Storvik HAL Compactor

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Abstract

The vibrocompactor is the main component in the forming stage of the green anode in the paste plant. The paste for an anode is densified in the vibrocompactor by dynamic and static mechanical forces during a time period less than a minute. The paper describes the different functionalities of the compactor and also different potentials it has for anode production capacity and quality in the future. The HAL vibrocompactor has been developed since 1959 and today the input of dynamic force is integrated to the cover weight (the vibrating mass on top of the anode paste). This results in more effective vibration and short vibration time. The feeding of paste is done from 2 hoppers in stationary position with minimal filling time of the mould. Production capacity is 36 anodes/hour at 22 s vibration time. The compactor has few movable parts and the maintenance and operation cost is very low. Availability factor is measured to 99,5% over 5 years.

Keywords: Anode forming, high effectivity, low maintenance cost, high availability factor

1. Introduction

The vibrocompactor is the main component in the forming stage of the green anode in the paste plant. The paste for an anode is compressed in the vibrocompactor by dynamic and static mechanical forces during a time period less than a minute. The report describes the different functionalities of the compactor and also different potentials for increased anode capacity and quality in the future.

In the period 1959-1975 was the basic principles for the HAL vibrocompactor developed where the input dynamic force was integrated to the cover weight (the vibrating mass on top of the anode paste) [1].

From 2000 has new functionalities added to this compactor:

a) Extra static force during vibration by implementation of pneumatic bellows pushing on the cover weight.
b) Vacuum vibration by implementation of a vacuum chamber around the vibrating process.
c) Softer isolators (damping shoes) to reduce the transmitted vibration forces to the building.
d) Increased hydraulic dynamical motor power, with soft start- and stop frequency curve, to increase stability and to vibrate larger anodes.
e) Individual control of the vibration period to equalize the “spring-damper” properties for each anode produced (control algorithm implemented in an executable file which communicates with the PLC system in the plant)~~.~~

The optimizations have led to a compactor which can be adapted to different paste qualities and produce anodes at a high production rate (anodes/hour) and larger anode size (ton/hour).

2. General Functionality

The vibrocompactor is built up with two vibrating masses which compacts the anode paste [2]. These two masses are named the cover weight and the table as the principle drawing shows in Figure 1.

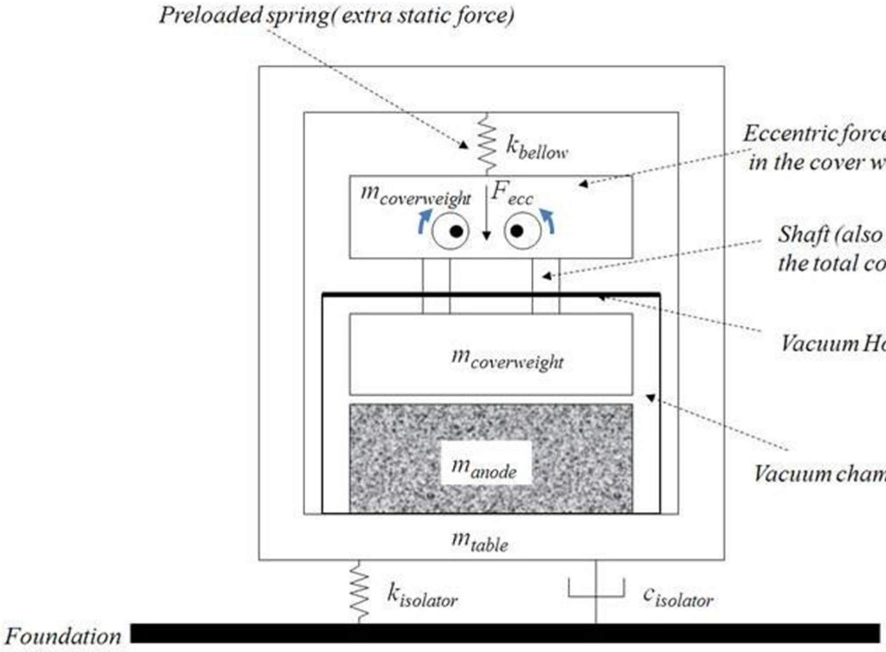

Figure 1. Principle drawing of the vibrocompactor.

The cover weight is the inner active weight since the hydraulic motor, generating the dynamic force, is integrated in this part. The table is an outer passive weight which vibrates due to the movements of the cover weight. The hydraulic motor gives energy to rotate two parallel shafts with unbalanced masses; the one shaft rotating clockwise and the other counter clockwise in order to create the vertical dynamic eccentric input force, F_{ecc}.

The rotational frequency of the shafts is large enough to let the dynamic force overcome the static force in the compactor. The right frequency is reached when the dynamic displacements [mm] do not any longer increase by increased frequency (even if the accelerations increase [m/s^2]). At this frequency, the weight and the table is vibrating at almost opposite phases, around 150°.

When the cover weight, the inner mass, is the active mass the dynamic displacements of it are generally twice the displacement of the table. Another reason for the difference is the weight balance; the cover weight is generally only half of the weight of the table. The displacement of the cover weight will therefore be closer to the table displacement when the mass of the cover weight increases.

The anode paste from the mixing- and homogenization phase can enter the two silos of the compactor in two ways; by one inlet using a paste splitter or by two inlets without the paste splitter. The compactor height could be reduced by dropping the splitter. See Figure 2.

The hoppers of the compactor is not thermally isolated. Experiences showed that the paste had minimum of sticking in the hopper without isolation. In addition, the sticking effect was further reduced by always changing which hopper to be first filled with paste. The maximum time between fillings of a hopper was therefore increased without decreasing the anode capacity.

The hopper system can also be rolled aside for maintenance, changing of mould and inspections.

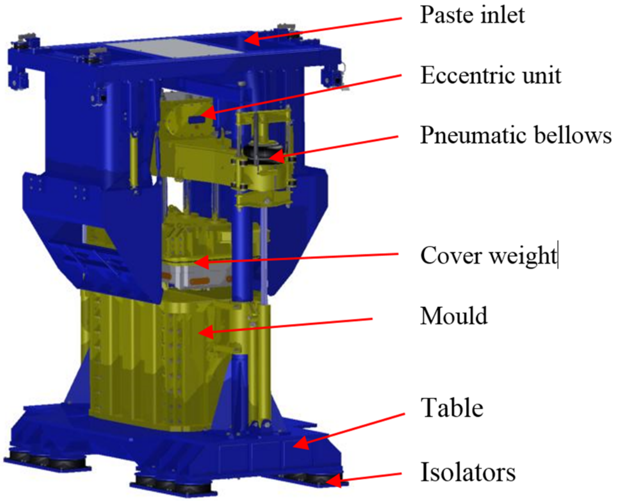

Figure 2. The main compactor parts

3. Function of dynamic motor power and the eccentric unit

The total compaction force, both the static, F_{st}, and the dynamic, F_{dyn}, in the anode can be written as shown in equation 1.

$$\begin{gathered} F_{comp} = F_{st} + F_{dyn} \\ F_{st} = m_{cover}g + F_{bellov} \\ F_{dyn} \propto (2\pi f)^2 x_{anode} \end{gathered} \tag{1}$$

The parameters, g, f, x_{anode} is the acceleration of gravity [m/s^2], vibration frequency [Hz] and dynamic anode displacement [mm] respectively during the vibration time. The dynamic compaction force is here derived by differentiating the dynamic displacement, $x_{anode}\sin(2\pi ft)$, two times to get a simplified expression for the acceleration which is proportional to the force.

The total compaction force can therefore mainly be changed by cover weight, static pressure in the bellows, the vibration frequency and the dynamic anode displacement. Both the magnitude of dynamic compaction force, F_{dyn}, and how the magnitude is achieved is important for the anode quality. The same dynamic compaction force can be given by low frequency, f, and high anode displacement, x_{anode} , or by high frequency and low anode displacement. The ratio, x_{anode}/f, is set by the eccentric unit of the compactor. The ratio increases when the unbalanced mass of the two parallel shafts in the eccentric unit increases (moving to the left in Figure 4). This can also be seen in Figure 5; the frequency decreases and the displacement increases by heavier unbalanced masses (moving to the left in Figure 5). The compactor is turned closer to a situation where the “hammer is lifted higher before it hits the nail”. The anode paste experiences more an impaction rather than a smooth movement following a sine function. When the ratio of, x_{anode}/f, increases the sine movement turns into a “deformed sine” (sum of sines with different frequency components) where the peak value of the new signal moves further away from the average in the signal. The average signal is defined here as the root-mean-square value of the signal, denoted . The signal contains more and more frequency components while the crest factor (peak-to rms ratio) of the signal increases.

There are two unbalanced masses (mass 1 and 2 in Figure 3) on each shaft in the eccentric unit. The eccentric position is defined as the angle between the two unbalanced masses on a shaft. At an angle of 180° the unbalanced masses cancel each other and the eccentric force will be zero as shown in Figure 4. Maximum unbalanced mass is achieved at a zero angle. The eccentric unit is designed with 18° intervals of the unbalanced masses. A typical range for the compactor in the production line is from 54-126°.

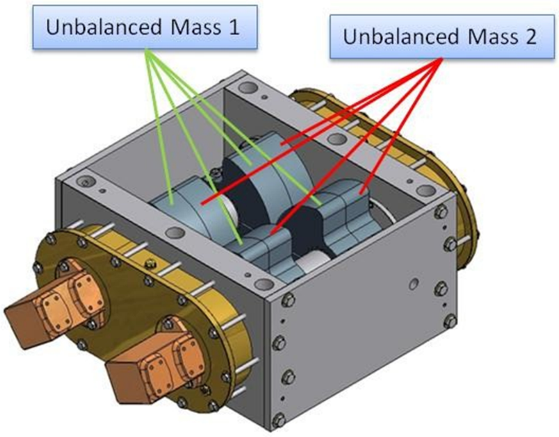

Figure 3. The eccentric unit of the compactor

The eccentric unit is designed with 18° intervals of the unbalanced masses. The operation of hanging the eccentric position demands a 15-minute stop of the specific compactor during the production of green anodes. The eccentric forces can also be designed to other specific demands by also designing new unbalanced masses.

Figure 4 shows the eccentric forces created at 22 Hz vibration.

Experiments from the production lines shows that the baked anode density increases by moving to the left in Figure 4 and 5, which means an increase of the ratio, x_{anode}/f . The left movement in the Figure (heavier unbalanced masses) demands also more hydraulic power since the eccentric force increases.

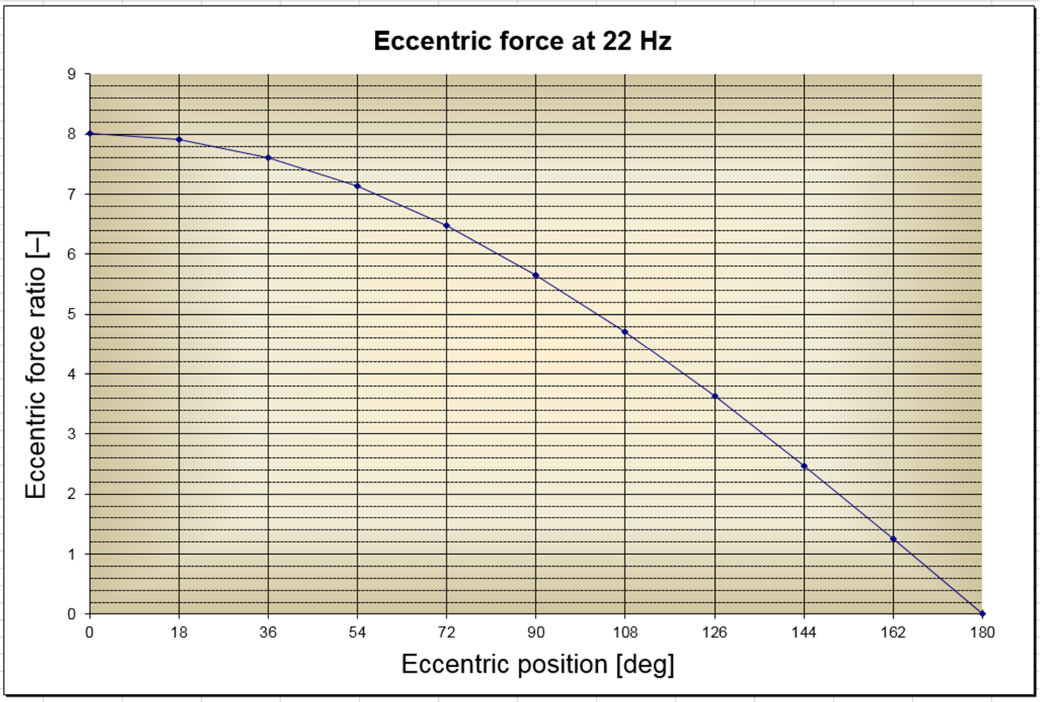

Figure 4. Eccentric force as function of the position of the unbalanced masses.

An increased ratio of x_{anode}/f means also a direction towards unstable vibration. A way to minimize instability effects is to increase the range of the hydraulic motor effect. Another way to minimize instability at high eccentric forces is to increase the static force; either by cover weight or pressure in the bellows.

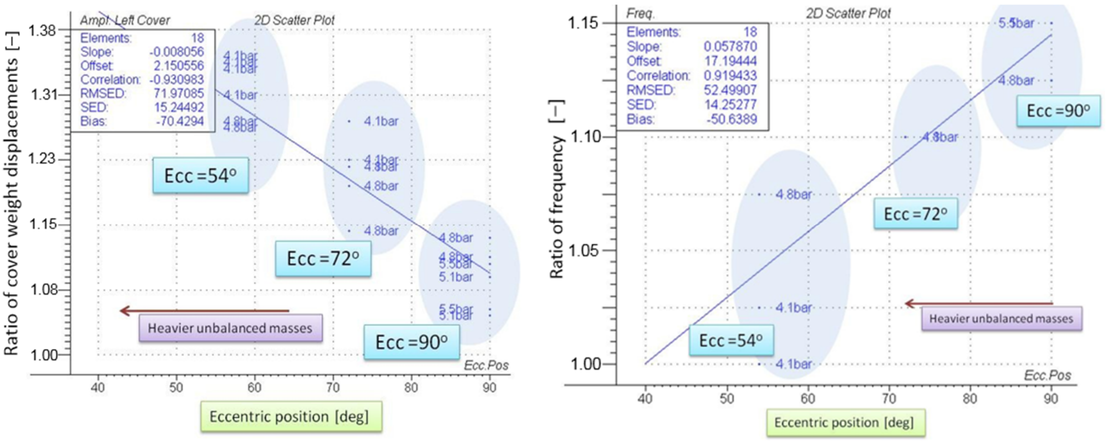

Figure 5. Measurements of vibrations by Fast Fourier Transform [FFT). The left plot shows the dynamic displacement of the cover weight as function of eccentric position and pressure in the bellows. The right plot shows the frequency of the compactor as function of eccentric position and pressure in the bellows.

From the FFT measurements, and using values from the displacements in the cover weight (Figure 5), the ratio, x_{anode}/f , increases by 30 % from ecc = 90 ° to ecc = 54 °. The FFT measurement is based on rms values of the dynamic displacements (an average value). But if the peak value of the signal is further away from the signal's rms-value at heavier unbalanced masses, the increase of the ratio would have been larger than 30% if it was based on peak values.

From the nonlinear model of the compactor set up in the software, SimulationX, the dynamic rms anode power (change in potential energy in the anode) was calculated for different eccentric forces and pressure in the bellows as shown in Figure 6. The spring- and damping coefficient of the anode paste, which are input parameters in the model, are unknown and will also vary during the compaction. These coefficients are also varying by the paste recipe and forming temperature. The spring- and damper coefficient was therefore set to values so that the model gave the same displacements and accelerations on table and cover weight as measured by FFT in the production line.

Figure 6 shows that there are two ways to increase the power on the anode from the vibrations. The power can be changed without changing the static force on the anode (by the eccentrics) and/or it can be changed by increasing the static force (changing the pressure in the pneumatic bellows). By the eccentrics, the power is changed by mainly increasing the dynamic displacements, but by the pneumatic bellows the power changes due to increasing the vibration frequency.

As an example, (stippled arrows in Figure 6), a compactor is running at 5 bar with and eccentric position, ecc = 126 ° (Figure 4). The frequency is then given by 23 Hz (left plot in Figure 6). The plant manager wishes to increase the eccentric to ecc = 108 ° without changing the static force (pneumatic bellows). From Figure 6, the anode power will increase by 24 %, even if the frequency has decreased with 5 %.

The model will never be completely accurate since different paste recipes and forming temperatures will change the spring- and damper properties of the anode. The model has no dependence to different paste properties. The relative values from the model (a change in a value) are therefore more accurate than the value itself (the example above found a change in the anode power).

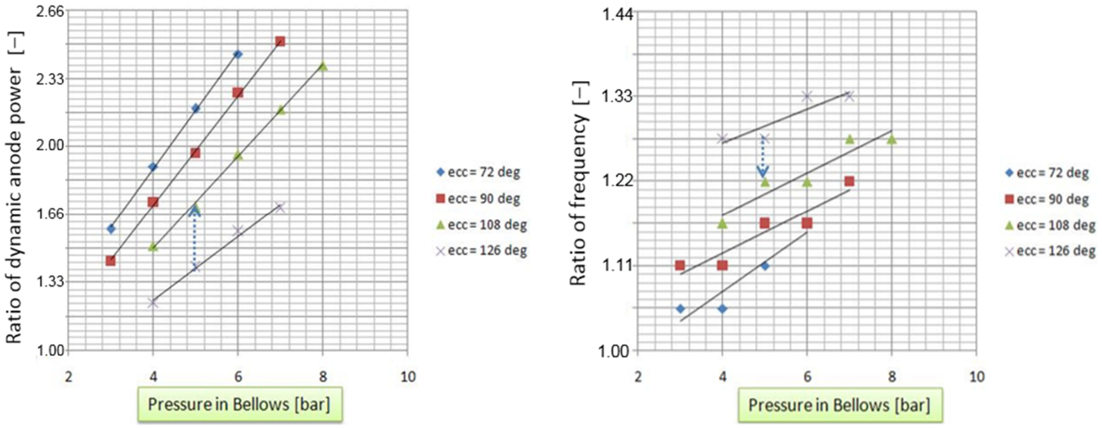

Figure 6. Results from nonlinear modeling of the compactor by the software, SimulationX. The left plot shows the anode power (change in potential energy in the anode) as function of eccentric position (from Figure 4) and pressure in the bellows. The right plot shows the frequency as function of the eccentric position and the pressure in the bellows.

4. Function of extra static force by preloading pneumatic bellows

The implementation of the pneumatic bellows is shown in Figure 1 as a spring, k_{bellow}, which are preloaded by a static force. The spring is realized with two pneumatic bellows on each short side of the compactor (Figure 2). The bellows is "squeezed" between the table and the cover weight and are placed outside the vacuum chamber. The static force can be adjusted by preloading an air pressure in the bellows from $1-6\ bar$. The pressure from $1-6\ bar$ is one of the set points of the vibrocompactor which can be set from an operator station. This set point affects the total vibration of the anode paste in different ways:

1) Increased pressure from the bellows increases the total static force on the anode paste (eq. 1)
2) Increased pressure from the bellows forces the set point of the vibration frequency upwards since more dynamic force has to be created and overcome the increased static force (right plots of Figure 5 and 6).
3) The dynamics of the cover weight are pushed further down into the paste. This means that the dynamic anode displacement becomes more equal to the total displacement between the cover weight and the table. The dynamic energy to the anode paste is increased (left plot of Figure 6)
4) The dynamic displacement of cover weight will experience a small reduction by increased pressure since more damping of the signal is realized when a greater part of the signal is forced into the anode paste (left plot of Figure 5).
5) The displacement of the table experiences a small increase by increased pressure. The passive table becomes "more active" when the pressure increases (vibration energy transfer through the spring, k_{bellow}). This is observed for lower eccentric forces (Figure 4) as long as the static forces do not overcome the dynamic forces. For higher eccentric forces and unbalanced masses the dynamic displacements are already higher and an increased pneumatic pressure does not necessarily increase the displacements further for the table. This

is shown in Figure 7. The displacements in the table increase only here by heavier unbalanced masses and do not systematically increase by pressure.

The pneumatic bellows can also be used to prevent instability. In a complex structure, as the vibrocompactor, there are hidden several resonance frequencies (a known resonance frequency of the compactor is the resonance frequency in the damping shoes (isolators) of the compactor which will be discussed in the chapter 5). If the vibration frequency coincides with one of resonance frequencies, instabilities can occur.

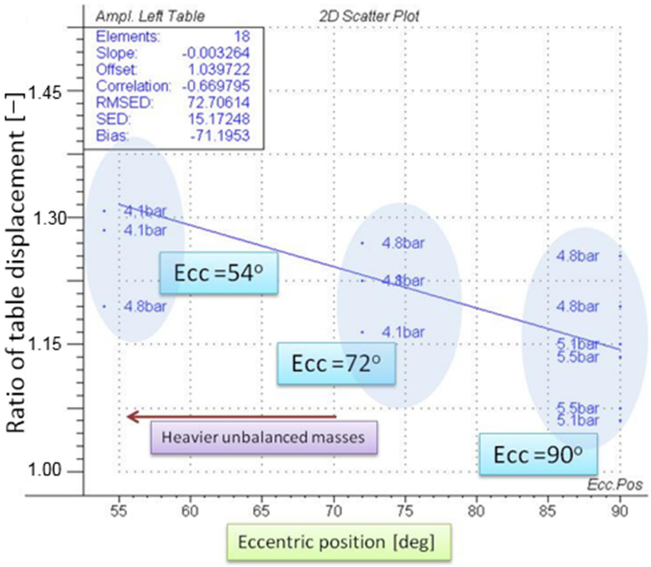

Figure 7. Measurements of vibrations by Fast Fourier Transform [FFT). For each eccentric position (with the new eccentric unit), six anodes with different pressure in the bellows were measured during compaction. The figure shows the dynamic displacement of the table as function of eccentric position and pressure in the bellows. The belonging frequencies and cover weight displacements are shown in Figure 5.

5. Function of reduced transmitted dynamic energy to environments

Vibrations from a vibrocompactor will also be transmitted to the environments, like vibrations to different parts of the building where the compactor is installed. The transmitted forces are mainly reduced by decreasing the stiffness in the damping shoes (isolators) of the compactor which makes them softer (Figure 3). The dynamic force through a spring is given by equation 2 as

$$F = k_{isolator} x_{table} \tag{2}$$

where $k_{isolator}$ is the stiffness and x_{table} the dynamic displacement of the spring which is the table displacement. The relation between the stiffness and the resonance frequency of the isolators are given by equation 3 for a “spring-mass-damper” system where the total compactor mass is m_{tot}.

$$f_{res} = \frac{1}{2\pi} \sqrt{\frac{k_{isolator}}{m_{tot}}} \tag{3}$$

From equation 2 and 3 it is seen that the transmitted force decreases also by a reduced resonance frequency of the isolators. A compromise of transmitted forces and softness of the isolators must be considered.

It is important for the compactor to run at frequencies far above the resonant frequency of the isolators. As a rule of thumb, the vibration frequency should be more than 4-5 times away from the resonant frequency to avoid large dynamic displacements from lower frequency components. The vibration on the resonant frequency of the isolators is an “in-phase” vibration, where both the cover weight and the table vibrate at the same phase (as one big mass).

The gain of the “in-phase” vibration for a second order “spring-mass-damper”-system will be reduced by 20 decibels pr. decade for frequencies at the $4Hz$-area to the $40Hz$-area. This means that the “in-phase” displacements are 1/10 in the $40Hz$-area compare to vibration below the $4Hz$-area. This is one of the reasons for not setting the vibration frequency too close to the resonance frequency of the isolators.

6. Typical production data

Anodes quality is an important factor in the electrolysis process and deviations in qualities is disturbing the process. Big variations in quality is reducing the possibilities to tune up the output of the posts. It is therefore important to have focus on the standard deviations in the production. Typical results from Årdal Carbon plant is presented below.

The statistics are given for all the product AHO 1650 at compactor C2650 from May 2015 to May 2016. The number of anodes are 20104.

Explanations:
- Green lines are Target
- Red lines are upper- and lower specification limits

Figure 8 is weight measurement of anodes. Target is set to 1165 kg. Mean weight is 1164,6 kg and 1 standard deviation is 8,07 kg. The deviations are relative high and are caused by inaccurate feeding system. Two compactors have common feeding from a sliding belt conveyor under the Eirich mixer. The feeding system is therefore important for the anode quality.

Figure 8. Weight of a specific product produced at compactor C2650 in Årdal from May 2015 to May 2016.

Figure 9 is GAD for the specific product in same period. The GAD is very much influenced by the weight deviations and also here the feeding system is very important.

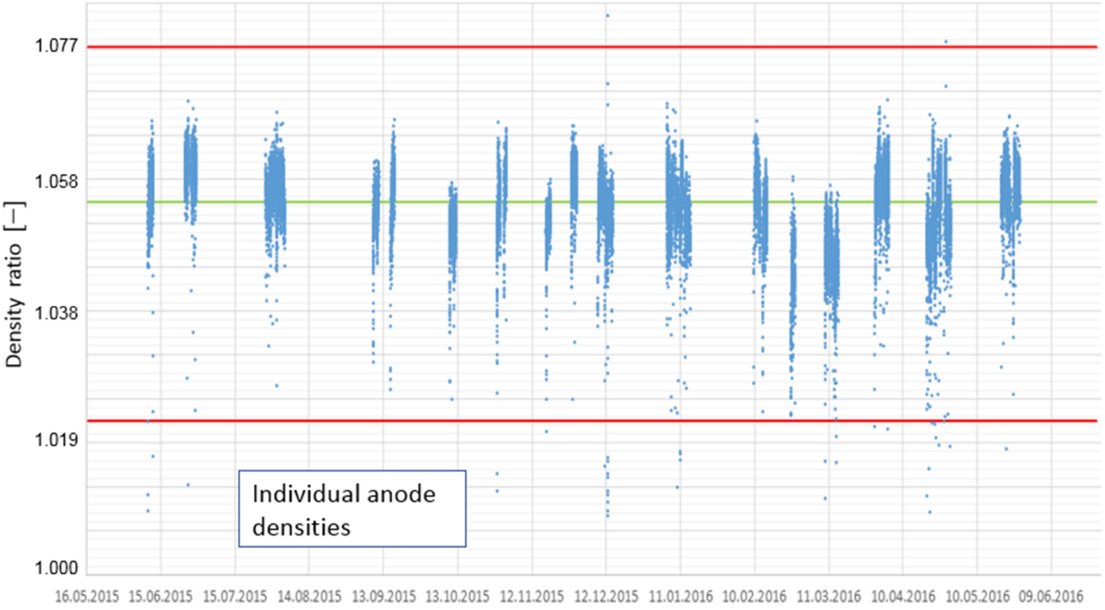

Figure 9. Green density of the specific product produced at compactor C2650 in Årdal from May 2015 to May 2016.

Figure 10 shows the height of the product in the period.

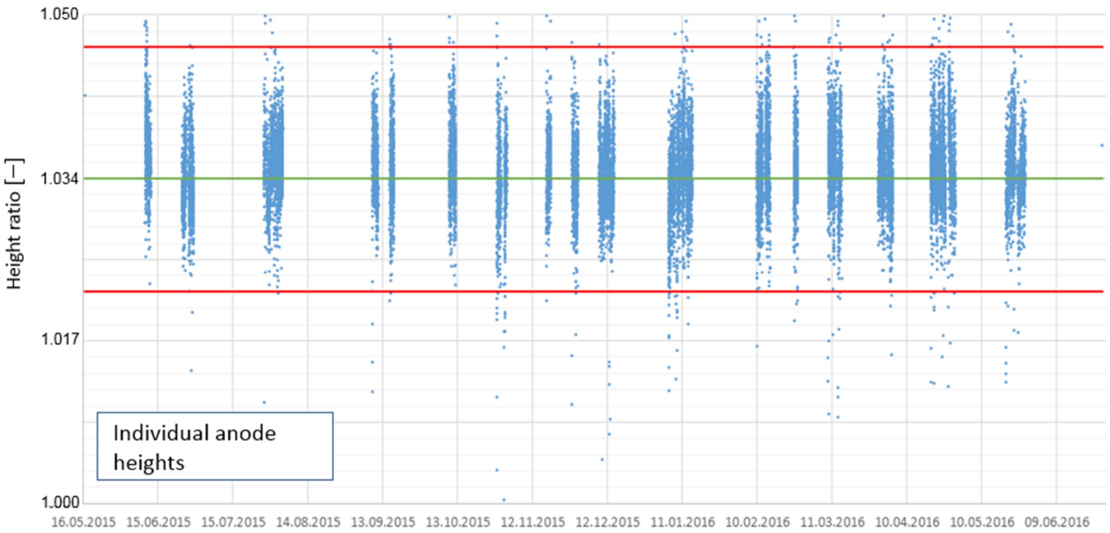

Figure 10. Height of the product produced at compactor C2650 in Årdal from May 2015 to May 2016.

Figure 11 is showing the vacuum in the mould in the period. The upper specification limit of vacuum is set to 60mbar. The first 4 anodes are anyway rejected in a startup (after a longer stop). Among these are a number of the anodes, which have higher values than 60 mbar. A total of 99,4% of all the anodes (produced at C2650 May2015-May 2016) in the curves have vacuum-values below 60 mbar (see Figure 11).

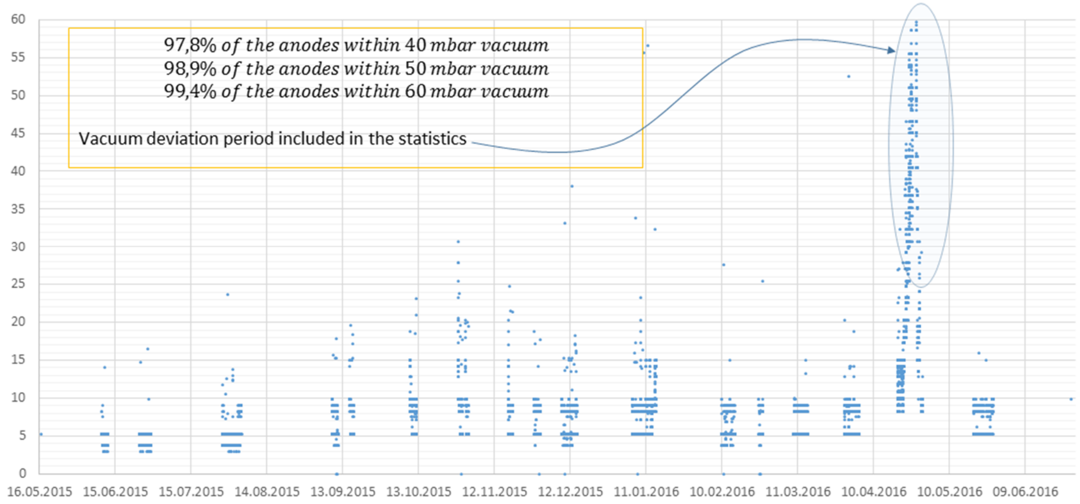

Figure 11. Vacuum [mbar] of the specific product produced at compactor C2650 in Årdal from May 2015 to May 2016.

7. Productivity, reduced maintenance and increased availability factor

The HAL compactor is designed for high productivity. The effective capacity consists of maximum production capacity and an availability factor.

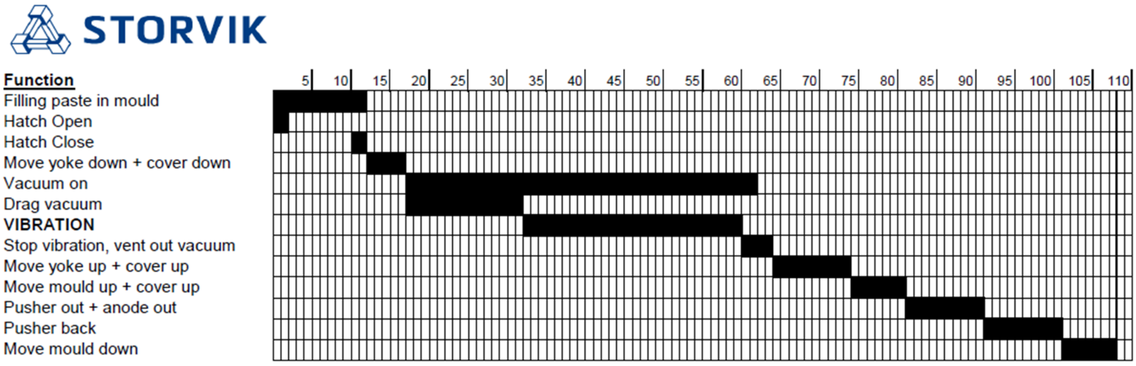

Figure 12. Typical sequences for HAL compactor.

This sequence is a typical running sequence for a production rate of 33 anodes per hour. A standard paste plant has capacity of 35 t/h or 60 t/h and it is therefore necessary to have a compactor capacity at 30 – 40 anodes per hour. The capacity of 40 anodes can be reached with optimizing the hydraulic movements and other functions.

The availability factor is important for paste plant machinery to achieve sufficient production capacity. Figure 13 and 14 is showing unplanned stops and reasons for these stops at two compactors in Årdal in the period 2010 – 2014. This gives an availability factor at 99,5% over these 5 years.

An important factor to achieve high availability factor is the complexity of the machine. It is therefore a goal to reduce the number of moving parts and make the machine even more robust than today. "Uninstalled equipment will never brake". Maintenance is another important factor and most of the unplanned stops can be avoided by regular and correct maintenance. It is also important to have a maintenance friendly design to reduce the maintenance cost.

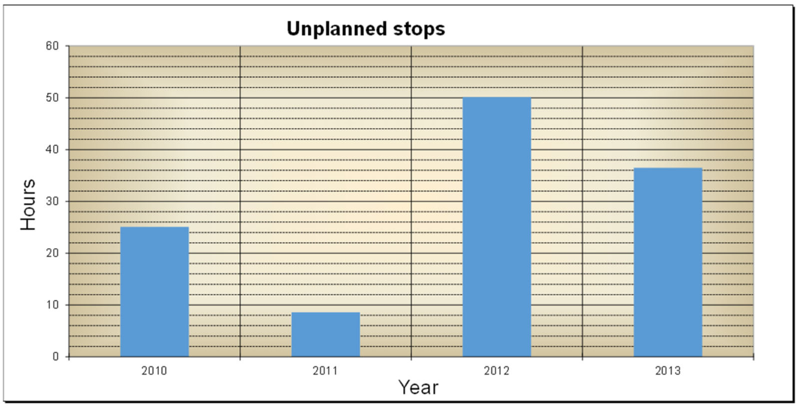

Figure 13. Example of Unplanned stops at a Hydro plant

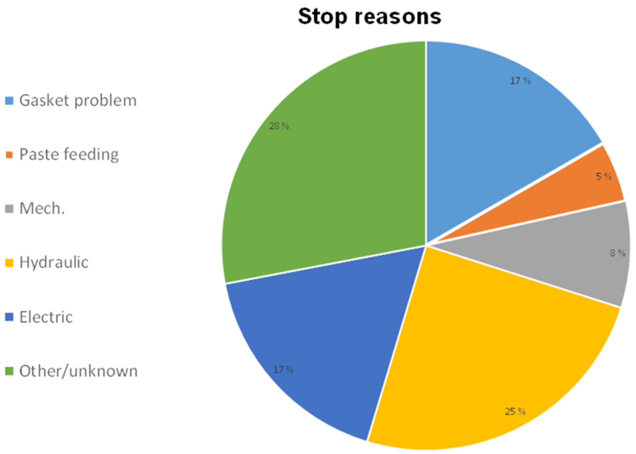

Figure 14. Example of reasons for unplanned stops at a Hydro plant.

8. Conclusions

The HAL vibrocompactor where the cower weight is the active mass, has been developed over the years where new functionality has been added as vacuum forming, extra static force with pneumatic bellows outside the vacuum chamber, modified eccentric forces and isolators to foundation. The dynamic input force comes from robust hydraulic motors which is small enough to follow the vibration (fixed to cover weight vibrations).

Both vibrational FFT measurements during anode compaction and nonlinear transient compactor models have been used as tools in the development of the compactor.

The compactors have run for several years at different Hydro paste plants and have shown to achieve high densities at low compaction times. Less vibration sequences, smaller vacuum chamber to reduce suction time and higher cover weight amplitudes (since this mass is active) makes the single-mould-compactor to achieve high anode capacities in a paste plant line.

9. References

[1] Patent no. NO132359B: Vibratory device for creation of mould bodies for the production of anode and cathode blocks for the melting industry, especially the aluminium electrolysis industry.

[2] Patent no. WO 03/068468 Al: A method and equipment for compacting materials