Earth Sciences
2019; 8(1): 20-44
http://www.sciencepublishinggroup.com/j/earth
doi: 10.11648/j.earth.20190801.13
ISSN: 2328-5974 (Print); ISSN: 2328-5982 (Online)

Energy Recovering System for Moving Bulk Materials

Michael Prenner

Department of Mineral Resources Engineering, Montanuniversität Leoben, Leoben, Austria

Email address:
michael.prenner@unileoben.ac.at

To cite this article:
Michael Prenner. Energy Recovering System for Moving Bulk Materials. *Earth Sciences*. Vol. 8, No. 1, 2019, pp. 20-44.
doi: 10.11648/j.earth.20190801.13

Received: October 23, 2018; **Accepted**: January 11, 2019; **Published**: February 21, 2019

Abstract: Bulk materials, which are transported on continuous conveyors, partly have a high energy content, depending on the specified mass flow and the conveying velocity. At discharge points to a storage area or at transfer points from one conveyor to another, the energy content often increases due to the elevation of the discharge conveyor. At these points it is possible to recover a large part of the energy due to the mass flow (conveying velocity) and the drop height of the bulk material. This energy is usually converted into "wear" of the conveying system or the bulk material at discharge or transfer points. Furthermore, it is available free of charge and could be used to achieve more environmentally friendly continuous conveying systems. This research paper is focused on a new method which has been developed and patented by the "Chair of Mining Engineering and Mineral Economics - Conveying Technology and Design Methods" at the Montanuniversität Leoben / Austria. This invention makes it possible to recover a large part of the above mentioned energy. The invented so-called "Solid State Material Driven Turbine" allows the recovery of this energy directly to the conveying system using a traction drive, or to the electric circuit using a generator. The paper describes the new method and presents turbine prototypes that have been designed using simulations and tested under laboratory condition and in operational trials. Additionally, a discussion concerning the costs and economical aspects of the invention is included. For a special application of such a turbine a permanent magnetic safety coupling can be used. First test results of such a coupling are presented. The paper also includes layout criteria for an overshot "Solid State Material Driven Turbine". All executed experiments showed, that a recovery of energy from moving bulk materials using a "Solid State Material Driven Turbine" is possible. An efficiency of more than 50% can be realised. The occurred challenges during the tests phase under real conditions can be managed effortlessly.

Keywords: "Solid State Material Driven Turbine", Bulk Material, Energy Recovery, Continuous Conveyor

1. Introduction

Bulk materials, transported on continuous conveyors, partly have a high energy content, depending on the specified mass flow and the conveying velocity. For instance, belt conveyors can handle a mass flow up to 40 000 t/h with a conveying velocity up to 15 m/s [1]. At discharge points to storage areas or at transfer points from one conveyor to another, the energy content often increases due to the elevation of the discharge conveyor. It is possible to recover a large part of the energy due to the mass flow (conveying velocity) and the drop height of the bulk material at these points. At the "Chair of Mining Engineering and Mineral Economics - Conveying Technology and Design Methods" at the Montanuniversität Leoben/ Austria a so-called "Solid State Material Driven Turbine" [2] has been developed, which allows recovery of this energy. This energy could be transferred directly to the conveying system or to the electric circuit. In order to illustrate the potential of such an energy recovering system, a simple calculation can be carried out. A mass flow of 40 000 t/h (11 111 kg/s) at a conveying velocity of 8 m/s with a drop height of 2 m leads to a power content of about 574 kW, which is stored in the moving bulk material. This power content is equal to a small hydro power plant [3]. Usually this energy or power will be converted into "wear" of the conveying system or the bulk material at these discharge or transfer points. This unused energy is available free of charge and could be better used to achieve more environmentally friendly continuous conveying systems. Energy recovery for continuous conveying systems is a well-established method to improve the efficiency of these systems. Usually it is only possible to recover energy during downwards conveying. The potential energy of the bulk material is used for recovery. For example, belt conveyors use regenerative braking during a downwards conveying process [1], [4]. For this process a contact between the bulk material and the conveyor is necessary. The new developed system recovers energy from the moving bulk material after leaving the conveying system. The functional principal is similar to simple hydraulic turbines, but instead of liquids bulk materials are used as an energy source. In a comprehensive literature research, which was carried out previous to the project start, no comparable publications could be found. In the meantime De Graaf from the Delft University of Technology in the Netherlands has published a research work [5], which is based on an early stage publication on this topic, written by Prenner and Kessler from the Montanuniversität Leoben [6]. The publication written by De Graaf critically questions the new technology. Meanwhile, several new research results have been generated by Prenner from the Montanuniversität Leoben. The following article will give an introduction into this topic. Further publications based on wear behavior, turbine types, economy and additional benefits of "Solid State Material Driven Turbines" are worthwhile for further research and publication (e.g. [7]).

2. Operation Theory of a "Solid State Material Driven Turbine"

The functional principle of a "Solid State Material Driven Turbine" is similar to a simple hydraulic turbine. Especially the principles of water wheels (overshot, undershot, breast shot or pitch back - Figure 1 [8]), cross flow turbines (Ossberger turbine - Figure 2 [9]) or horizontally mounted water wheels ("Stoßrad" - Figure 2 [10]) could be adapted. The main difference between a water turbine and a "Solid State Material Driven Turbine" is the wear behaviour. Bulk materials induce significantly more wear to the turbine than water. The turbine blades have to be lined with wear plates. These necessary wear plates are difficult to machine and have to be replaced frequently. The geometry of the turbine blades should be constructed to be as simple as possible to reduce the operational costs. A simple construction with highest efficiency decreases the payback period.

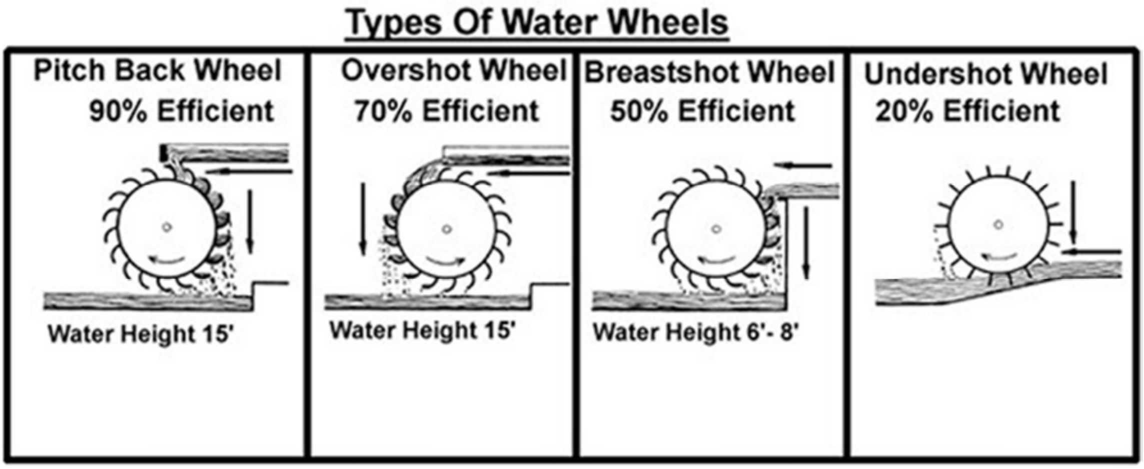


Figure 1. *Types of water wheels [8].*

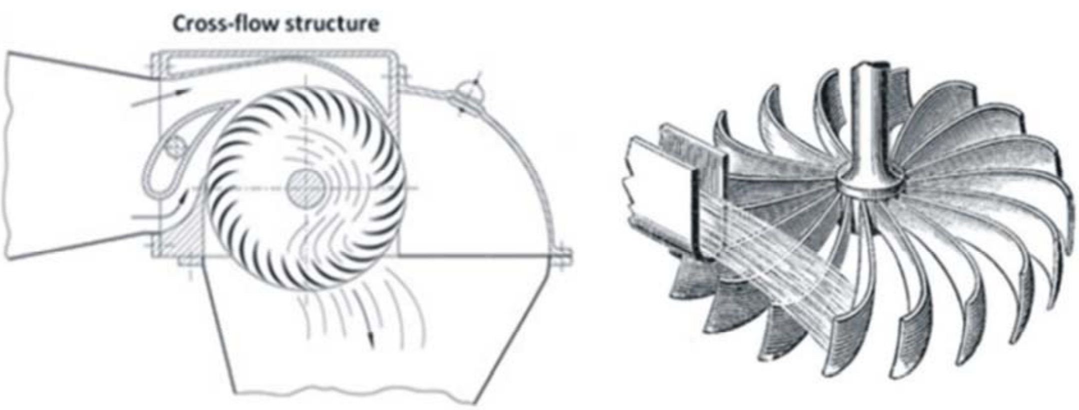


Figure 2. *Crossflow turbine – "Ossberger turbine" (left side) [9], horizontally mounted water wheel – "Stoßrad" (right side) [10].*

The "Discrete Element Method" (DEM) [11] was used for the design and calculation work. This simulation method was developed to calculate the movement of a large number of particles and is predestined to calculate bulk material movements. Figure 3 shows a simulation example of an undershot "Solid State Material Driven Turbine".

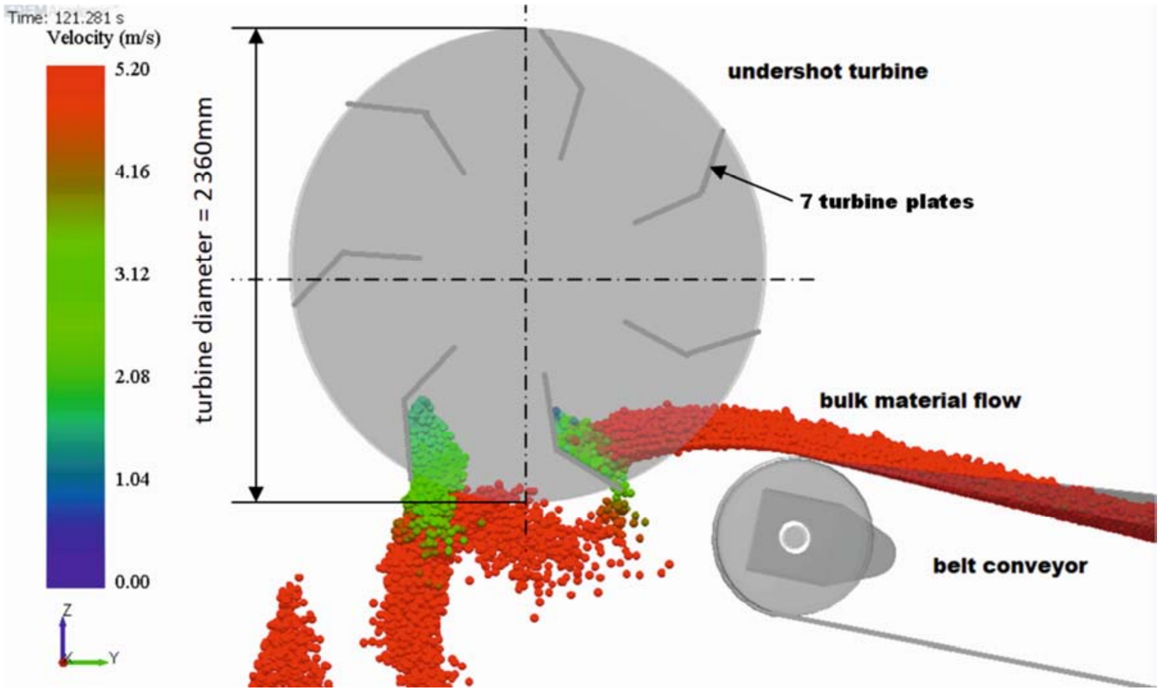


Figure 3. *Simulation example of an undershot "Solid State Material Driven Turbine".*

A mass flow of 1 600 kg/s was used for this simulation. At a turbine diameter of 2 360 mm, a driving speed of 18 rpm and a belt speed of 5.2 m/s, a power output of 5.5 kW was calculated. The power content of the bulk material is 21.6 kW in that case. This leads to an efficiency of about 25%, which is similar to an undershot water wheel (see Figure 1).

3. First Functional Test of a "Solid State Material Driven Turbine"

To get a better idea whether the principle works or not, a small laboratory overshot turbine was designed. Figure 4 shows the turbine behaviour in a simulation.

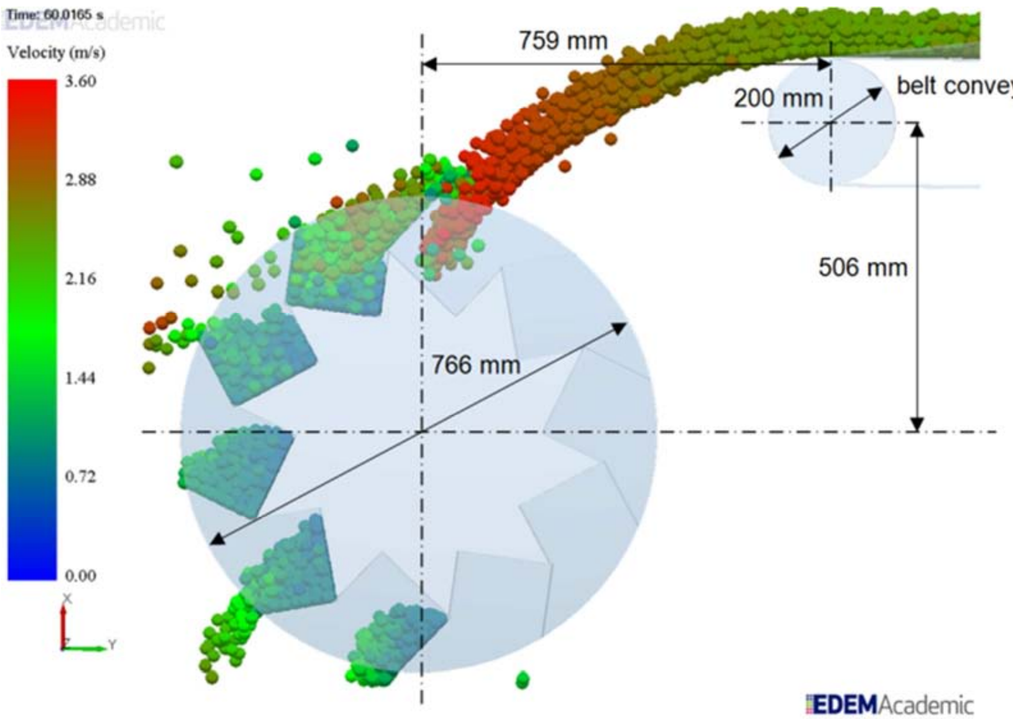


Figure 4. *Simulation of the laboratory turbine.*

The maximum power output of such a turbine depends on the mass flow, the belt speed and the drop height, but is also influenced by the rotational speed of the turbine. Figure 5 depicts the correlation between the rotational speed and the power output for a mass flow of 28.125 kg/s and a belt speed of 2.5 m/s by simulation. The geometric data are shown in Figure 4.

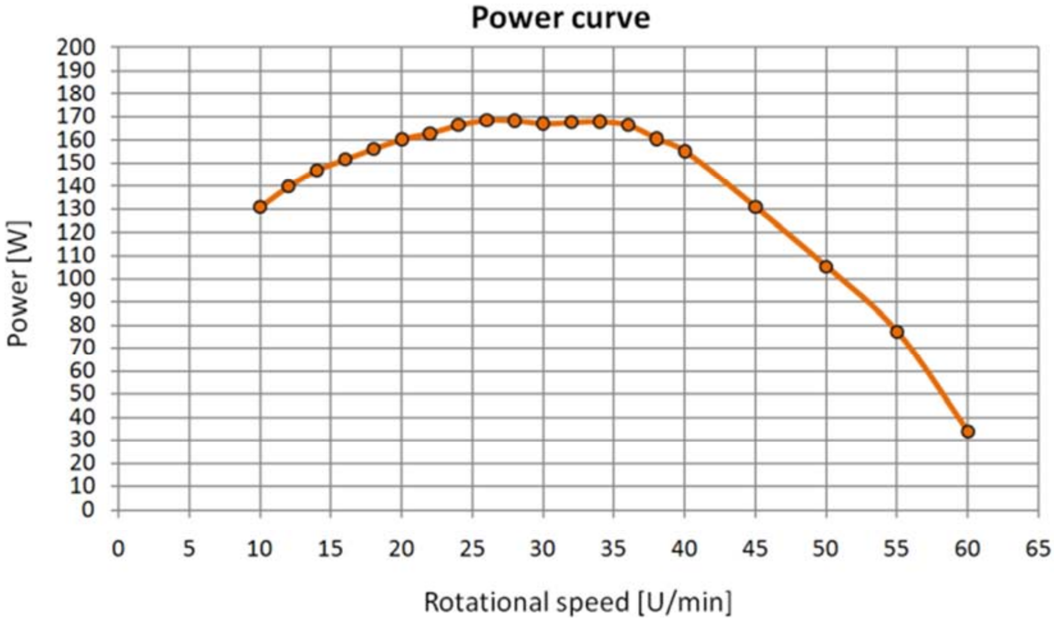


Figure 5. *Power curve of the laboratory turbine at a mass flow of 28.125 kg/s [12].*

A real turbine was built and installed in a belt conveyor circuit at one of four transfer points (Figure 6), to verify the simulation results. This simple test turbine was made out of 3 mm sheet steel without any wear plates. The recovered energy was directly transferred to the discharge pulley of one of the four belt conveyors via a chain drive.

Figure 6. *First test turbine [12].*

The power content of the moving bulk material (crushed stone), which includes the power content due to the discharge velocity and the drop height (turbine outlet) is 362 W. The measured power output of the test turbine was 152 W. This leads to an efficiency of about 42% at 20.1 rpm. The difference between the actual test (152 W) and the simulation (163 W) is based on losses due to the bearings, the chain drive, the gear box and the electric motor. The power consumption of the electric motor used at the discharge belt could be reduced from 928 W to 776 W, which is a reduction of about 16.4%.

4. Long-Time Test Under Operation Conditions

To test the applicability of a "Solid State Material Driven Turbine" it was decided to carry out a long-time test under operation conditions. For this long-time test the company W&P Zement GmbH provided a transfer point for lime stone from a belt conveyor to a crusher, where it was relatively simple to install a turbine in an existing plant. Figure 7 illustrates the position of the turbine at this transfer point.

Figure 7. *Transfer point from a belt conveyor to a crusher.*

The specified maximum mass flow of this discharge conveyor belt was 400 t/h at a belt speed of 1.6 m/s. The grain size distribution for the conveyed lime stone was specified between 10 mass percent smaller than 4 mm and 10 m% bigger than 70 mm with a maximum of 300 mm. Due to the available space, it was necessary to design a breast shot turbine with less efficiency. However, the efficiency was not the main focus for the planned long-time test. Focuses were on feasibility and durability. Due to the critical grain size distribution and the very cohesive material conditions of fine grained lime stone, for such an energy recovering system, the chosen installation position was challenging.

4.1. DE - Simulation of the Lime Stone Turbine"

By using the DEM, a suitable "Solid State Material Driven Turbine" was designed. A simulation of the final design is shown in Figure 8.

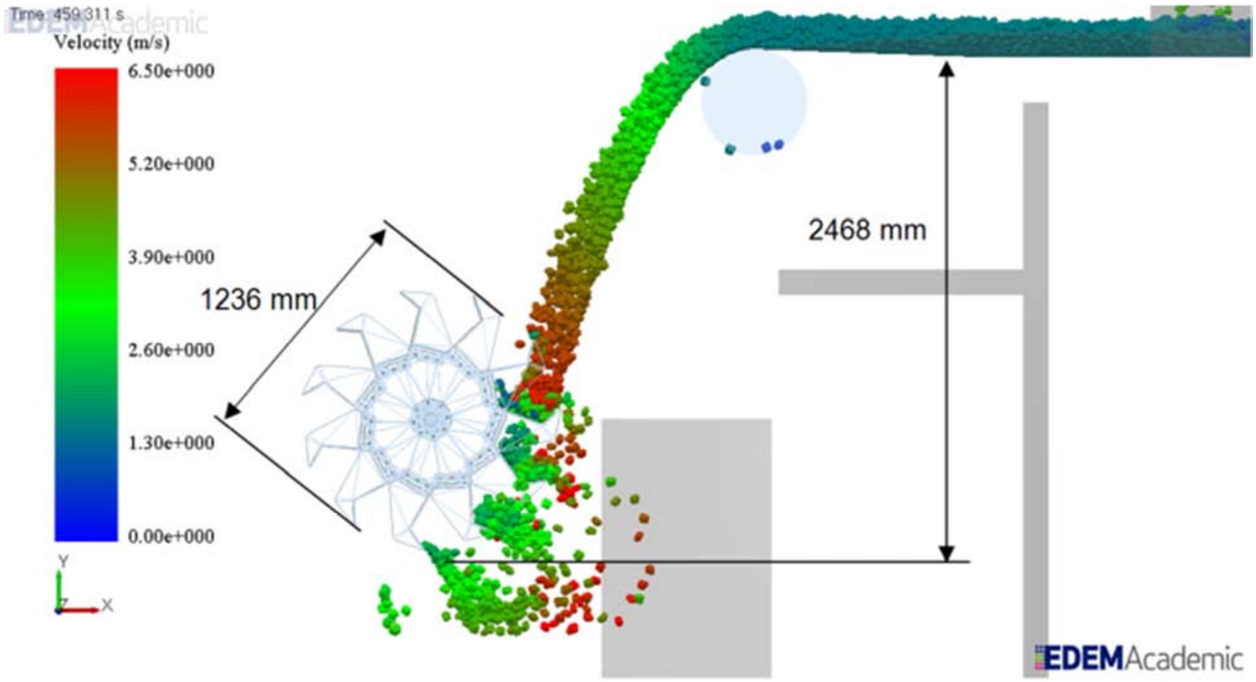


Figure 8. *Simulation of the final turbine design [13].*

According to the simulation a maximum power output of 1 219 W at 37.9 rpm and 307 Nm torque (Figure 9) without any losses should be expected. The power content of the bulk material at the lowest point of the turbine is 2 832 W, which implies an efficiency of 43%.

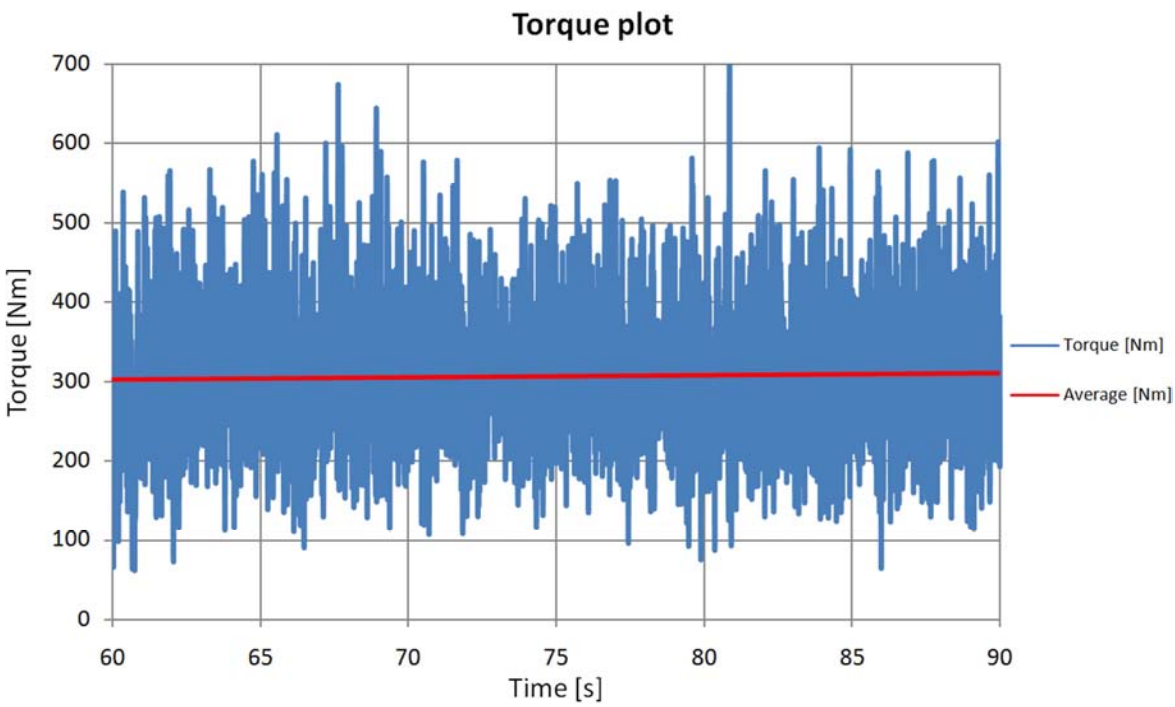


Figure 9. *Simulation result - torque at 37.9 rpm.*

4.2. Turbine Construction and Installation

After the design phase using DEM, the turbine was calculated, engineered, manufactured and installed by a team of scientists and students of the "Chair of Mining Engineering and Mineral Economics - Conveying Technology and Design Methods". In that case the recovered energy or power was also directly transferred from the turbine to the discharge pulley by using a chain drive. To transfer the energy via a chain drive it was necessary to install an additional single-stage spur gear box, because the rotating direction of the turbine was clockwise and counter clockwise for the pulley (Figure 10). Compared to an overshot construction with the same rotating direction of the turbine and the pulley, further losses due to this gear box have to be expected. As already mentioned, focuses were on feasibility and durability. Steel wear plates (Hardox 400) were used, to increase the durability of the turbine blades. Lime stone causes significant less wear compared to iron ore for example, so that wear of the turbine blades should be less of a problem. The power output of the turbine was measured by two semi-bridge strain gauges (HBM - Type 3/350 XY41), each of which was positioned on the opposite side of the turbine shaft between the turbine and the gear box. A single channel data logger with a rechargeable battery from the company PJ Messtechnik GmbH was used for data recording. The torque measurement system was calibrated by a 50 litre barrel filled with water, which was mounted at the outside edge of a turbine blade. The static torque could be measured and calibrated by taking into account the barrel mass and the turbine radius.

Figure 10. *Test turbine during the manufacturing and installation phase [12].*

4.3. Test Phase

The tests were carried out over two months with remarkable results. The turbine was demounted after this period, because a belt conveyor with higher capacity had to be installed. At maximum mass flow of 420 t/h (20 t/h beyond the specification), which was measured by the plant operator, a maximum average turbine torque of about 200 Nm was measured. A corresponding torque plot is shown in Figure 11. The rotational speed of the turbine was pre-set by the ratio of the chain drive. This speed of 40.6 rpm was checked with a high speed camera. The torque of 200 Nm at a rotational speed of 40.6 rpm led to a power output of 850 W.

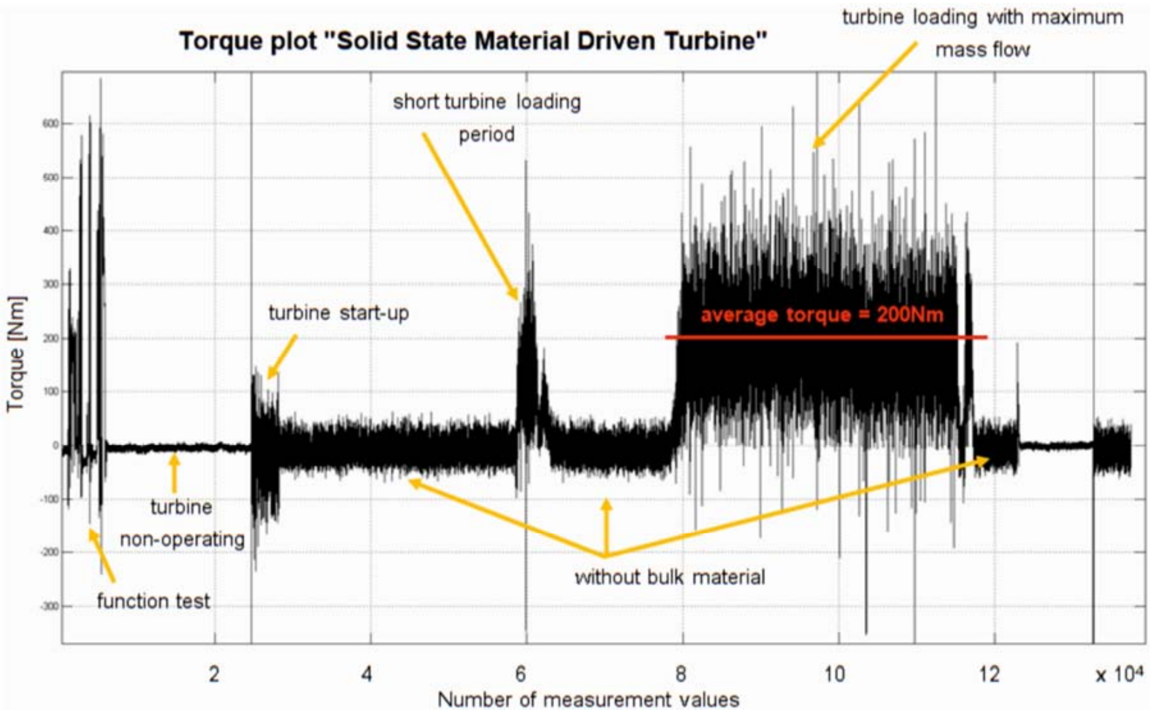


Figure 11. *Average torque at maximum mass flow [12].*

The measured power output is significantly lower than the prediction by the DEM - simulation. The reason for this behaviour is a grain size distribution which differs from the specification. Especially the percentage of the coarse-grained particles was much higher than the specification. Additionally the maximum grain size was higher than the specified 300 mm. Figure 12 illustrates the actual grain size distribution of the conveyed lime stone. Furthermore a particle with a grain size way more than 300 mm can be seen in Figure 12. During the impact against the turbine blades this particle broke into pieces. The course grained particles have a different trajectory than the fine grained particles and could not be correctly received by the buckets. A torque against the rotational direction of the turbine was induced by the large grains.

Figure 12. *Actual gain size distribution [12].*

4.4. Further Simulation of the Turbine

After analysing the actual grain size distribution a further DEM - simulation with the appropriate distribution was carried out (Figure 14). The simulation also shows the same behaviour as the actual test (Figure 13 with Figure 11). A maximum power output of about 850 W could be calculated. The turbine was designed for the specified and not the actual grain size distribution. The actual grain size distribution differed from the specified, because the mining conditions in the lime stone pit changed during the installation of the turbine. The grain size distribution changes were not communicated by the plant operator.

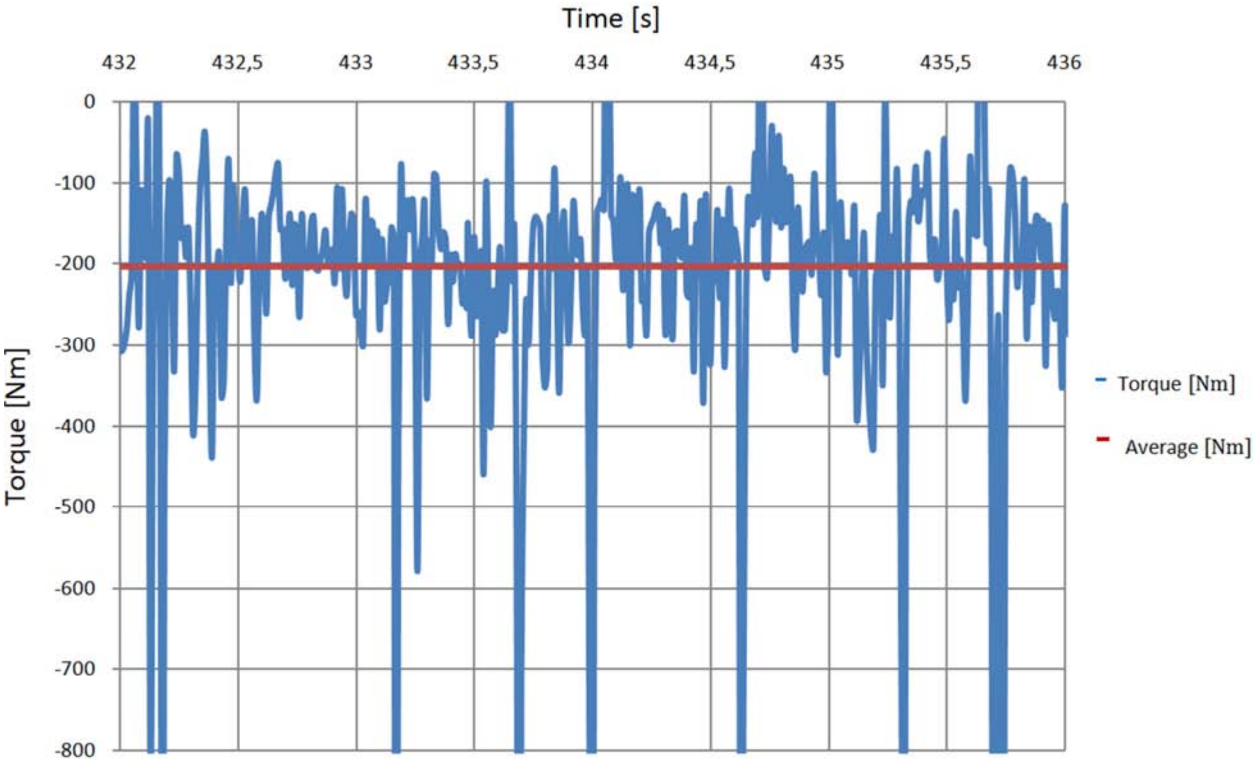


Figure 13. *Torque with actual grain size distribution.*

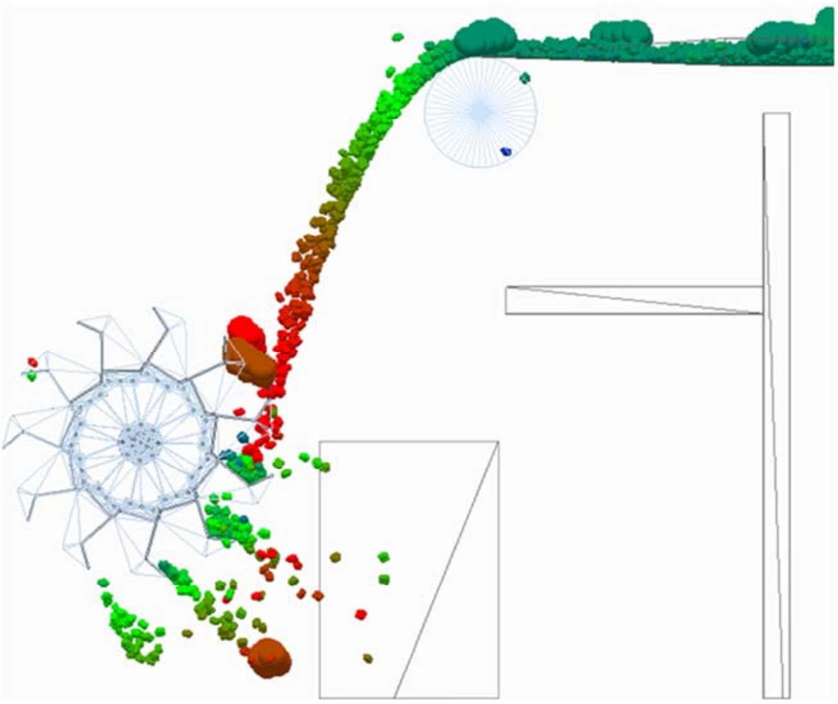


Figure 14. *Simulation result with actual grain size distribution [12].*

4.5. Occurred Challenges During the Test Phase

In addition to the lower power output, the large number of grains bigger than 70 mm (more than 10 m%, rather 50 m%) caused additional challenges at the turbine. Figure 15 shows the challenges encountered at the turbine during the two month operation. The main issue was the loosened bolts at most of the eleven turbine blades. Due to the higher impact force, based on the high number of large particles, the bolting lost its pre-load whereby the threaded holes were deformed. Some of the bolts got totally lost and had to be replaced at certain time intervals. Due to the deformed threaded holes, the problem got even worse, so that the bolts had to be tightened once a week.

The weld seam breakages at the turbine blades were mainly caused by the different steel grades used in combination with the coarse grain contacts. The wear plates of the buckets were made out of 10 mm Hardox - steel wear plates and the side plates of the buckets out of 5 mm structural steel. Both parts are weldable, but the material combination was not suitable for the predominant load. Both parts should be manufactured out of 10 mm Hardox. The broken sealing slips were also made out of structural steel and should also be replaced by Hardox. The sealing slips had no support function and caused no failure of the turbine. All other components of the turbine operated without any problem. No wear at the turbine blades could be detected during the two month operation. As expected, the limestone did not cause any wear problems. The expected clogging problems due to the cohesive fine grained particles of the limestone were also not critical. Only during rain periods, which caused wet bulk material conditions, the fines started to stick at the turbine blades, but were knocked off by the coarse grained particles over a short period of time.

The turbine geometry has to be slightly redesigned for the actual grain size distribution of the conveyed lime stone. It is necessary to increase the bucket size and thereby the turbine diameter. Thus, it is possible to pick up the coarse grained particles without any further losses or a torque affecting the rotational direction of the turbine. The bucket bolting must also be changed. The turbine blades may need to be strapped so that they can support each other. The necessary improvements could easily be achieved. An increased turbine diameter will also slightly increase the power output of the turbine. This will compensate for the higher costs due to the increased turbine size.

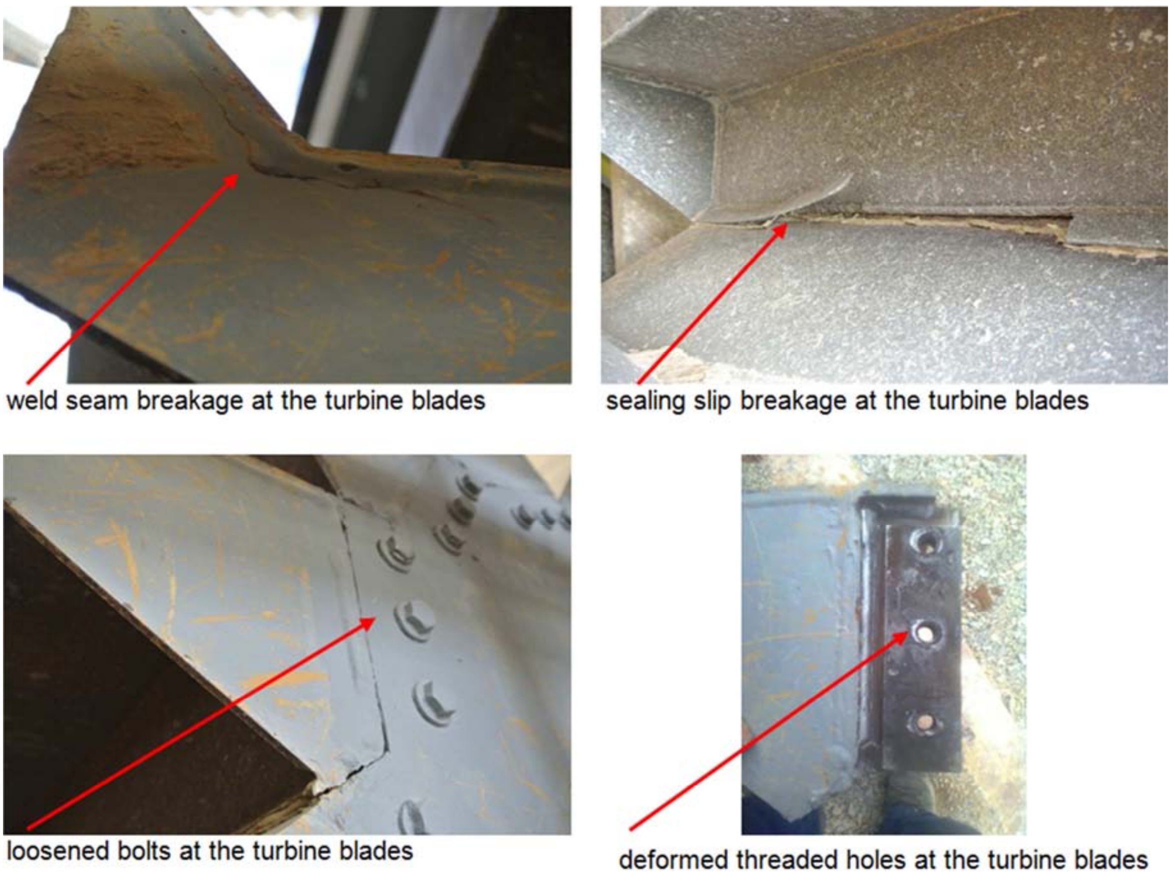


Figure 15. *Additional challenges due to the unspecified grain size distribution [12].*

5. Turbine Costs and Payback Period

The turbine which is presented in chapter 4 was simulated, calculated, designed, manufactured and installed by employees of the Chair of Mining Engineering and Mineral Economics - Conveying Technology and Design Methods.

The costs for the turbine excluding the installation costs were 38 465 Euros and included an overhead of 87.51% for the hourly rate of the employees involved. For the installation about 160 man hours were necessary. With an hourly rate of 50 Euros (locksmith) an additional cost of about 8 000 Euros was necessary. The total costs for this energy recovering system prototype were about 46 465 Euros. The daily switch-on time for the conveyor relating to the turbine was 12 hours seven days a week. With a maximum turbine power output of 850 W a maximum of 3 672 kWh (five days maintenance per year) could be recovered every year. Using 0.15 Euros energy costs per kWh, approximately 551 Euros energy costs could be saved. The payback period would be more than 85 years for this machine.

If such an energy recovering system prototype were implemented in a new conveying system with the same mass flow, drop height and a non-critical grain size distribution, the price could be significantly reduced (Table 1). The turbine could be designed as an overshot turbine, thus no additional single-stage spur gear box for changing the rotational direction will be needed. There is also no requirement for a safety coupling, as used in the turbine prototype shown. The installation costs will also decrease, as the turbine is installed into a new conveyor plant. Table 1 shows the individual costs for such an operation case.

Table 1. *Costs for an 1.6 kW over shot turbine prototype [14].*

simulation costs	6529 Euros
design and calculation costs	6000 Euros
material costs	3600 Euros
manufacturing costs	8000 Euros
installation costs	1200 Euros
total costs	25329 Euros

Using an overshot turbine about 1.6 kW of power could be recovered. At the same daily switch-on time (12 h and 360 days a year) about 1 037 Euros worth of energy costs could be saved per year. The payback period would be about 25 years without maintenance costs. For such applications with low mass flows, a run production of the turbine is required to significantly lower the costs. A payback time of about 10 years would be realistic then.

However, such an energy recovering system was intended for much higher mass flows. Based on the costs for the turbine prototype from chapter 4, the costs for a 100 kW "Solid State Material Driven Turbine" were calculated (Table 2). The calculation for the material costs is based on the turbine surface which increases with a factor 10 compared to the 1.6 kW turbine. The manufacturing and installation costs will be increased by a factor of 5 because the complexity of the 100 kW turbine is similar to the 1.6 kW turbine. Only the dimensions increase. Simulation costs will be the same and the design and calculation costs will double. For the calculation of the costs, bulk materials without critical conditions will be required. Table 2 shows the costs estimation for a 100 kW turbine.

Table 2. *Costs for an100 kW over shot turbine prototype [14].*

simulation costs	6 529 Euros
design and calculation costs	12 000 Euros
material costs	36 000 Euros
manufacturing costs	40 000 Euros
installation costs	6 000 Euros
total costs	100 529 Euros

Using the same daily switch-on time (12 h and 360 days a year) as for the 1.6 kW turbine 432 000 kWh could be recovered every year. With the same electricity costs (0.15 Euros electricity rate and network fee) 64 800 Euros could be saved every year. This leads to a payback time of about 1.55 years without maintenance costs. The wear plates costs for the turbine prototype from chapter 4 are 822 Euros. Using the described increasing factor of 10, a change of all wear plates will cost about 8 220 Euros. The replacement interval of the wear plates depends on the wear behaviour of the conveyed bulk material and the wear plates used. For lime stone use an interval of once a year will be realistic. If, for example, iron ore is to be conveyed and Hardox steel wear plates are used, in the worst case a weekly change interval has to be expected [7]. The replacement costs will be much higher than the savings. For such conditions, Hardox will be the wrong wear protection material. Wear plates with higher wear resistance have to be used.

6. "Solid State Material Driven Turbine" in Combination with a Bulk Material Stockpile

A further possible site for such an energy recovering system would be a belt conveyor in combination with a bulk material stockpile. Figure 16 shows an open-air stockpile for sand and gravel which is excavated from the seabed.

Figure 16. *Stockpile for sand and gravel.*

The belt conveyor shown handles 1 000 t/h bulk material with a belt speed of 2 m/s. The drop height is 15.16 m. The energy which can be recovered mainly depends on the useable drop height. To avoid storage losses it is planned to spill the turbine with the bulk material. Figure 17 depicts the design geometry for this stockpile turbine.

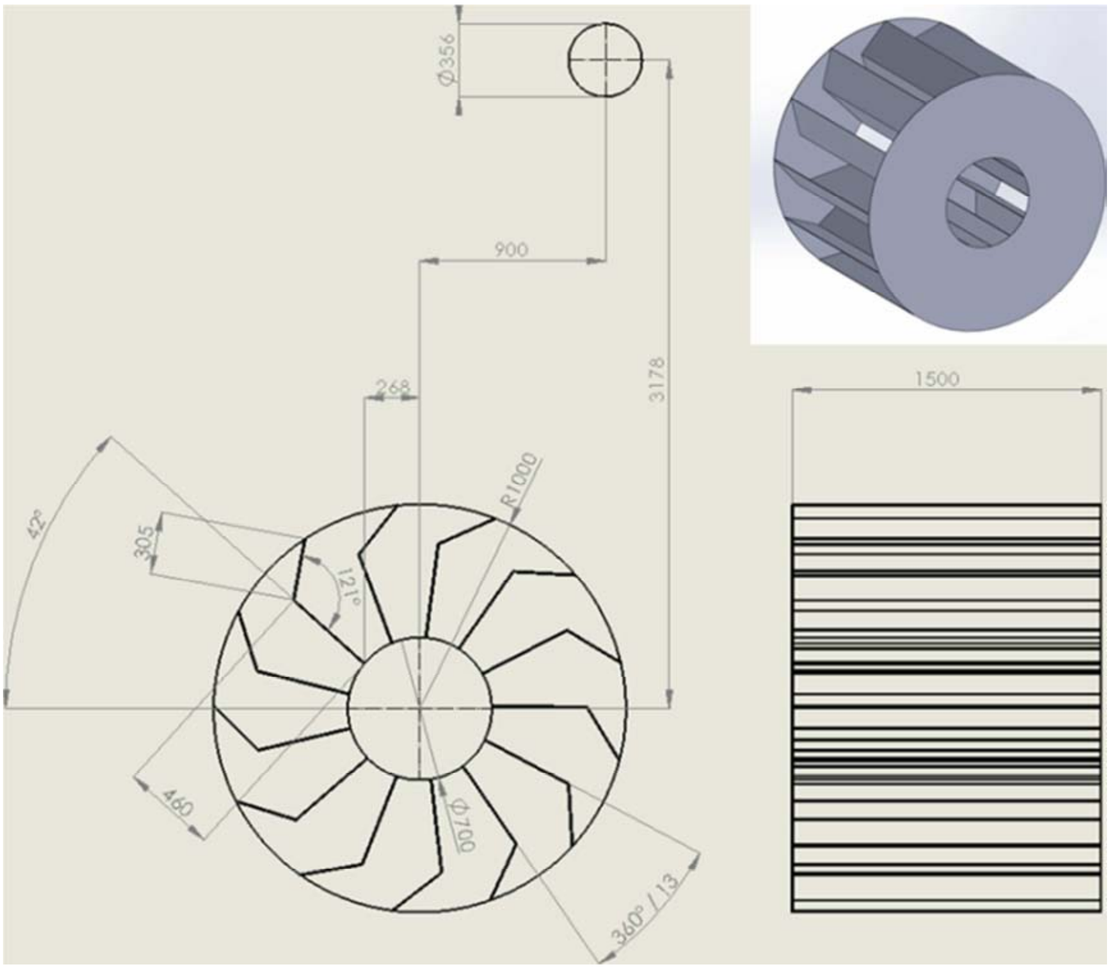


Figure 17. *Design of the stockpile turbine with the chosen drop height.*

An additional guide plate was introduced (Figure 18) to increase the efficiency of the turbine. Using a guide plate, undesired escape of particles out of the turbine, due to the bucket movement through the bulk material stream, could be avoided. The turbine and the guide plate geometry were developed via the DEM.

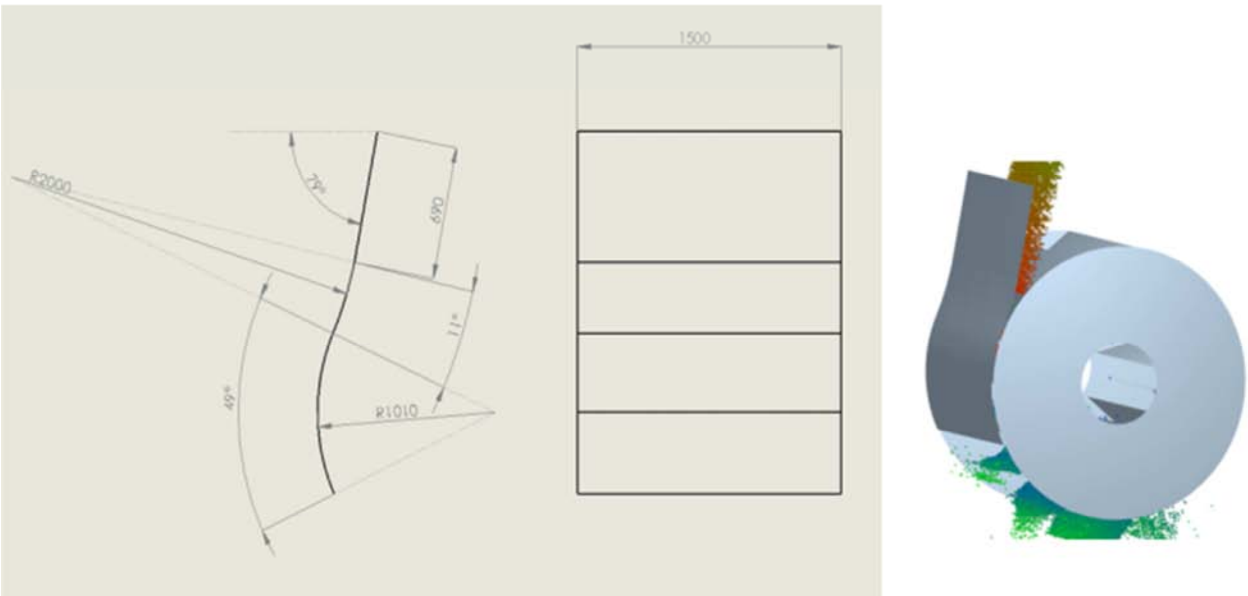
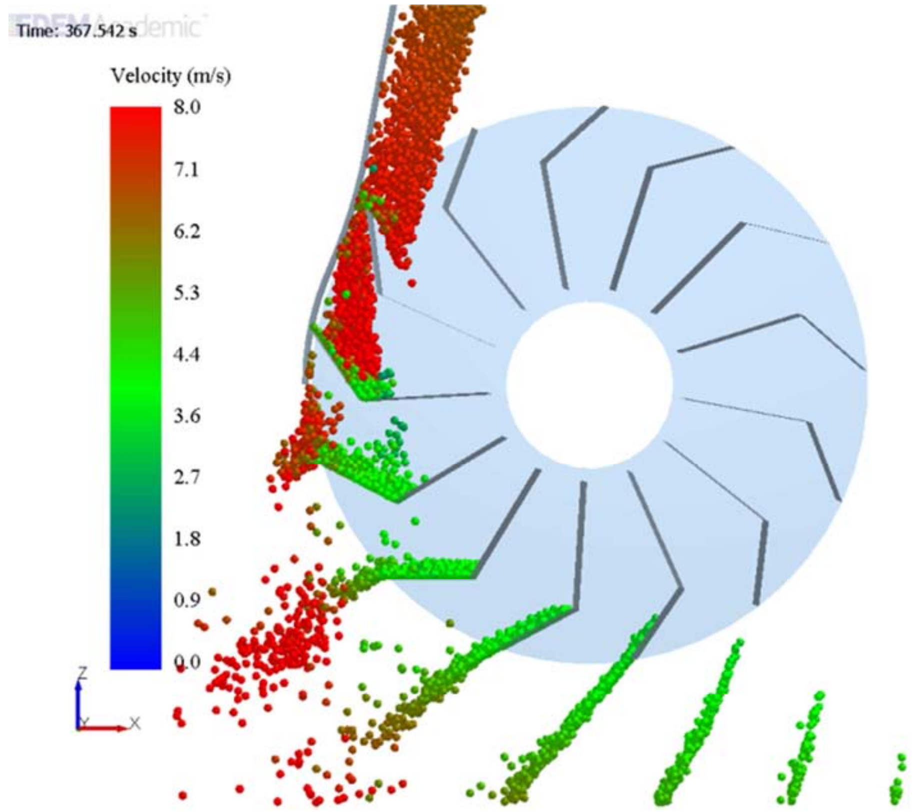


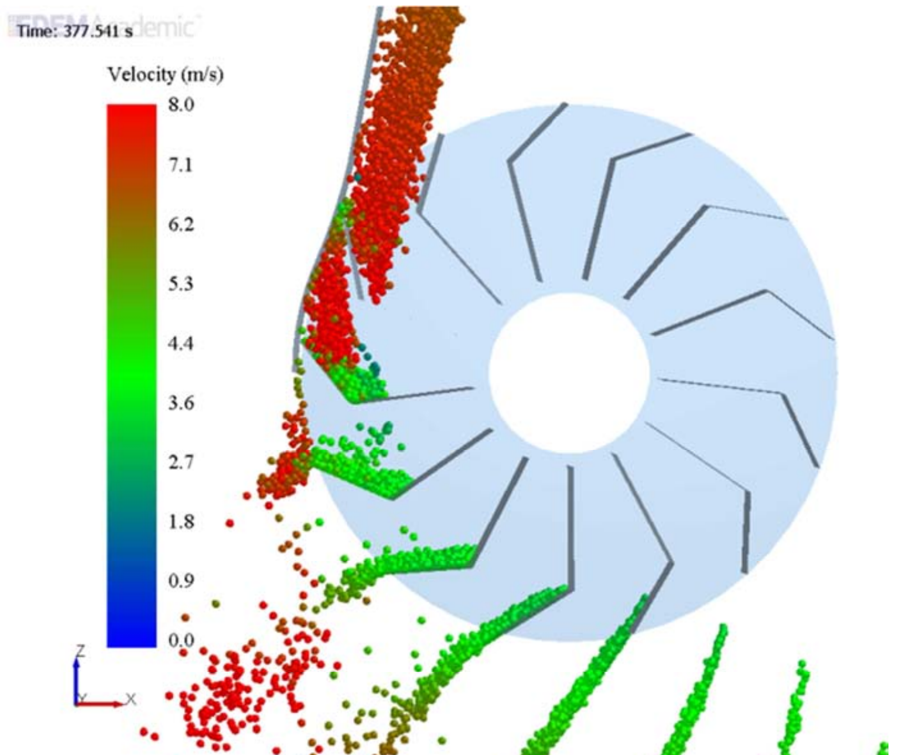
Figure 18. *Guide plate design for the stockpile turbine.*

6.1. Efficiency of the "Stockpile Turbine"

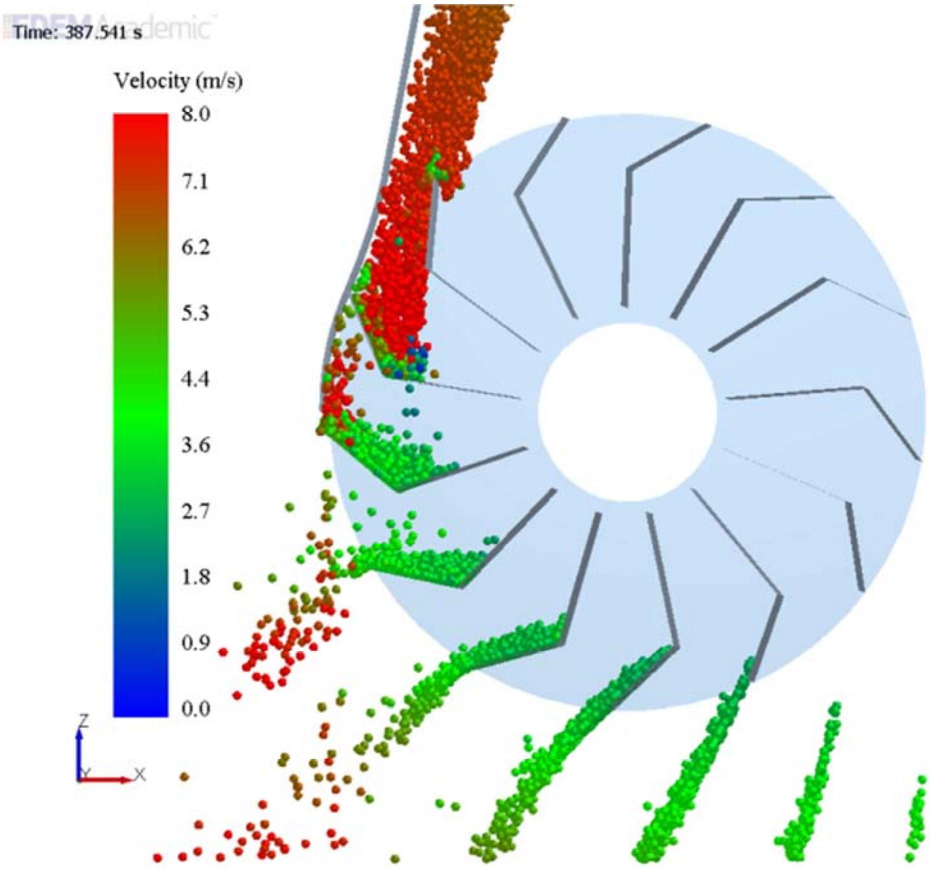
To obtain the maximum efficiency of the turbine, the optimal rotational speed was calculated. Therefore the rotational speed was decreased from 50 rpm down to 20 rpm in intervals of 5 rpm. The simulation results are shown in Figure 19.



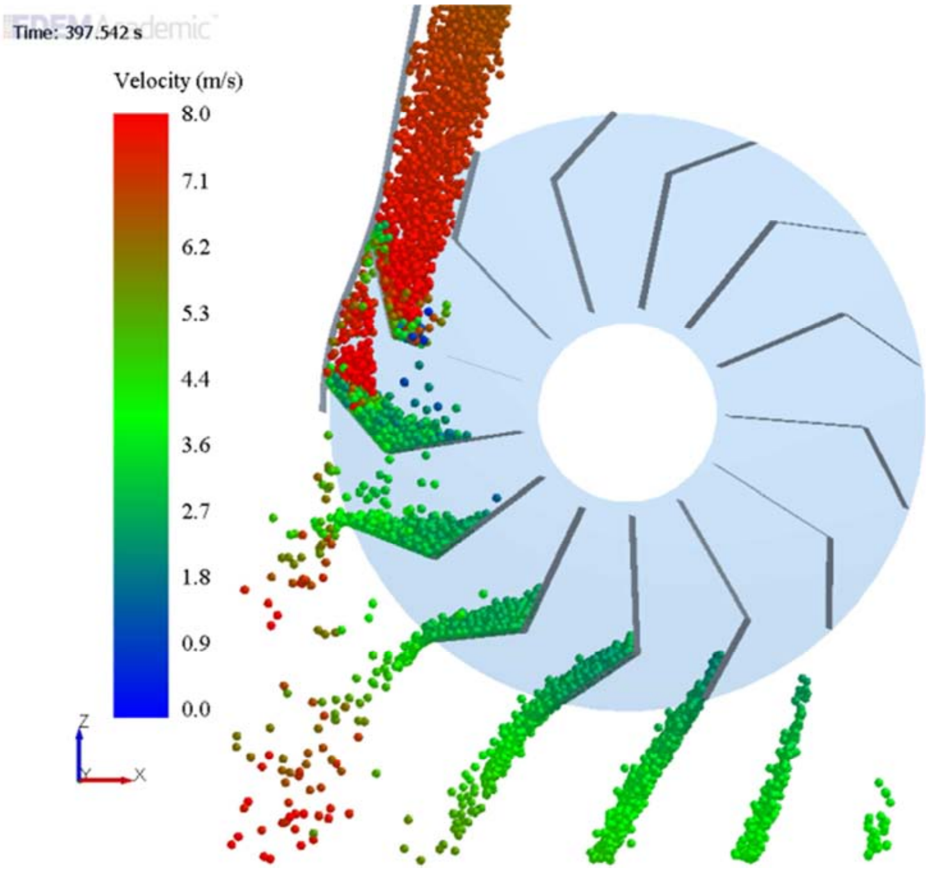
rotational speed = 50 rpm



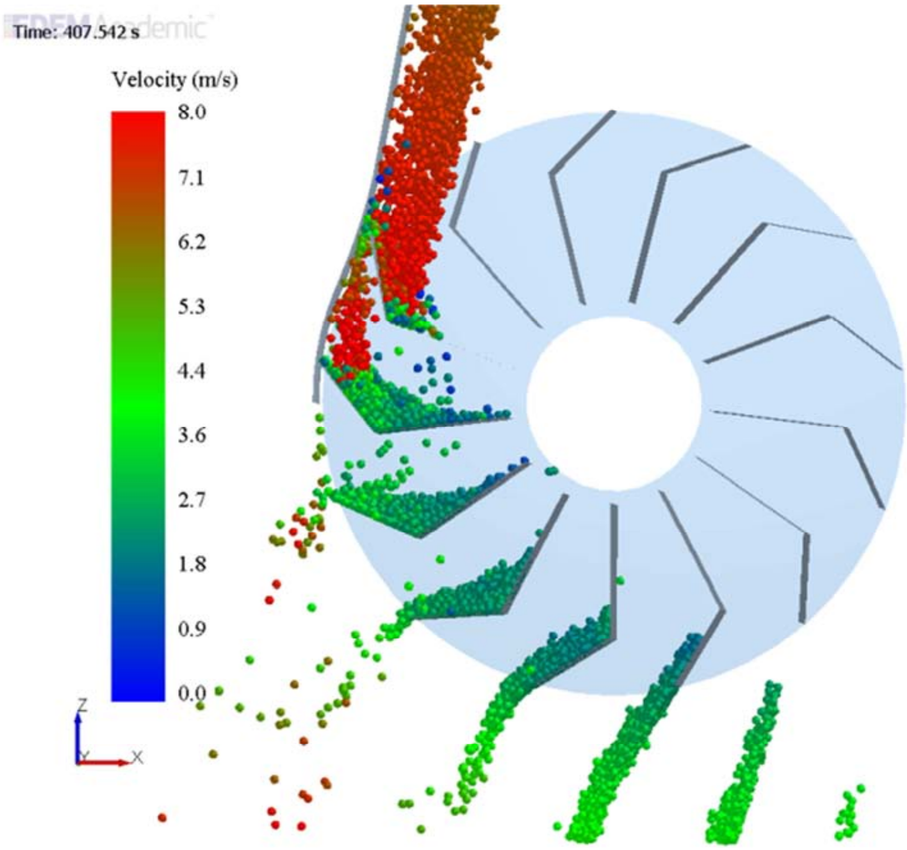
rotational speed = 45rpm



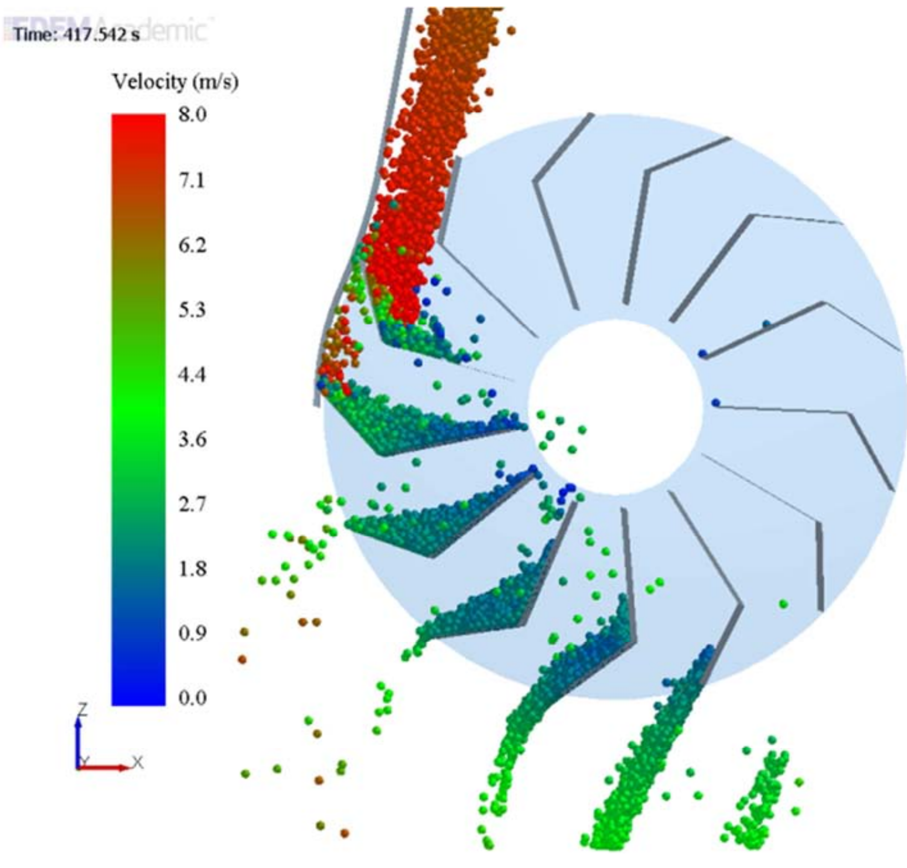
rotational speed = 40 rpm



rotational speed = 35 rpm



rotational speed = 30 rpm



rotational speed = 25 rpm

Figure 19. *Rotational speed of the turbine decreased from 50 rpm down to 25 rpm.*

Figure 20 shows the torques, which were generated by the turbine at different rotational speeds. At about 35 rpm, which leads to a torque of 1 881 Nm, a maximum power output of 6 893 W could be calculated. The power content of the bulk material at the lowest point of the turbine is 12 426 W. An efficiency of 55% without any mechanical losses was calculated. The complete power plot for different rotational speeds is shown in Figure 21. The simulations were carried out with a short guide plate, which is shorter than the plate shown in Figure 18. The arc length was shortened by about 440 mm.

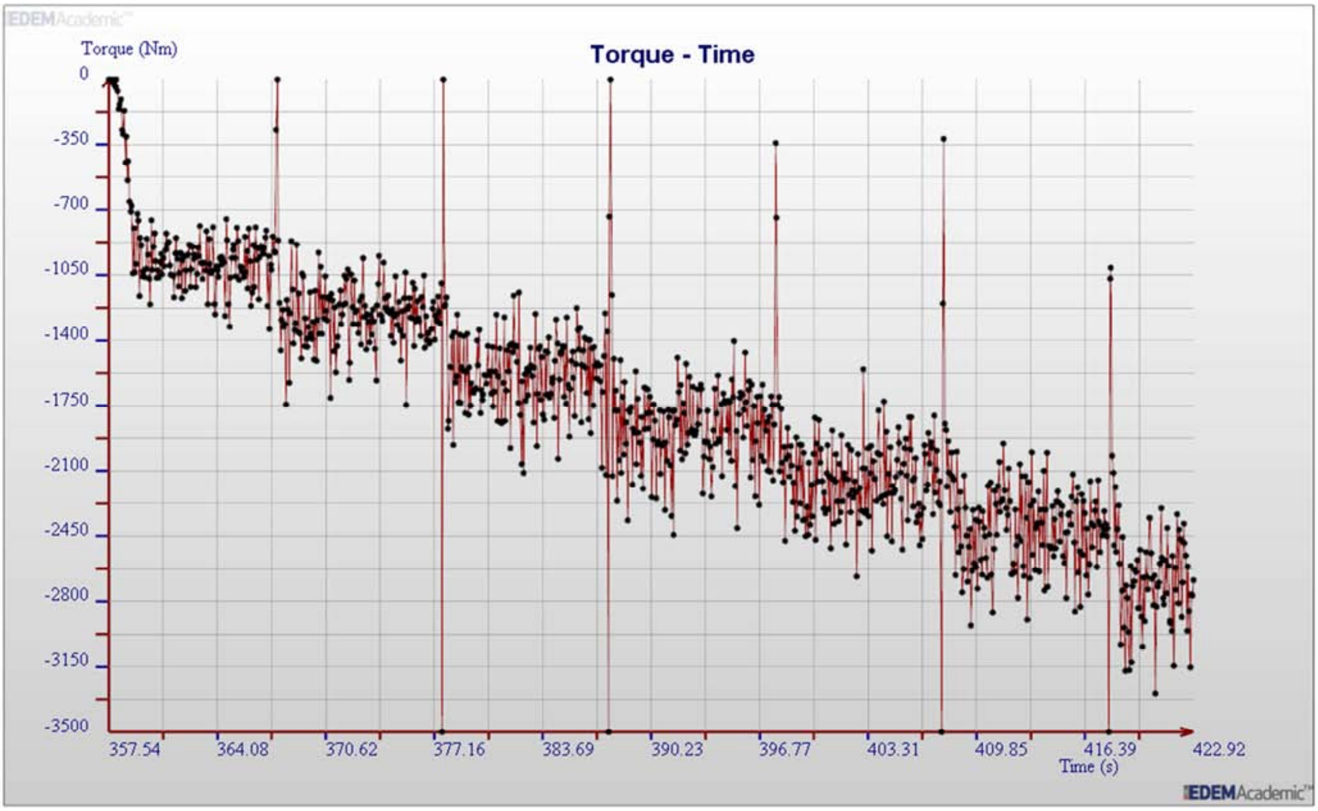


Figure 20. *Torque of the turbine at different rotational speeds.*

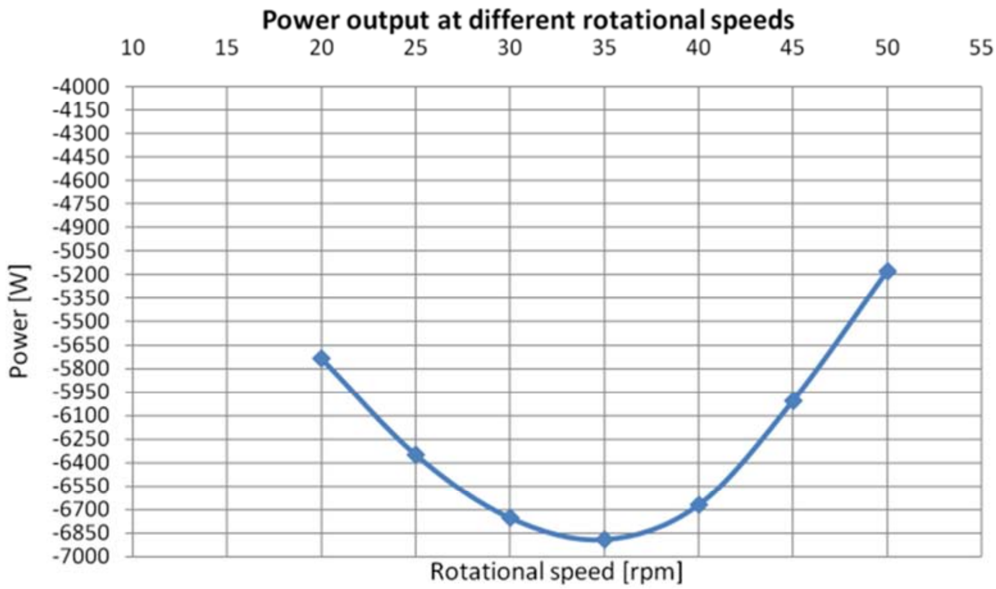
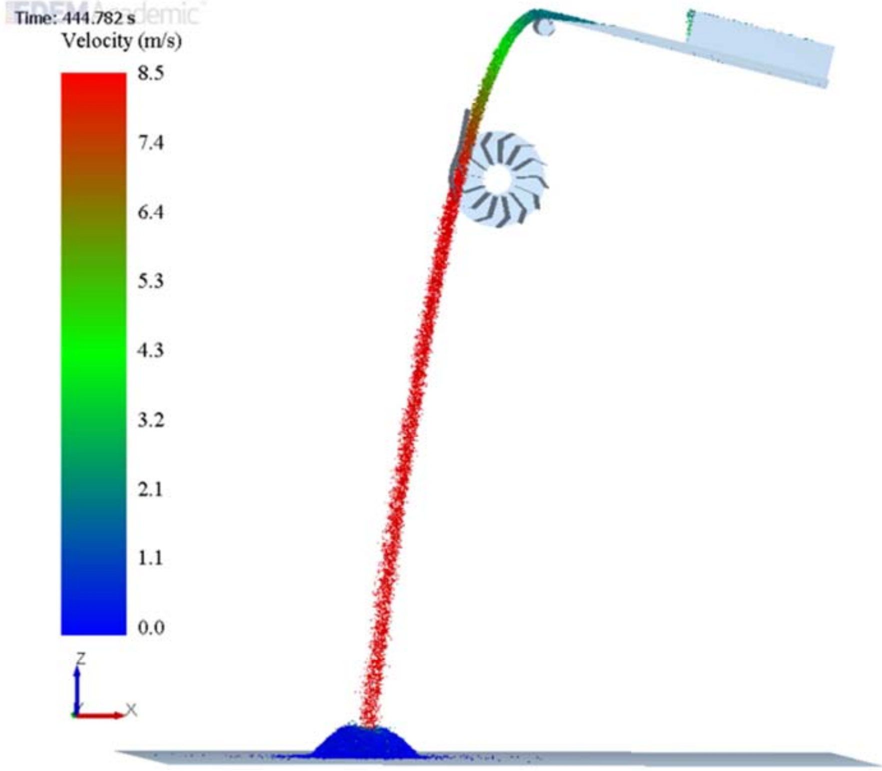
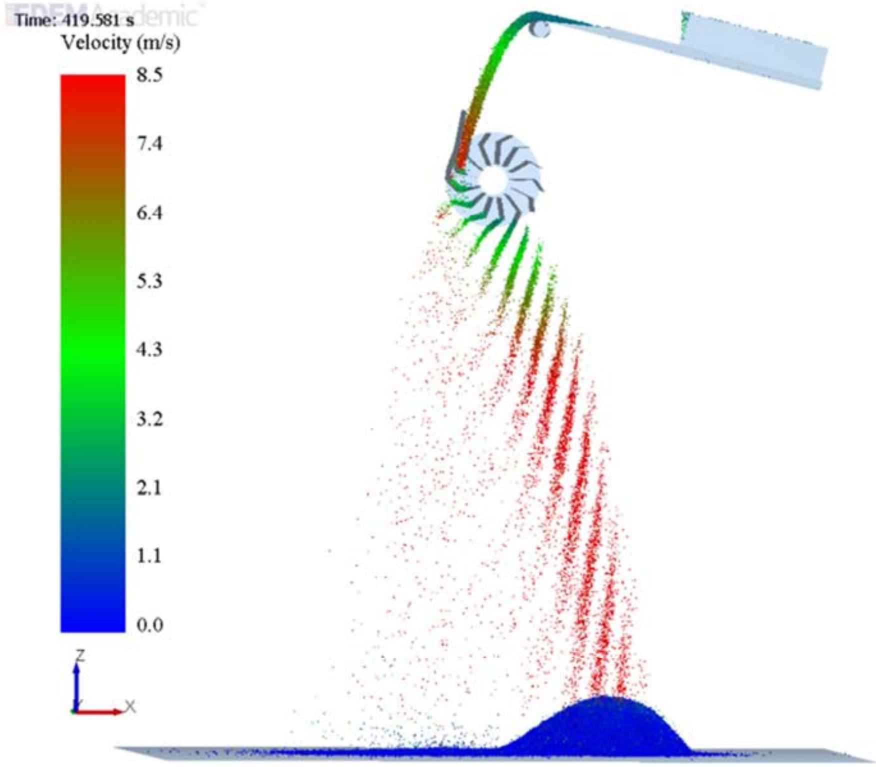


Figure 21. *Power output at different rotational speeds.*

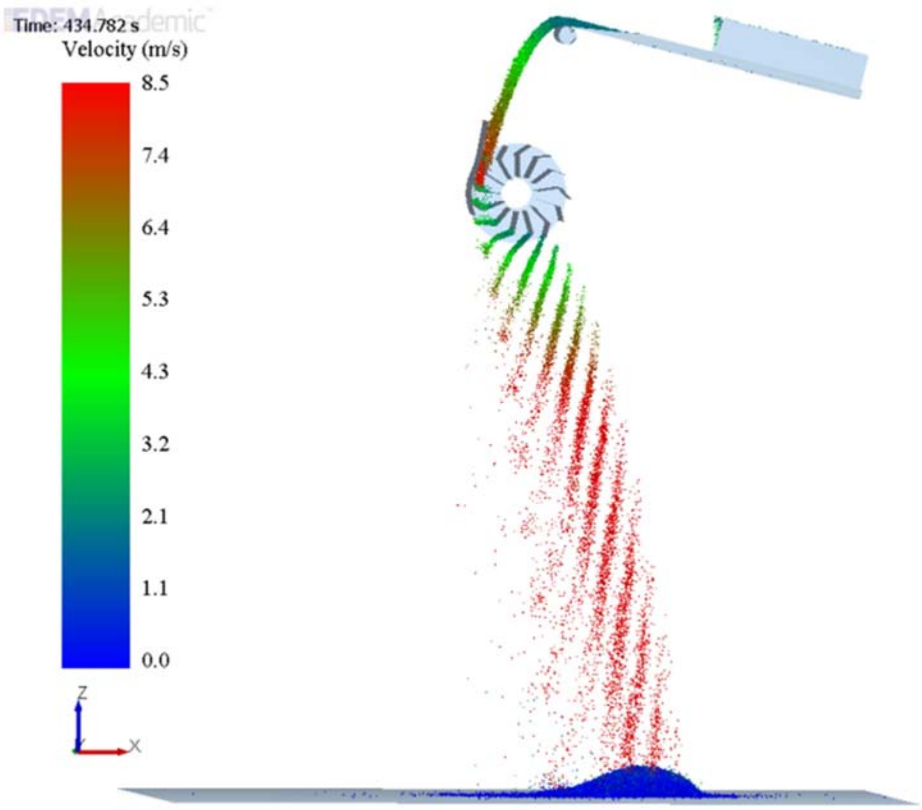
The geometry of the bulk material heap is important for a stockpile. Due to the turbine used, the heap geometry changes. Figure 22 shows the differences between no turbine, and a turbine with a short and a long guide plate at 35 rpm and 25 rpm. It can be seen that the heap peak moves towards the pulley of the conveyor belt, when a turbine is used. When the turbine operates with a short guide plate and a rotational speed of 35 rpm (maximum power output), the bulk material stream widens and particles get separated. This behaviour leads to a deformed bulk material cone and, especially with dry material condition, due to the particle separation, dust generation has to be expected [15]. The longer guide plate was used to reduce this spraying or separation behaviour. Reducing the rotational speed to 25 rpm further improves this behaviour, but reduces the efficiency of the turbine to 51%.



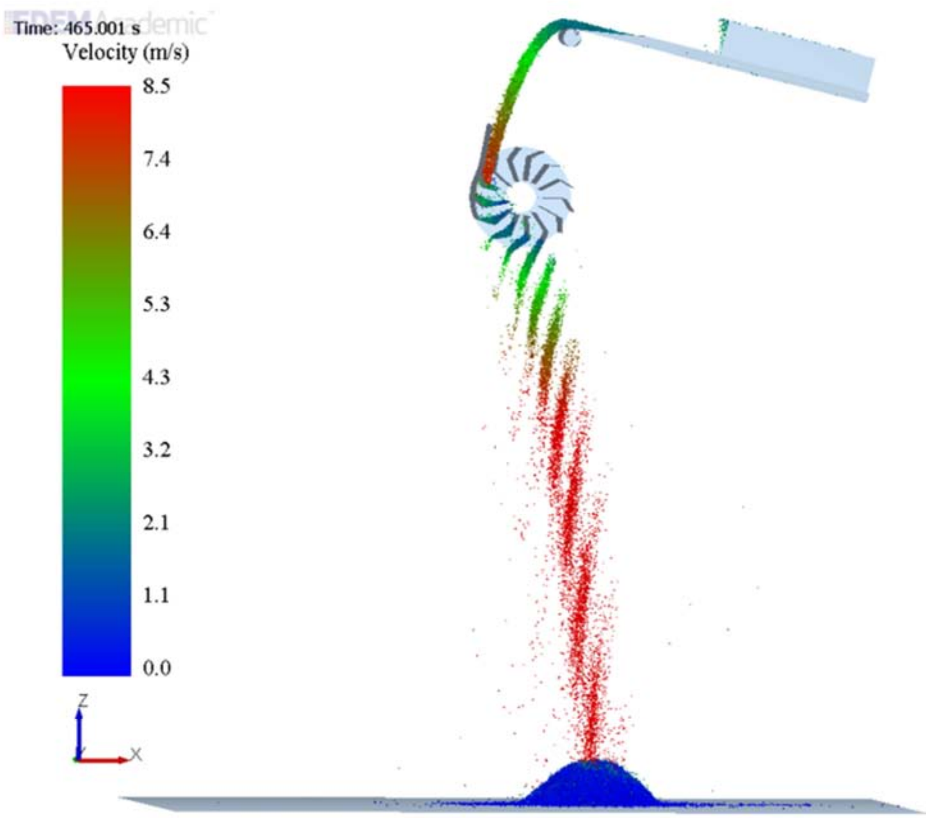
without a turbine



short guide plate - 35 rpm



long guide plate - 35 rpm



long guide plate - 25 rpm

Figure 22. *Bulk material heap geometry of the stockpile with and without a turbine.*

Especially for that conveyed bulk material (sand and gravel), which is excavated from the seabed, dust generation is not an issue because the material is always wet. Due to the longer guide plate, the efficiency of the turbine can be slightly improved up to 57% at 35 rpm. The simulation leads to a maximum power output of 7 061 W. Considering all the mechanical and electrical losses, a power output of 5 kW is realistic, if a generator is used. Using the daily operating time of 13 h and 0.15 Euros for the electricity rate and the network fee, about 3500 Euros could be saved every year. The amortisation period will be less than 10 years, if a turbine prototype is used.

6.2. Stockpile Turbine Spill

As mentioned before, it is planned to spill the turbine with the bulk material to avoid storage losses. Figure 23 shows the difference between the idealized bulk material cones with and without a turbine spill. Without a turbine spill, the storage capacity decreases to about 40%, which is unacceptable for the plant operator.

The disadvantage of a turbine spill is the shorter operation time of the turbine during the heap build-up. The operation time is proportional to the storage losses without spilling the turbine. The energy or power, which could be recovered also decreases to 40%, if the heap is totally filled up before being reclaimed. This leads to a significantly longer amortisation period. For such conditions it is absolutely necessary to reduce the turbine costs. For turbines with a power output of less than 10 kW a serial production of the turbine is needed. An alternative solution would be a turbine, which moves vertically with the growth of the bulk material cone. Such a construction will be cost intensive. These additional costs must be compared with the extra earnings due to the increase of the sum of the power output and the increase of the operating time.

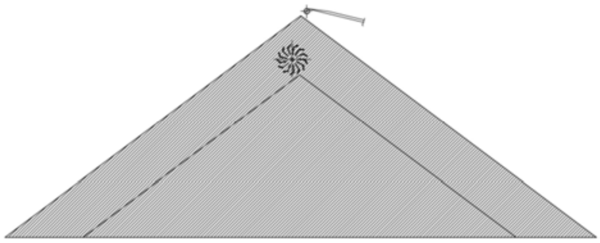
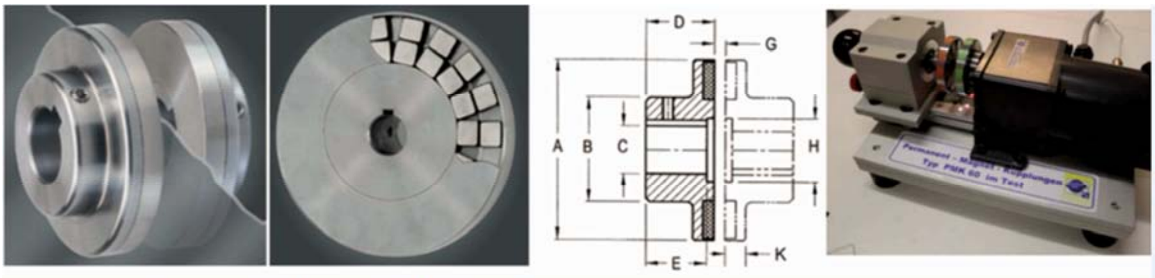


Figure 23. *Turbine spill by the bulk material heap.*

6.3. Safety Coupling for a Turbine Spill



Part-No.	max. speed [U/min] [rpm]	weight per disc [kg]	max. torque at G = 1 mm [Nm]	max. Axial force [N]	dimensions A [mm]	B [mm]	H [mm]	D [mm]	E [mm]	K [mm]	max. bore Ø C [mm]
PMK20	42000	0,1	0,3	51,0	26,9	20,6	—	15,9	—	9,6	8H7
PMK40	26000	0,1	2,0	140,0	43,7	20,6	—	15,0	—	8,6	10H7
PMK50	23000	0,2	3,0	210,0	50,0	28,5	—	15,0	—	8,6	12H7
PMK60	19000	0,3	3,8	255,0	60,0	38,0	—	19,1	—	10,2	19H7
PMK70	15000	0,6	7,4	345,0	73,0	51,0	—	25,4	—	11,9	22H7
PMK91	11000	1,1	15,5	446,0	97,5	70,0	—	25,4	—	11,9	35H7
PMK110	10600	1,2	27,7	807,0	106,5	70,0	—	25,4	—	14,2	35H7
PMK130	9000	2,0	45,0	1100,0	129,5	76,3	52,0	38,0	31,6	15,0	40H7
PMK150	9200	2,2	54,0	1630,0	125,0	70,0	52,0	38,0	31,6	25,3	40H7
PMK170	9200	3,1	65,0	1900,0	125,0	108,0	68,0	47,6	41,4	24,6	55H7
PMK190	7800	4,1	95,0	2370,0	147,0	108,0	77,5	52,5	45,5	24,0	60H7
PMK200	7800	3,1	146,0	3114,0	147,0	60,0	48,0	42,0	32,0	25,7	38H7

Figure 24. *Standard permanent magnetic safety coupling [16].*

If the turbine is spilled, a safety coupling between the turbine and the power unit is necessary. The power unit could be a generator or a traction drive to the pulley of the conveyor as shown in Figure 10. If the bulk material is nonmagnetic, a permanent magnet coupling could be used as a safety device and to transmit the energy from the turbine to the power unit. A standard permanent magnetic coupling for low torques is shown in Figure 24. Such a coupling has the advantage to overspeed as soon as the torque reaches its maximum transferable value. This happens, when the bulk material heap reaches the turbine. The turbine immediately slows down and due to the mass inertia the torque rises. If that torque is higher than the maximum transferable torque of the coupling, the system overspeeds and no further torque can be transmitted. When the whole system (turbine and e.g. generator) has stopped, it can be run up once again.

At the "Chair of Mining Engineering and Mineral Economics - Conveying Technology and Design Methods" tests with permanent magnet couplings were carried out. Instead of a coupling with magnets installed at the face side (Figure 24), couplings with magnets mounted on the surface shells of the two cylindrical shaped coupling halves (Figure 25) were tested.

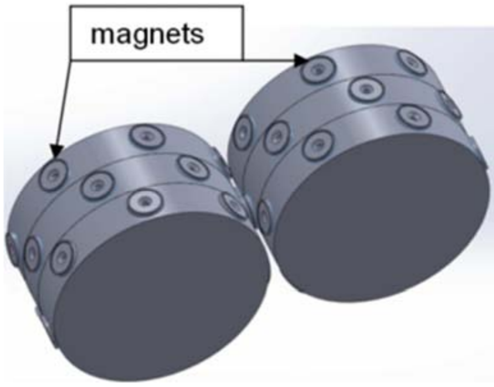


Figure 25. *Permanent magnet coupling with magnets on the surface shells.*

Such a system could be used as a speed transformer, if different diameters of the coupling halves are used. The tests were carried out without a speed transformation. Figure 26 shows the design of the coupling test stand. The test stand consists out of an electric motor from the company Lenze - Type MDFKAIG100 - 22 and a second one, which was used as a generator. For torque and speed measuring, two torque measuring flanges Type T10F from the Company HBM were used. The air gap between the two coupling halves could be adjusted by means of a positioning device.

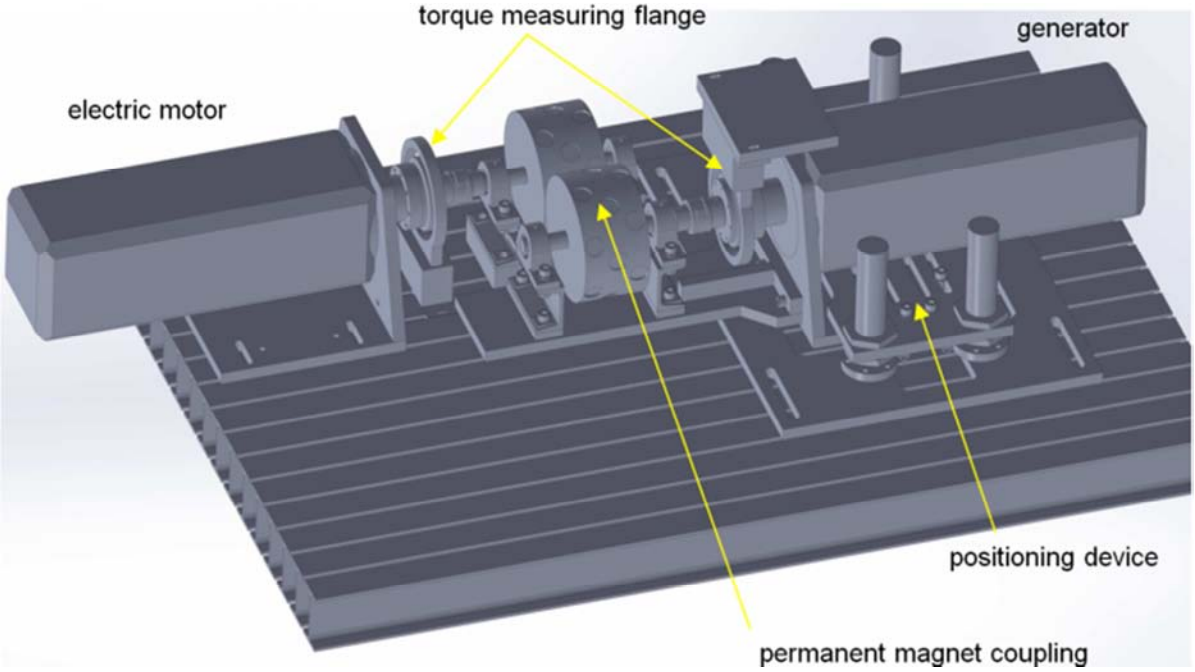


Figure 26. *Test stand for the permanent magnet coupling.*

The two coupling halves had a diameter of 200 mm and were each equipped with 21 permanent magnets. The NdFeB - magnets had a diameter of 32 mm and a height of 7 mm. The maximum adhesive force of a magnet was 294 N. The two permanent magnet coupling halves, which were made out of plywood, had a separate bearing unit and were connected to the generator and the motor via a claw coupling.

Figure 27 shows the test stand during operation. A minimum air gap of 0.75 mm was achieved due to the simple construction of the coupling and the magnets used.

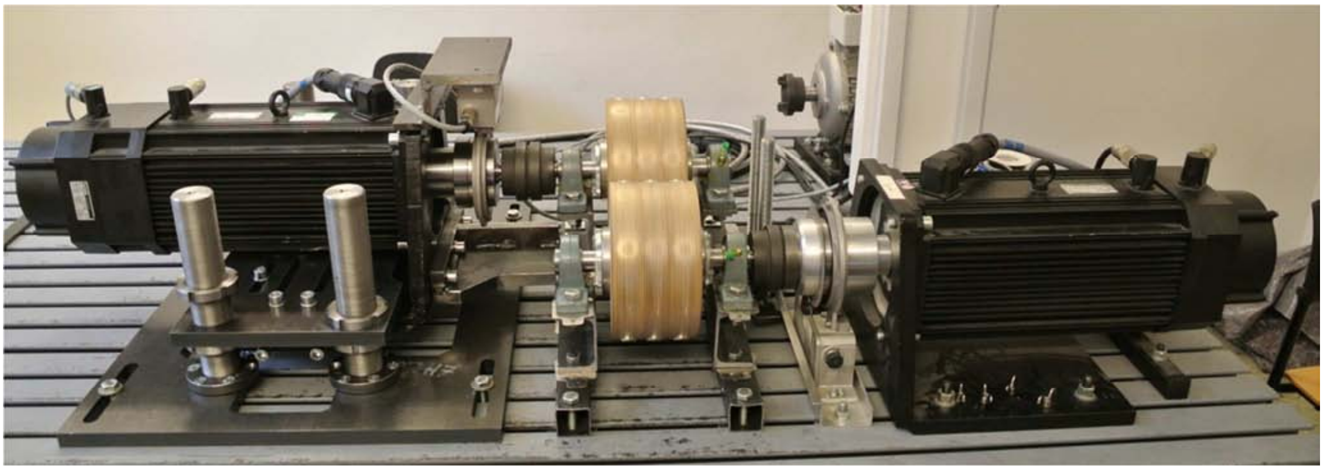


Figure 27. *Permanent magnet coupling during operation.*

Figure 28 illustrates the measured power transmission of the coupling at 1500 rpm and 500 rpm. The torque was gradually increased until the maximum of about 5 Nm was reached. With an air gap of 0.75 mm a maximum power of about 770 W (1500 rpm) was transmitted. At higher torques the coupling overspeeds. This well-known behaviour relates to the necessary safety function of the turbine. The experiments were mainly carried out to learn more about the efficiency of such a coupling type.

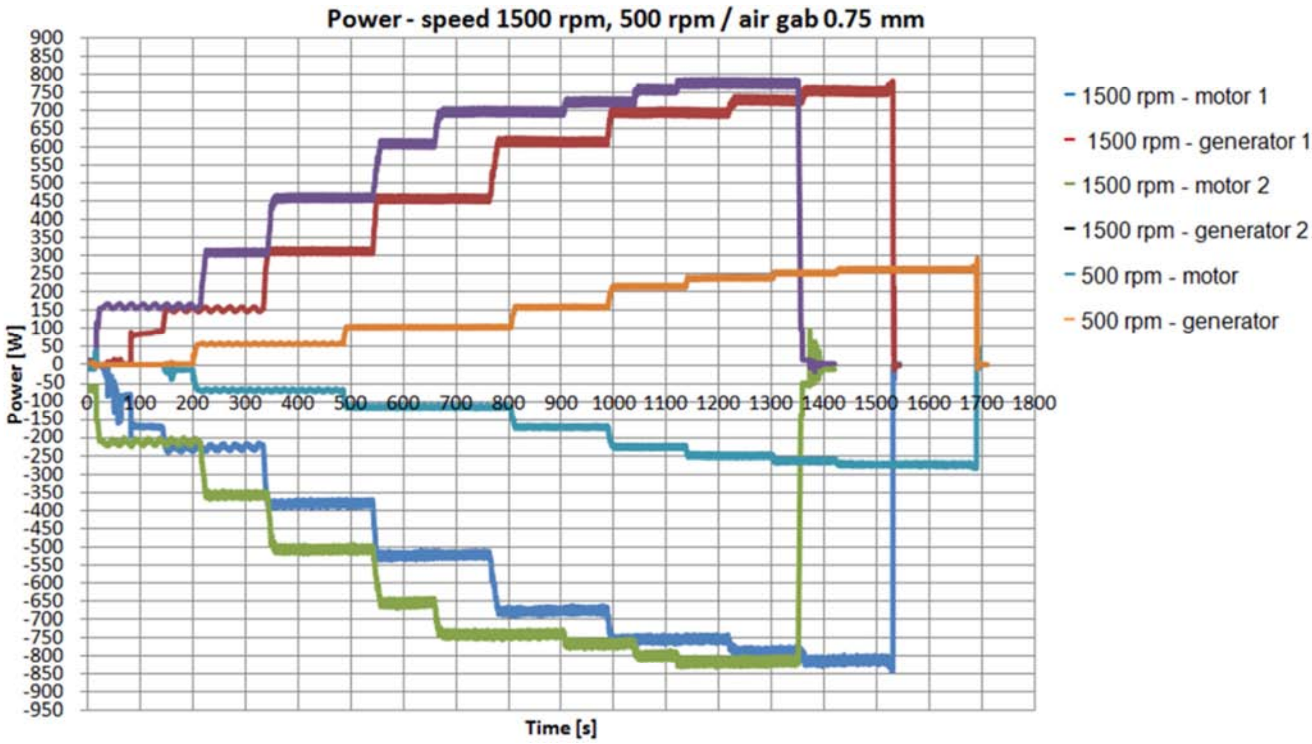


Figure 28. *Power curve of the coupling.*

The two coupling halves were pivoted on two pedestal bearings, which are normally not necessary. The coupling halves can be directly connected to the motor and the generator shaft. The bearings were necessary to avoid any damage to the torque measurement flanges. The bearings and the additional claw coupling causes losses which normally do not occur. These losses were separately measured via a no-load test for the generator and the motor. Figure 29 shows the efficiency of the permanent magnet coupling with and without these losses. With the available measuring system no losses due to the permanent magnet coupling were measured. In Figure 29 an efficiency, without losses, of more than 100% for 1 500 rpm can be seen. The reason for this behaviour is a small zero point offset of one of the torque measuring flanges, which occurred during this measurement.

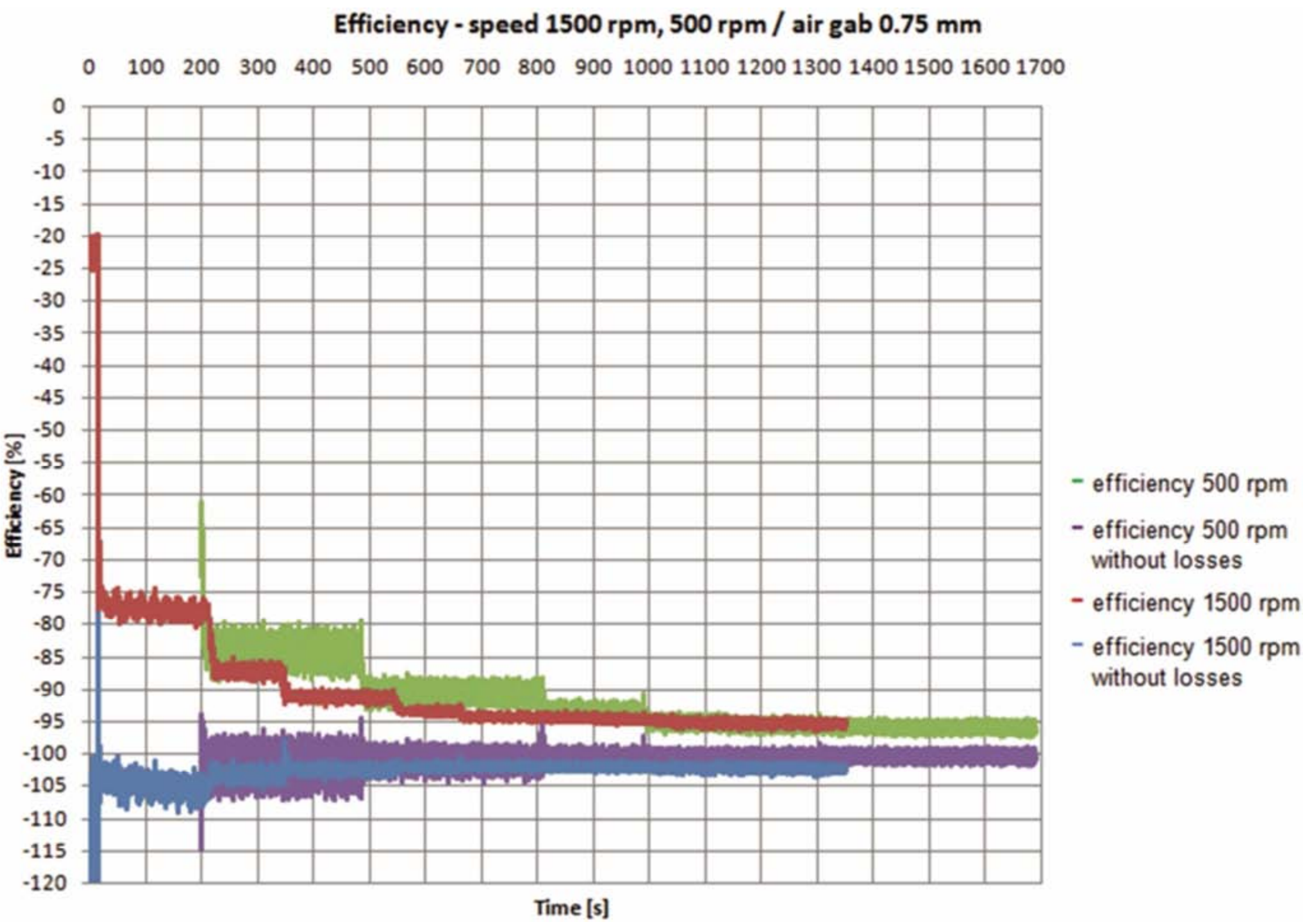


Figure 29. *Efficiency of the coupling.*

The tests showed, that a permanent magnet coupling is applicable for a spill-type "Solid State Material Driven Turbine". Usually the torque of such a system (Figure 20) is significantly higher than the measured transmittable torque of the presented permanent magnet coupling. For safety reasons, the best position for such a coupling would be between the turbine shaft and the in-line power transmission system. At this position the highest torque will occur. It is doubtful that any permanent magnet coupling could handle such a high torque. Therefore, a position with higher speed and less torque, for example between a gearbox and a generator, has to be used for the coupling installation. Also the spillage behaviour should be tested. In cooperation with the company RöchlingLeripaPapertech GmbH & Co. KG a production oriented turbine prototype for the presented stockpile application was built (Figure 30).

The presented turbine should generate more insight into the turbine behaviour in combination with a generator.

The turbine development was carried out for the company Lafarge Tarmac. Due to in-house reasons of the partner, the cooperation was suspended. However, the continuation of this project is uncertain.

Figure 30. *Production oriented turbine prototype for a stockpile [17].*

7. Layout Criteria for an Overshot "Solid State Material Driven Turbine"

The following layout criteria apply to overshot "Solid State Material Driven Turbines". Due to the higher efficiency, only overshot turbines were considered. Figure 31 and Table 3 show the necessary layout parameters. The turbine layout should be verified by a DE - simulation.

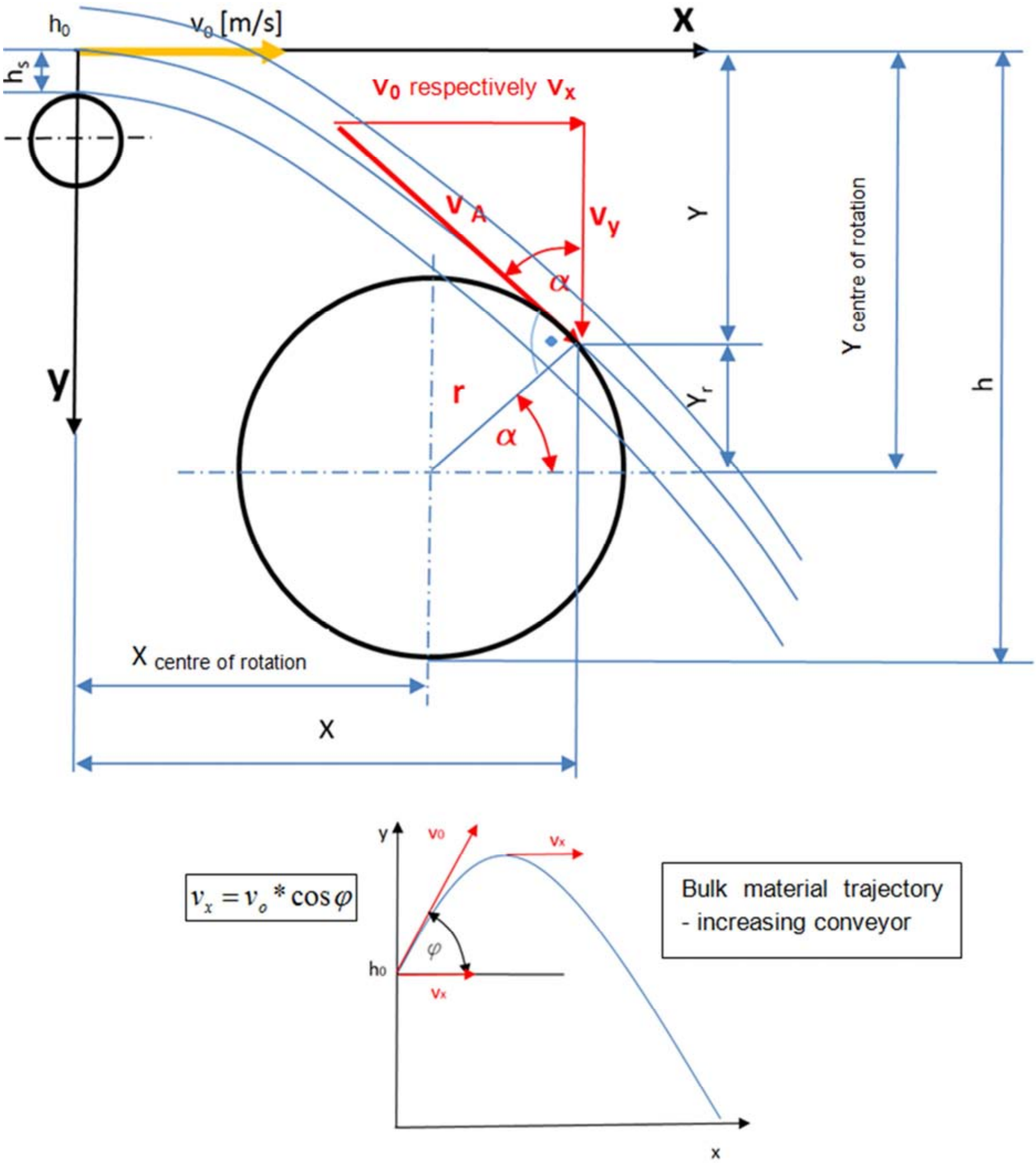


Figure 31. *Layout parameters.*

Table 3. *Layout parameters.*

v_A	Contact speed of the particles perpendicular to the bucket [m/s]
v_0 respectively v_x	Bulk material discharge speed from the conveyor = conveying velocity or horizontal component of v_A [m/s]
v_y	Vertical component of v_A [m/s]
a respectively a (y)	Contact angle [°]
r	Contact radius [m]
h_s	Half of the bulk material loading at the belt conveyor [m]
x and y	Horizontal and vertical component of the bulk material contact with the bucket relating to the discharge point [m]
$x_{centre\ of\ rotation}$ and $y_{centre\ of\ rotation}$	Horizontal and vertical component of the centre of rotation relating to the discharge point [m]
y_r	Vertical component of r [m]
h	Total height [m]
h_0	Drop height [m]

f	Angle of elevation of the belt conveyor
n	Rotational speed of the turbine [1/s]
$N_{buckets}$	Number of buckets
$\dot{V}$	Volume flow rat [m3/s]
e	Tilt angle of the inlet plate (see Figure 32)

The turbine design procedure, using the layout criteria, is as follows (the procedure is illustrated in Figure 32):

1. Calculating the trajectory parabola based on half of the bulk material loading and the conveying velocity, afterwards transmitting the data to a 3D - CAD software

$$y(x) = h_o + x * \tan\varphi - \frac{x^2 * g}{2 * v_0^2 * \cos^2\varphi} \quad (1)$$

2. Moving the trajectory parabola by half the height of the bulk material loading in both directions

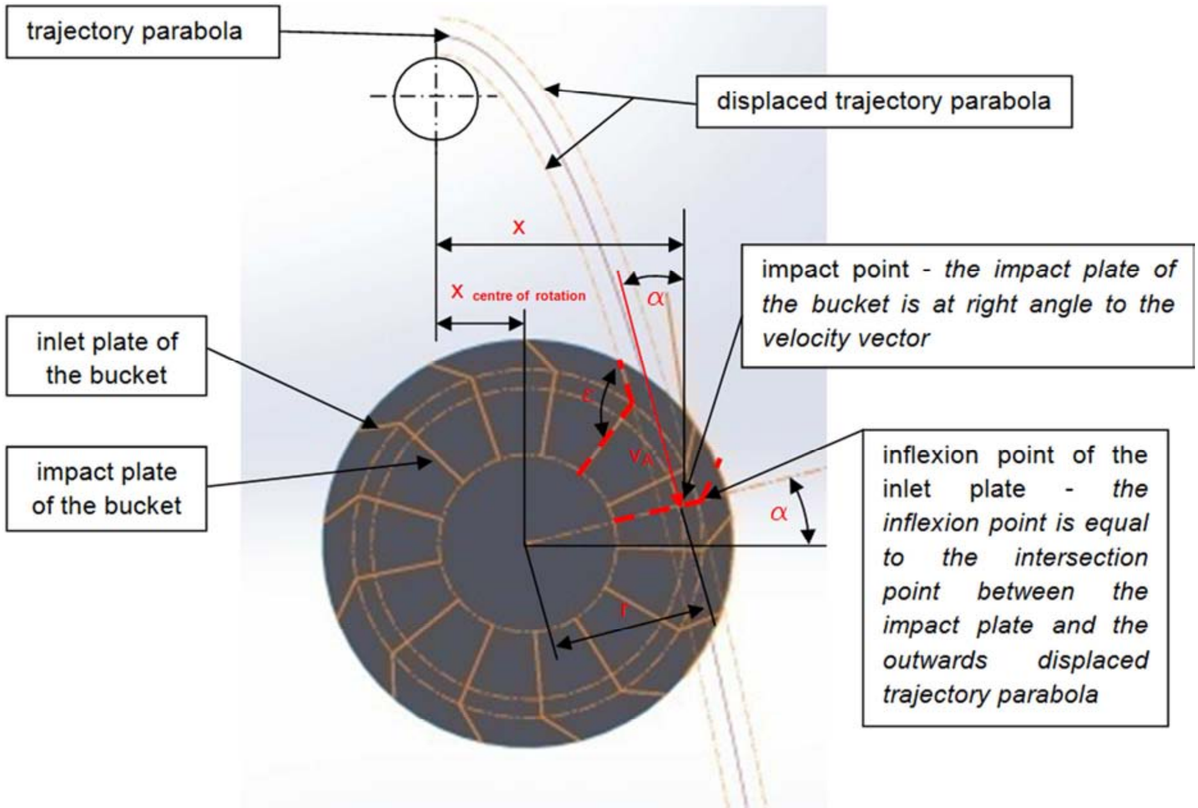


Figure 32. *Design procedure of an overshot: Solid State Material Driven Turbine.*

3. Selecting the bucket size and the number of buckets (the bucket size depends on the bulk material volume, which has to be picked up by each bucket)

$$y_{centre\ of\ rotation} = y + y_r \quad (2)$$

$$y_r = r * \sin\alpha \quad (3)$$

$$x_{centre of rotation} = x - r * \cos\alpha \quad (4)$$

$$\alpha(y) = atn\frac{v_o}{\sqrt{2 * g * |y|}} \quad (5)$$

4. Selecting the bucket size and the number of buckets (the bucket size depends on the bulk material volume, which has to be picked up by each bucket)

$$\text{bulk material volume per bucket} = \frac{\dot{V}}{n_{bucket} * n * \text{fill factor}} \quad (6)$$

$$\text{min . bucket cross sec tion} = \frac{\text{bulk material volume per bucket } [m^3]}{\text{bucket with } [m]} \quad (7)$$

$$\begin{aligned} &\text{bucket with} = \text{belt with } [m] * \text{safety factor} \\ &\text{safety faktor} = 1{,}25 \end{aligned} \quad (8)$$

The length of the impact plate and the inlet plate could be derived from the minimum bucket cross section. The selected geometry data should be verified via the DEM. The safety factor can be varied if necessary.

5. Positioning of the inlet plate and determination of the inclination by using 3D – CAD

The inlet plate must be inclined at a certain angle to the impact plate of the bucket in order to avoid premature escape of the bulk material out of the buckets. The inclination of the impact plate at a rectangular impact of the bulk material towards the impact plate, is equal to the angle between current velocity vector of the bulk material and the vertical direction. The extension of the impact plateintersects the projection of the axis of rotation of the turbine. Due to this condition, no torque against the rotational direction of the turbine will be generated during the inflow of the bulk material into the buckets. The inclination point of the inlet plate is equal to the intersection point between the impact plate and the trajectory parabola, which isdisplaced outwards by half of the bulk material loading at the conveyor belt (see Figure 32). The trajectory parabolas for the outerand the inner position will not becalculated separately, they will be displacedby half of thebulk material load, since this is closer to the actual bulk material movement. The particle trajectory differs from the theoretical trajectory for the outer and inner position due tothe particle interaction during the movement. The tiltangle of the inlet plate must be chosen in such a way, that it enters tangentially into the particle flow (see Figure 32). The design of the inlet plate will be tangentially aligned to the inner trajectory.

6. Design of a 3D surface model of the turbine for verification via the DEM

After the confirmation of the functional principle of the turbine via a DE - simulation and possible slight adjustments, the turbine could be designed and manufactured based on the surface model.

8. Discussion

The main doubts relating to this new technology are always economy and wear. Especially small turbines with a power output less than 10 kW have problems with an acceptable payback time. An important advantage of "Solid State Material Driven Turbines" is the possibility to create additional benefits. These benefits could have a higher economic impact than the actual energy recovery. First tests showed, that turbines have a significant wear benefit compared to standard transfer chutes for continuous conveyors [7]. Standard chutes can be replaced by turbines. At transfer or discharge points particle size segregation occurs. By using a turbine this problem could be avoided. During the long-time test under real conditions (cf. Chapter 4) this behaviour was confirmed. Simulations also showed that it is possible to realize a soft loading effect at a transfer point from one conveyor to another by using a special turbine design. By using such a transfer turbine also particle breakage can be reduced. To investigate all these additional benefits further research projects are in preparation.

9. Conclusion

The paper points out, that it is possible to recover energy from moving bulk materials by using a so-called "Solid State Material Driven Turbine". This system is particularly suitable for continuous conveyors, especially belt conveyors. At discharge or transfer points energy could be recovered. With an overshot turbine design an efficiency of almost 60% is achievable. The amortisation period for such a system strongly depends on the energy content of the moving bulk material. The energy content depends on the conveyed mass flow and the usable drop height. For turbines with a rated power of less than 10 kW, a series production of the turbine is necessary, to obtain a payback time significantly shorter than ten years. For a combination with a high capacity conveyor, an amortisation time of shorter than two years is realistic. This unused but recoverable energy is usually converted into particle and plant wear. By using a "Solid State Material Driven Turbine" energy and costs can be saved.

The "Chair of Mining Engineering and Mineral Economics - Conveying Technology and Design Methods" at the Montanuniversität Leoben / Austria currently plans to test a turbine with a rated power output higher than 10 kW with an interested suitable industrial partner.

References

[1] Conveyor Equipment Manufacturers Association. (2014). Belt conveyors for bulk materials. 7th ed. Florida: Conveyor Equipment Manufacturers Association.

[2] Prenner Michael, inventor; Montanuniversität Leoben, assignee. Solid State Material Driven Turbine. European Patent EP 2788622 B1. 2018 May 23.

[3] ANDRITZ HYDRO GmbH. (2018) SMALL AND MINI HYDROPOWER SOLUTIONS. https://www.andritz.com/products-n/group/markets/small-mini-hydropower-plants.

[4] Rodríguez, J., Pontt, J., Becker, N., and Weinstein, A. (2002) Regenerative Drives in the Megawatt Range for High-Performance Downhill Belt Conveyors. IEEE TRANSACTIONS ON INDUSTRY APPLICATIONS, VOL. 38, NO. 1, JANUARY/FEBRUARY 2002.

[5] De Graaf, M. C. (2013). Viability study of an energy-recovery system for belt conveyors. Department Marine and Transport Technology, Delft University of Technology.

[6] Prenner, M., Kessler, F. (2012). Energy-recovery System for Continuous Conveyors. Conference publication, BulkEurope 2012.

[7] Prenner, M., Grübler, C. & Zeiler, S. (2018) Vorteile von Feststoffturbinen. Schüttgut, 5, 68-72.

[8] Ashburner, R., Hull, P., Lloyd, M., Partington, R., Romain, P. (2017). Dulverton Weir & Leat Conservation Trust. http://dulvertonweir.org.uk/gallery/.

[9] Tanaka Hydropower CO., LTD. (2017). http://www.tanasui.co.jp/productsTurbinesEn.html.

[10] Julius, M. H. (1909) Meyers Großes Konversations-Lexikon, 20th ed. Leipzig and Vienna: Bibliographisches Institut.

[11] Jakob, C., Konietzky, H. (2012). Partikelmethoden - Eine Übersicht -. Freiberg: Technische Universität Bergakademie Freiberg.

[12] Prenner, M. (2015). Feststoffturbine zur Energierückgewinnung in Kombination mit Gurtförderanlagen. BHM Berg- und Hüttenmännisches Monatsheft, 1, 21-31.

[13] Prenner, M. (2014). Feststoffturbine zur Energierückgewinnung für stetige Schüttgurförderanlagen. Wer Was Wo. Forschung@MUL, 2014, 186.

[14] Prenner, M. (2013). Endbericht - Energierückgewinnung, FFG-Programm: Intelligente Produktion, FFG Projektnummer: 834274.

[15] Liu, Z. Q. (2003). Air entrainment in free falling bulk materials, Doctor of Philosophy thesis, Faculty of Engineering, University of Wollongong. http://ro.uow.edu.au/theses/1816.

[16] Technische Antriebselemente GmbH. (2017). http://www.tea-hamburg.de/en/.

[17] Röchling Leripa Papertech GmbH & Co. KG. (2015) http://www.leripa.com/de/aktuelles/news/action.view/entity.detail/key. 35.html. MAWEV, Enns, 2015 March 18. - 21.