capable of 16 kN.m of torque in 0.748 seconds, installed on an inclined belt, 14m above ground level, on its own support structure straddling the production conveyor has to be aligned overall to a tolerance of +-0.5mm. This engineering feat that was achieved, even though more costly than conventional hammer sampler installation. The client's strict requirement to be a reputable, quality assured mineral supplier overcame quality (prototype concerns, technical concerns and most importantly prior unreliable SEM influences), speed (project deadlines) and cost influences to support the successful implementation of the world's largest hammer sampler.

A bias from a statistical perspective is observed, but from a practical point of view, the standard deviation is low and falls within the minimum tolerable bias limits set by the client. Verification of the performance and material variances using variographic analysis according to methods described in DS 3077 (2013[5]) is recommended for future work.

The combined material parameters, conveyor parameters (particularly belt speed) and hammer sampler component mechanical design, results in a drive system power requirement that was unique and cannot be achieved with conventional induction motors and gearboxes. An existing servo-drive technology was pioneered in this application because it was the only known available option; even though its own considerations and constraints added complexity to the system it was successfully integrated and now boasts functionality of torque control and precision monitoring that is invaluable to its application.

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