Claus Zander

Actuator technology for hydraulic steel structures

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English translation by Barbara Wolf

ACTUATOR TECHNOLOGY FOR HYDRAULIC STEEL STRUCTURES

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INTRODUCTION

Electric actuators for valves were standardised in 1968. Flanges and pertaining torques were introduced. At the same time, self-locking was specified for large speed ranges. For classical valves – gate valves and globe valves – the running times are typically less than one minute and the torque is quickly reduced after unseating.

Completely different situations prevail for hydraulic steel structures. Large strokes and long running times are applying high forces across the overall travel and might cause overheating of electric actuators, resulting in consequential damage if actuators are not correctly sized for the hydraulic steel structure application.

The author of this book, Claus Zander, is an expert of actuators. He worked as a field test engineer in MAW Magdeburg, Germany, and subsequently for AUMA in Sales. In terms of civil engineering constructions for water applications, he was often involved in projects from the planning phase to the final handover of the sites. Mr Zander automated installations ranging from small sites to giant locks.

This book is meant to support all stakeholders involved in planning and sizing within civil engineering constructions for water applications. It further allows young engineers to get insight into the hydraulic steel structures segment, but also provides valuable information and ideas for experienced engineers. This book will serve as obligatory literature for manufacturers in actuator technology, in particular with AUMA or other manufacturers.

I would like to thank Mr Zander for all his endeavours. He has compiled many illustrations, has spoken to numerous experts for hydraulic steel structures and incorporated the combined expert knowledge into this reference book “Actuator technology for hydraulic steel structures”.

Werner Riester

Werner Riester
Co-founder of AUMA

PREFACE

Why do we need a specialised literature on "Actuator technology for hydraulic steel structures"?

Insiders know the operation installations of old hydraulic steel sites such as weirs and locks. They often comprise open gearboxes which are easy to lubricate with a slightly dripping brush (figure 1).

Today, customers require reliable, environmentally friendly and safe systems. Typically, automation should not be manual – except in emergencies – but motor-driven and networking is at the forefront. Long expected lifetimes shall be paired with low maintenance. Solid engineering and modular systems for easy combinations are required.

For quite some time, the demand for automatic and manual operation of globe and gate valves as well as ball valves is steadily increasing. The industry has adjusted to these needs. Furthermore, there are special market segments which call for adapted actuator technology fulfilling specific demands.

Civil engineering for hydraulic steel structures are one of these segments. Many task statements and requirements go far beyond the scope of the available valve engineering portfolio and seem impossible to meet. However, civil enginers for water applications like to rely on

Figure 1: Shutter weir with open mechanics

established and proven technology. Manufacturers continuously adjust to market requirements. The information necessary to size the perfect solutions was not always accessible for interested stakeholders. A multitude of particularly special and useful elements have been designed but they hardly became known.

The goal of this book is to provide comprehensive technological and practical information and expertise to constructors for hydraulic steel structures, planning engineers and consultants. In order to compile a complete tender specification, the contracting entity shall be capable of drawing up an initial sizing of the required actuator technology with the objective that customers receive comparable offers by the bidders.

Claus Zander

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1 HISTORY OF HYDRAULIC STRUCTURES AT A GLANCE

The first traffic routes for humans were waterways. These include rivers as navigable waterways apart from lakes and oceans. They could be used as they were – with all bends, sandbanks and rapids. They were even more useful after river alterations.

Until today, dykes make sure that the river stays in its bed even in the event of floods. Residents of the area claim: "Build the dyke before floods strike!" Bends can be cut-off. The river course is straightened. It is even possible to reroute the river to shorten ways or for irrigating dry landscapes. To improve navigability, it is possible to erect a **fixed weir**. For this purpose, a bank or a wall is erected across the river bed at an appropriate height to raise the water level (**figure 1.01**).

Figure 1.01: Fixed weir

Figure 1.02: Stop log weir

Vessels can navigate from such a weir to the next and have to be unloaded to a vehicle to get to the subsequent weir and lock. Any type of hydraulic structures were also used for agriculture, fisheries as well general industrial applications. In the beginning, this concerned mainly the usability of water mills. Today, adaptations of fixed weirs are becoming increasingly popular for creeks and smaller rivers. They are known as **rock ramps**. They combine the advantages of fixed weirs with virtual ecological flow of water due to their considerably lower slope. Fish can pass the rock ramps.

Due to **stop log weirs**, dam heights are no longer an issue. Depending on the desired water level, beams are placed on top of each other and dropped into slots inside a weir (**figure 1.02**).

Needle weirs are also used for these applications (**figure 1.03**). A solid frame is anchored diagonally to the flow direction of the water bed. Wooden needles designed as straight and not too thick trunks or even squared timber are placed closely and vertically into the needle shoe and leaned against the needle support. The crossway is accessible. Individual needles can be added or removed to modify to the desired level and maintain it fairly stable. In winter, this type of damming is not possible due to the risk of ground freezing: All needles are then removed and the needle blocks as well as footbridges tied to chains are lowered to the weir bed by a cable winch.

Recently, metal pipes are used for this purpose. These needle weirs are often used for repair work. since they allow draining water from the weir construction sites.

Lateral weirs were not very practical for river navigation. To allow unhindered river navigation, locks had to be developed. For maintaining water levels, flexible weirs gained importance compared to the previous adjustable stop log weirs and needle weirs. The development of **sluice gate weirs** were of crucial importance since they are still the standard solution for specific applications. A board dropped into lateral slots was lifted and lowered by means of a chain wound onto a drum. For coiling and uncoiling the chain, squared timbers were inserted into the drum and used as lever (**figure 1.04**).

A significant difference to today's sluice gate weirs

Figure 1.03: Needle weir

Figure 1.04: Early sluice gate weir

Figure 1.05: Panel gate weir

is the material used. With the dawning of the 20th century, previously used wood and cast iron were increasingly replaced by steel. Steel plates, bars and pillars provide larger flow diameters than wood. The use of steel shaped the term: **hydraulic steel construction** in civil engineering constructions for water applications. This includes mainly flexible steel gates in weirs and locks but also inspection gates, rakes, bitts, impact protection devices and canal bridges [6]. One of the first **sluice gate weirs** made of steel – finished in 1875 – was the Pretziener weir close to Magdeburg, Germany which is still operating today and was completely restored in 2010 (**figure 1.05**). It is the largest **panel gate weir** in Europe.

Nine weir sections divided by means of weir needles in small openings with respectively four gate panels block the water drain at low water to maintain the navigability of the Elbe river through Magdeburg. In turn, manually operated chain hoists (**figure 1.06**) drain the flood peaks into the a flood relief canal to protect the city against floods.

During night time, the water level is equalised (**figure 1.07**).

Figure 1.06: The weir is operated with hand winches

Figure 1.07: During the night, the flood relief channel is open

Figure 1.08: Large sluice gate weir

By their nature, weirs must be capable to dam up water systems up to a certain level, irrespective of their size or depth. It was possible to build smaller dams in previous times, but some, as in Pretzien, could only be built little by little. For this reason, spectacular large-scale building projects were always of significant interest. Today, sluice gate weirs are built larger and are no longer operated manually (**figure 1.08**).

To increase their usability, a certain number of weir panels are arranged in horizontally and in parallel as **double gate weirs** (**figure 1.09**).

Around 1900, new revelation was that it was sufficient to place hollow cylinders into the river bed since it is rather warp resistant and must not even been lifted. A sprocket on each side acts as guide by engaging into a toothed rack located in the wall. The weir is moved on ramp via a plate link chain. Weir widths of 50 m are feasible. Due to easy handling even during harsh and extreme winter months, this **roller gate weir** [7] is particularly suited for deployment in Nordic countries (**figure 1.10**).

Figure 1.09: Double gate weir

And since the steel version functions perfectly, why not try applications using extremely sturdy rubber, a material obtained following intensive research. A tube is mounted at both sides of the water bed and arranged at the desired height by means of a dry air compressor. This was the origin of the **inflatable weir** with potentially high weir widths. They are of low operation and maintenance cost; corrosion protection and lubrication are not required. Of course, it is an exotic element in the world of hydraulic steel structures. "Maintaining the level" is easy according to experts, but "controlling is horror" (**figure 1.11**). Instead of filling with air, sometimes water is used which is supposed to increase stability. Longer frosty periods are, however, undesirable and have to be considered during the planning phase.

Figure 1.10: Roller gate weir

Figure 1.11: Inflatable weir

Large weir width can also be covered easily by the **sector weir**. A sluice gate with a circular profile is pivot-mounted on the water bed (**figure 1.12**).

On the one hand, it can be sunk into a prepared weir chamber within the water bed. For maintenance work, each sector can be individually placed flat and sealed with emergency closure devices. On the other hand, it can float to raise the water level (**figure 1.13**), acting as pressure element.

The changes are hydraulically controlled by **tubular gates** – a similar design also known as **ball-type or cylindrical gate** – requiring low energy in combination with the water pressure. Globe and gate valves act as control elements. Actuators take over operation. For example, the Geesthacht weir with lock comprises four sector weirs of 50 m length each.

When the Elbe weir with lock was put into operation in 1960, sector weirs where considered as most economic and state-of-the-art types. However, it did not gain popularity and was rarely built. The clear trend using the same design and effect developed from the hydraulic to the mechanic drive resulting in a simple gate. A rectangular plate (torsion-proof

Figure 1.12: Pivot-mounted sector weir

Figure 1.13: Water retention at sector weir

Figure 1.14: Shutter weir

design by hydraulic steel constructions) was pivot-mounted to the water bed like the sector weir. Water retention was not achieved using hydraulics but through mechanical movements. The **shutter weir** was created (**figure 1.14**).

The improved gate is lenticular, resembling the shape of a fish-belly. Therefore, it is designated as **fish-belly flap gate**. Due to the peculiar shape, it is rather wrap resistant and can therefore be driven one-sided, if required, even across quite large weir widths of 20 m. (**figure 1.15**).

A more sophisticated solution are **combined weirs**, for example the combination of sluice gate weirs and gates or fish-belly flap gates. The water level is controlled by lifting or lowering the sluice gate including the mounted gate. This mounted gate regulates ice discharge and performs minor adjustment (**figure 1.16**).

Figure 1.15: Fish-belly flap weir

A combined version has been implemented at the historical “Palmgarten” weir in Leipzig, Germany. The wide sections in the middle are taken over by roller gate weirs for rough regulation. The lateral sluice gate weirs ensure minor regulation (**figure 1.17**).

However, it is not always sunshine and roses. Following rain and snow melting, the rivers Pleiße and Elster are fairly troubled waters (**figure 1.18**) which looks spectacular but is not really dangerous.

But the ice during long frosty periods might present a problem (**figure 1.19**).

Rivers at high water are far rougher than at normal water level (**figure 1.20**). Depending on the conditions (**figure 1.21**), sluice gates are operated to increase the usable flow diameter and gates are lowered. The fixed weir – visible in the background – is overflown. For safety reasons, locks can be closed in this type of situation.

Figure 1.16: Combined weir: Sluice gate with vertical (left) and horizontal (right) flap gate

Figure 1.17: Roller gate and sluice gate weir

Figure 1.18: Flood drainage

Figure 1.19: Ice load and ice pressure affect sites

Figure 1.20: Normal water level at sluice gate and fish-belly flap gate

Figure 1.21: Same installation as figure 1.20: the flood water drains below or above opened locks and a fixed weir.

Automatic solutions have already been introduced to alleviate flood damage. Shutter weirs are one of them: Wooden **shutter weirs**, pivot-mounted at a third of their height, maintain the desired headwater level and sufficient gradient for the turbines located tailwater. They tilt if the descending water exceeds a preset level (**figure 1.22**). Once the situation has returned to normal, they have to be raised again manually. Today, this type of "flow level control" is achieved by level measurement and remote-controlled or automatic actuator technology.

The same way, locks undergo further developments. Depending on their design, they are named for example **shell lock, head lock, chamber lock** and so on (**figure 1.23**).

Figure 1.22: Shutter weir operated in the event of floods

Among other specialities, Leornardo da Vinci has studied the lock technology [11]. In Italy, they claim that he invented the first **chamber lock**. Irrespective of their names, they need at least two closing elements: an upper and a lower gate, also called head. Distinction is made on the basis of the closing element:

Figure 1.23: Old foursquare lock

Mitre gate locks are the most common ones. Their gates are guided at one side collar and pivot bearings and operated by Galle chains, lantern gears or linear thrust units (**figure 1.24**).

Smaller locks are predominantly equipped with mitre gates or flap gates.

Flap gates are pivot mounted on bottom of the lock basin. After level equalisation, they are lowered until they completely rest outside the chamber (**figure 1.25**).

Very large locks are equipped with **lift gates** (**figure 1.26**) or **lifting swing gates**, **radial segments**, this means **radial gates with compression gate arms** or **radial gates with tension gate arms** (**figure 1.27**) with the profile of a circular section.

Figure 1.24: Mitre gate within upper gate

Figure 1.25: Flap gate

Essential elements for the planned considerations relating to lock gates are furthermore the **lock filling and lock drainage systems**. The following solutions are the most renowned:

Figure 1.26: Open and closed lift gates

Figure 1.27: Radial gate with compression gate arms

– The most straightforward solution is that four-sided sealing **sluice gates in the lock gates** regulate inlet and drainage of the water.

– So-called **circulators** feed the water into the lock chamber – water level equalisation is made via underground channels around the gates. Alternatively, it is distributed via side channels along the lock chamber or via **base channels** underneath the lock base. This means that water is fed at several points which significantly contributes to calming the water movement during filling.

– Radial gates with compression gate arms fill the lock chamber during upstream locking by means of **filling shells**, integrated within the lock chamber.

– More than 100 years ago, a completely different technology approach was introduced by means of the **Hotopp siphon lock**. Up to then, this principle was known for wine levers. A vacuum generated by pumping or flow either sucks water from or presses it into the lock chamber.

The systems in water technology shown and mentioned here are an extract of known designs. In the following, we will discuss some recently successfully implemented weir and lock concepts in more detail. In particular, these weirs are sluice gate weirs and gates. For locks, mitre gate locks as well as their filling and draining mechanisms are of prime focus.

2 WEIRS

Flow characteristics of water systems are controlled by weirs. They allow retention and flow control. Based on their large variety as mentioned in the previous chapter, we will now focus on sluice gate weirs but also on double gate, shutter and combined weirs.

2.1 Sluice gate weirs

Weirs consisting mainly of a stabilised gate leaf are used to protect against uncontrolled water levels. Flood water shall be retained in order to not inundate villages and towns or agricultural districts.

Furthermore, weirs might also be used to retain normal water for agricultural use. In a rather simple way as shown in **figure 2.001**.

To preserve navigability of waterways at sufficient depth for vessels, water retention might be required. In this instance, the weir gates are only lifted in case of floods to increase flow diameter and ensure drainage of potentially hazardous high water floods.

Figure 2.001:
Simple sluice gate weir

Weir gates are mainly subject to **undercurrents**, which means headrace operation. This allows flushing the water bed and level change on both sides – upstream and downstream. Height of weir gates is mainly adapted to the mean flood levels, but can also be adjusted to the highest flood levels so that overflowing occurs rather infrequently.

The sluice gate weirs prime task is to retain water. The prevailing headrace operation allows flushing the water bed and reduces required upkeeping.

Sluice gate weirs are built for both small and large weir widths – sliding gates up to 15 m and roller gates up to 20 m or up to 30 m, if fortifications are provided. They operate with one or two **connecting element(s)** (**figure 2.002**). It is important to observe the following rule of thumb: the ratio width to height. If the width of the weir gate is wider than the height, it is generally suspended at two points, since it could get wedged in case of single suspension. Beside the lifting of the gate leaf, the even more challenging pushing of the gate leaf in direction of the water bed must be considered during planning.

Force transmission can be made via connecting elements such as spindles, lantern gears, ropes, chains, lantern gear chains and lever arrangements. In case the gate leaves are also required to be pushed and lifted, spindles (**figure 2.003**) or lantern gears (**figure 2.004**) are often used.

Flood control reservoirs are subject to a particular construction type when used in combination with a **baffle** (**figure 2.005**). After opening, the undercurrent passes **by-pass seal sluice gates** also called **shutters** – with lateral, head and bed seal. If it is completely closed, the water can rise until it drains above the concrete edge of the baffle. Consequently, the baffle reduces the movable closing element and hence actuation technology. For locks, this

Figure 2.002: Sluice gates weirs with one connecting element

Figure 2.003:
Sluice gate weir with two spindles

Figure 2.004: Sluice gate weirs with two lantern gears

Figure 2.005: By-pass seal sluice gate in front of a baffle

Figure 2.006: View from the top onto a flood protection sluice gate: from left baffle – head block – head seal – sluice gate

principle is implemented at the lower gate of **shaft locks**. If the flow direction is changed due to draining flood water, then the sluice gate can be set up for **reverse operation**. In this case, lateral seals have to be provided at the outlet side.

Figure 2.006 shows the top view: Typically, a tripartite sealed sluice gate is additionally sealed horizontally at the top against the **head block** of the baffle.

Sluice gates with four-sided seals are also deployed as **submerged gates**. At dam walls (**figure 2.007**), high water pressure might be acting on or behind the sluice gate. The occurring friction force is accordingly high due to the hydrostatic force acting on the closing element. A further aggravating factor for the actuation technology is the weight of the long and consequently heavy connecting elements or their threadless extensions.

In general, once set up and put into service, only the pulling equipment and the water surface are visible (**figure 2.008**).

At user level, metal plates with respectively two holes designed for perfect sealing are provided for spindles at the height of caverns. Spur gearboxes mounted on top of the stems can be operated via actuators (**figure 2.009**).

Figure 2.007: River dam

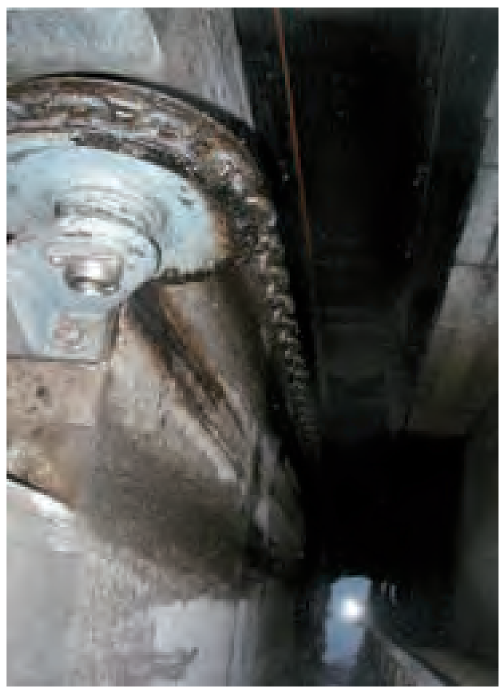


Figure 2.008: Suspension of a submerged gate

Figure 2.009: Actuator/gearbox combination option for a submerged gate

Among the sluice gate weirs, the **sliding gates** are by far the oldest versions and still quite widely spread until today. Their special advantage is that they are easy to build and cost less than other weirs. Their major drawback is the unfavourable sliding friction coefficient μ with regard to actuation technology. This coefficient is likely to occur due to the sliding of sluice gates on their guiding channels. This results in comparatively high friction forces and consequently driving forces. For the required tensile force, the basic formula still applies:

$$F_T = G + \mu * F_W$$

wherein
F_T = Tensile force
G = Weight of sluice gate
μ = Coefficient of sliding friction
F_W = Water force

Figure 2.010: Roller gates, top: Upper side with lifting eyes and rubber sealing strip,
bottom: Lower part with gasket and rubber sealing strip

Roller gates were developed due to the favourable rolling friction. The first known application was implemented in England [7] in 1880. Rollers are fixed to the sluice gate which then rolls on the guide runners (**figure 2.010**).

The rolling friction coefficient – also considering the additional journal friction – is more favourable than that of the sliding friction coefficient. The required driving forces are approx. 15 % below the values of sliding gates [6]. But this is not the only decisive factor. Experience had to be gained first since already slight deflection of the sluice gate – which might be even more significant for large weir widths – can lead to roller jamming. In particular **the wheel flange friction** (**figure 2.011**) might virtually correspond to a sliding friction as is likely to happen for temperature-dependent shaft reductions.

Figure 2.011: Wedge roller gate with running wheels and wheel flanges

Deflections might already occur when resting sluice gates on supports. A sluice gate of 12 m is only supported on one side by two rollers, rarely three. The later implementation of wheel flanges by lateral guides (DIN 1970-2/9.1) and the spherical version of running surfaces (DIN 19704-2/10.17) eliminated these problems so that roller gates can be deployed more often.

The question of lubrication and grease properties is of great importance. Using fixed installation of grease hoses to cams which allowed for regular relubrication was only a provisional solution, since bio-degradable greases were subject to early resinification.

The main reason for rejecting roller gates is the tendency of blocking rollers in case of insufficient lubrication.

There are personal preferences for specific weir types, but both weir types and other variations have their merits. Plant consultants and contractors will carefully consider the pros and cons. The decision will also be based on size and application. The following procedure is quite feasible:

- The torque required is very high. A roller gate should be favoured.
- Appropriate maintenance is not guaranteed. Then a sliding gate should be preferred.
- The high contact force would crush the slide strips. The decision goes in favour of a roller gate.

2.1.1 Characteristic impact on machine design

The following chapter leads to preparing sluice gate calculations. For designing hydraulic steel structures, DIN 19704, version May 1998, applies with the following parts:

- DIN 19704-1 Criteria for design and calculation
- DIN 19704-2 Design and manufacturing
- DIN 19704-3 Electrical equipment

Their demands are the basis for the following considerations and calculations.

Any deviations from the DIN clauses must be justified and the compliance of the alternative solutions must be proved.

The objective of driving torque determination in movable water installations is the provision of reliable results for direct manual operation by humans – mechanical operation via muscular strength – or indirect operation – pneumatic, electric or hydraulic automation. A sluice gate must be lifted, lowered and maintained at a certain position.

For this, the value of the dead load is required – the total weight – supplemented by corrosion protection agents, water, ice and pollution. This is called **permanent impacts**.

The total load to be handled is subject to **variable impacts**. Further inclusion must be made of **hydrostatic** and **hydrodynamic** impacts.

The hydrostatic impact is the hydrostatic pressure or hydrostatic force generating the friction and buoyant force.

Hydrodynamic impacts occur in case of water movements. This includes the overflow and undercurrent of the sluice gates. The pull for example results from the undercurrent. Lock gates have to cope with swell and downsurge generated by vessels entering and leaving the lock. In turn, hydrodynamic impacts are generated by water displacement of the closing element. This happens when opening and closing a lock gate or when lifting and lowering a sluice gate. Variable impacts include water load, applied ice load, ice pressure, influence of temperature and wind.

Predefined impacts are movement prevention by foreign bodies and/or single-sided retaining or temporary activation of two-sided driven closing elements.

Exceptional impacts of actuators in case of an accident. For one-sided closing element blocking due to jamming or freezing, the highest actuator impact which could lead to installation damage must be considered (DIN 19704-1/5.5).

For the following identification of components, the definitions of DIN 19704-1/8.1 must be observed: The most unfavourable position of the closing element must be assumed. This applies irrespective of the closing element being static or in movement (uniform, accelerated, delayed). On the other hand, variable impacts occurring at the same time must only be considered if might actually happen (compare DIN 19704-1/table 5, index 1).

The following sections 2.1.1.1. through 2.1.1.9. Include the **characteristic variables** of the load to ensure the **load-bearing capacity**. The closing pressure – section 2.1.1.10 – is part of the **proof of usability.**

2.1.1.1 Dead load

Depending on their immersion depth, gate leaves are subject to differing hydrostatic pressure – the pressure is higher at the water bed than at the surface level. For this reason, the gate leaves must be reinforced by horizontal and vertical profiles (**figure 2.012**). Since the bars used have identical moments of inertia, each lock will adopt the same hydrostatic pressure to avoid uneven deflection. Consequently, they are at a shorter distance at the river bed level than at the upper gate edge. The set-up is called **bar design**.

When high ice pressure is to be expected, the upper gate part should be reinforced accordingly.

The **dead load** of the sluice gate F_{DL} includes the weight of the gate leaf F_{GL}:

$$F_{GL} = l_{GL} * h_{GL} * w_{GL} * \rho * g$$

wherein
F_{GL} = Dead load of gate leaf
l_{GL} = Length of gate leaf
h_{GL} = Height of gate leaf
w_{GL} = Width of gate leaf
ρ = Density of steel
g = Gravitational force

Figure 2.012: Tailwater side of a sluice gate with bars, slide and sealing strips

as well as the weight of the following supplements:

- Weights of spindles or lantern gears, bars, joists, cross ribs, suspensions, slide and sealing strips and connecting equipment such as screws and welded seams.
- According to DIN 19704-1/5.1, a supplement of 10 % has to be calculated for coatings, adhesion of water and ice, encrustation of fouling matter and pollution. The total amount of the added weight can vary depending on the application and is defined by the contractor. Extra weights are not always available to their full range and cannot be included for gate lowering which would be an advantage.

2.1.1.2 Hydrostatic force

Hydrostatics is one of the basic elements of fluid mechanics. It deals with liquids which do not change their position in relation to the container wall or a body encompassing the liquid. This consideration forms the basis for calculating the fluid forces of still fluids on fixed walls.

The **hydrostatic pressure p** is identical at any spot of a horizontal reference surface across a liquid. This means, it is a scalar variable regardless of the direction, only depending on the location (**figure 2.013**). Air pressure p_0 shall not be considered, since it is effective at any location, of same level and has no impact on the forces resulting from the static pressure. The hydrostatic pressure p is the ratio of normal force dF and pressure area dA.

$$p = \frac{\text{normal force}}{\text{area}} = \frac{dF}{dA}$$

At the centre of gravity of the liquid section in figure 2.013, with cross-sectional area dA and the height h, the following downward acting weight dG applies – as a result from the liquid weight:

$$dG = \rho * g * dV = \rho * g * h * dA$$

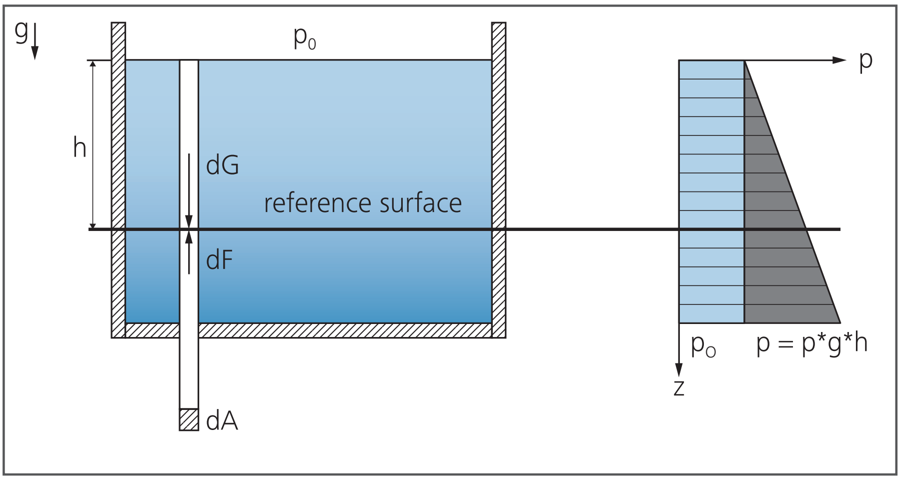


Figure 2.013: Hydrostatic pressure within a liquid and the resulting so-called hydrostatic pressure triangle

Since the liquid is unmoved, the weight dG is compensated by a uniform vertical compression force dF acting from below on the pressure area dA.

$$dF = p * dA$$

$$dG = dF$$

Resulting:

$$\rho * g * h * dA = p * dA$$

$$p = \rho * g * h$$

- The hydrostatic pressure increases linearly with the liquid depth. For this reason, it is called gravitational pressure.
- At points of same depth h, the same pressure p applies.
- On a vertical level, it generates the hydrostatic buoyant force. On a horizontal level, the pressure acts upon the closing element and the steel frame resulting in friction forces at the sluice gate guiding channels. The hydrostatic pressure distribution at the gate leaf is considered. It is clearly presented by a **hydrostatic pressure triangle** (**figure 2.014**).

The hydrostatic force F_H arising from the hydrostatic pressure is a vector in normal relation to the pressure area.

We limit our consideration of hydrostatic forces on plane and vertical walls. **Figure 2.015** shows the ratios as clearly as possible.

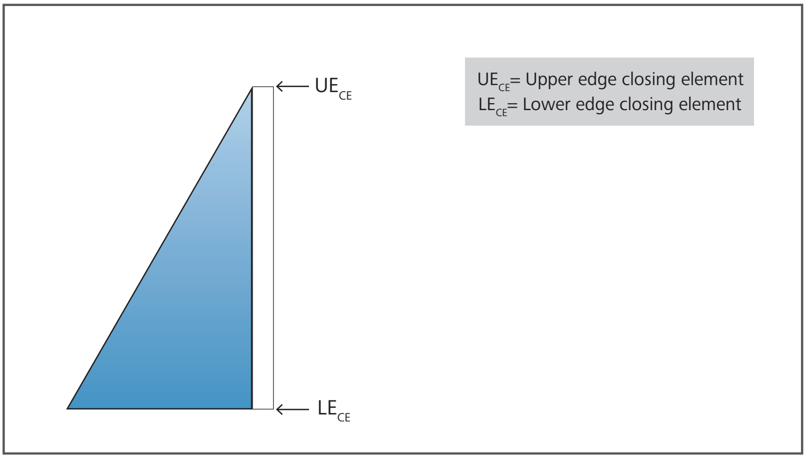


Figure 2.014: Hydrostatic pressure triangle at a dam gate

The area element dA of area A with coordinates x and z, pressure p acts with:

$$p = \rho * g * z$$

According to the general pressure definition, the compression force dF acting perpendicularly on area element dA amounts to:

$$dF = p * dA = \rho * g * z * dA$$

The total compression force F is achieved by integrating all compression forces dF across area A:

$$F = \int_{(A)} dF = \rho * g \int_{(A)} z * dA$$

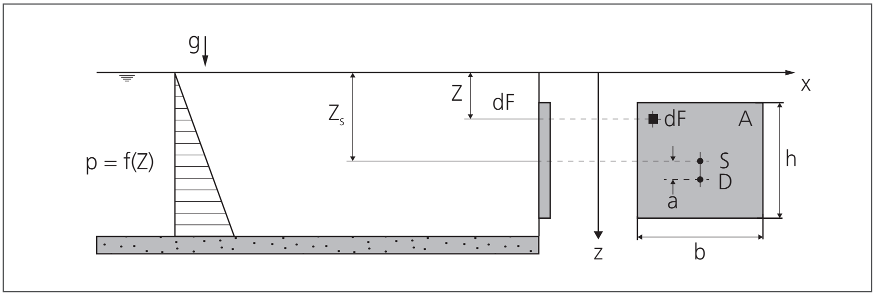


Figure 2.015: Flat, vertical wall: hydrostatic impacts

The integral

$$\int_{(A)} z * dA$$

represents the "static moment" of area A in relation to the x-axis. According to the moment theorem, the following applies:

$$\int_{(A)} z * dA = z_s * A$$

whereas z_S represents the z coordinate of centre of area S. The following formula results for compression force F:

$$F = \rho * g * z_s * A$$

or

$$F = \rho * g * z_s * A = p_s * A$$

- The lateral force F acting on an **even area A** results from multiplying the surface area A and the pressure p_S in the centre of area S.
- The calculation for a standard rectangular sluice gate is as follows:

$$F_H = A * p_H$$

$$F_H = l * h * h/2 * \rho_{Wat} * g$$

wherein

A = Wetted area

p_H = Hydrostatic or water pressure

F_H = Hydrostatic force

l = Length of sluice gate

h = Height of water

ρ_{Wat} = Density of water

g = Gravitational force

Since the hydrostatic pressure p increases linearly to the liquid depth z, an asymmetric pressure distribution forms across the area under compression A. This means that the calculated force F_H does not interfere with centre of area S but with the lower **pressure centre D**.

For a **rectangular load** consisting for example, of wind pressure, the pressure distribution is ideally constant across the complete height of the sluice gate. The resulting force impacts on the **centre of gravity of the rectangle**, which means at half the height of the sluice gate. For a **triangular load** consisting of hydrostatic pressure, the pressure distribution corresponds to the hydrostatic pressure triangle. Therefore, the hydrostatic pressure impacts on the **centre of gravity of the triangle**, meaning at the lower third.

For stability calculations or for submitting proofs of gate leaf and bar design, the torque acting on the frame and closing element can be calculated while assuming the hydrostatic force at the centre of gravity of the hydrostatic pressure triangle at 1/3 h (figure 2.016).

This is considered as the **hydrostatic force**. By considerable overflows and additional loads like applied ice loads, the centre of gravity of the overall load is shifted upward.

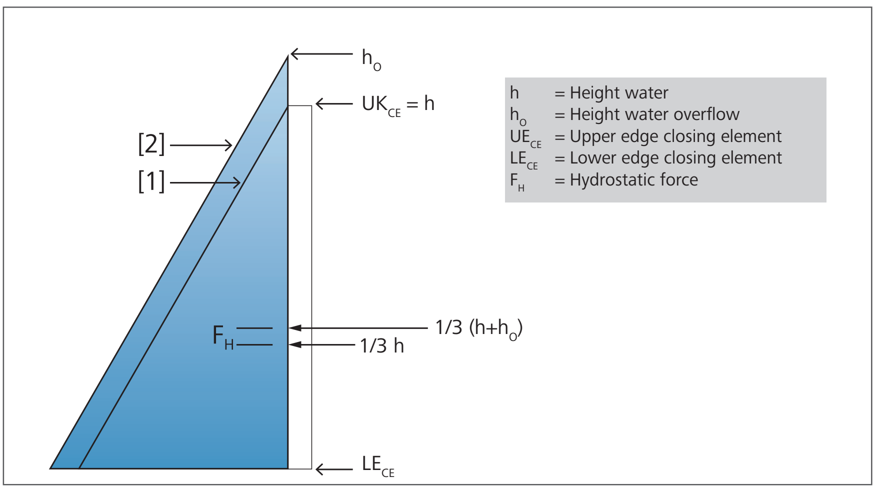


Figure 2.016: Hydrostatic pressure triangles relating to hydrostatic impacts onto a dam gate; 1 with, 2 without overflow

If the full height of the sluice gate is available without compensation, the wave effect – swell and downsurge = adding upstream side while deducting downstream side – does not have to be considered (DIN 19704-1/5.2.2.b).

If a downstream side counterforce is always present, the lowest occurring force can be subtracted from the headwater side counterforce (**figure 2.017**).

$$F_H = F_{HHW} - F_{HTW}$$

2.1.1.3 Friction forces from hydrostatic force

The sluice gate slides into two vertical U profiles. To limit lateral movement in any direction as far as possible, guides are mounted at three sides to the outer left and the outer right of the gate leaf. These are named **guide blocks** (**figure 2.018**), unless we are dealing with traversing **guide strips** occurring mostly at the sealing face. Often simple strips are used.

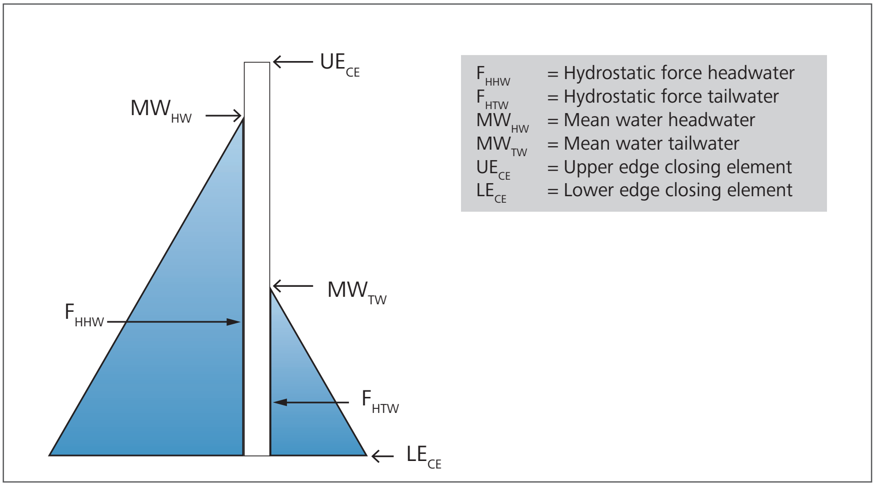


Figure 2.017: Opposing hydrostatic pressure triangles at a closing element

The guide blocks or strips are made of either steel, copper, synthetic material, wood or rubber. Their objective is to improve the sliding or sealing behaviour of steel sluice gate – steel frame combinations which can only be used with maximum effort.

If only blocks are used at the sealing face which lead to open sections toward the upper and lower section of the gate leaf, a rubber profile (with good sealing properties but difficult to slide on) must be placed together with a clamping strip. As a matter of principle, it can also be placed next to a slide strip. Another option would be using a beaded rim seal, a three-point seal or a rubber profile in the shape of a bulb and tail seal. The bulb and tail seal (**figure 2.019**) can be made with or without a hole at different shore hardness levels with diverse characteristics. Pretension can be adjusted. The sliding and sealing procedure can be from upstream or downstream.

Figure 2.018: Sluice gate with guide blocks

Figure 2.019: Sluice gate with suspension, slide strip and sealing strip in bulb and tail profile

The hydrostatic pressure jams the sluice gate and sealing onto the guide frame. Depending on the material combination, different friction factors occur. Depending on the hydrostatic pressure, the resulting friction forces may also differ. The selected actuator must operate the sluice gate and, consequently, must overcome the static friction. Long service life might even lead to adhesions similar to **microscopic positive locking** in the event of significant roughness. This makes the transition from static friction to sliding friction more difficult. This is one of the reasons, why the actuator to be selected should have a reserve to provide an increased **unseating torque** for a short time,if required.

2.1.1.3.1 Friction force of support strips from hydrostatic force

The following applies for the hydrostatic frictional application force F_{FAH}:

$$F_{FAH} = \mu * \mu_0/\mu * F_H$$

wherein
μ_0 = Static friction
μ = Sliding friction
F_H = Hydrostatic force

DIN 19704-1/6 indicates the ratio of the static friction coefficient and sliding friction coefficient for different material combinations: μ_0/μ. The standard indicates a maximum value for μ which has to be considered for pulling and pushing the sluice gate, and a minimum value to be considered for maintaining the closing element.

2.1.1.3.2 Friction force of guide runners from hydrostatic force

Depending on the safety requirement and mounting quality, the plant consultant will only assume single digit percentage values x of the hydrostatic frictional application force F_{FAH} for the **hydrostatic guide runner friction** F_{GRH}, acting on the guide runners, located at the narrow side of the sluice gate with slight lateral distance to the frame:

$$F_{GRH} = x * F_{FAH}$$

wherein
F_{FAH} = Hydrostatic frictional application force
x = Percentage fraction of F_{FAH}

In case of symmetry, lateral guidance friction is theoretically unavailable.

2.1.1.4 Buoyant force

Constructors for hydraulic steel structures will design the supporting structure for gate leaves to avoid undesired **hollow spaces** for preventing unintentional floating. This could lead to surprisingly high torques for the closing movement. In the past, this type of sluice gates had to be subsequently burdened with concrete seals to allow lowering. On the other hand, specifically designed hollow spaces can be used to reduce the required tensile force.

This has to be carefully considered.

It is quite common to include hollow spaces into sliding gates, shutters and fish-belly flap gates. In mitre gates of locks, the hollow spaces help to reduce the static load and thus the friction in pivot bearings.

Buoyant force is basically available in two types: Depending on the type of design, one of the two types might be included. In accordance with DIN 19704-1/5.2.1, the buoyant force must be checked for all construction components submerged in water and considered if applicable. The buoyant force helps when lowering the gates.

2.1.1.4.1 Hydrostatic buoyant force

The hydrostatic buoyant force depends on the respective design type, the size and the direction of the resulting hydrostatic pressure. Examples for better illustration below (**figure 2.020**).

In all three cases, the sluice gate is submerged on one side up to the upper edge. No relief is given from the opposite side. The frictional application force is at the highest level in the position shown.

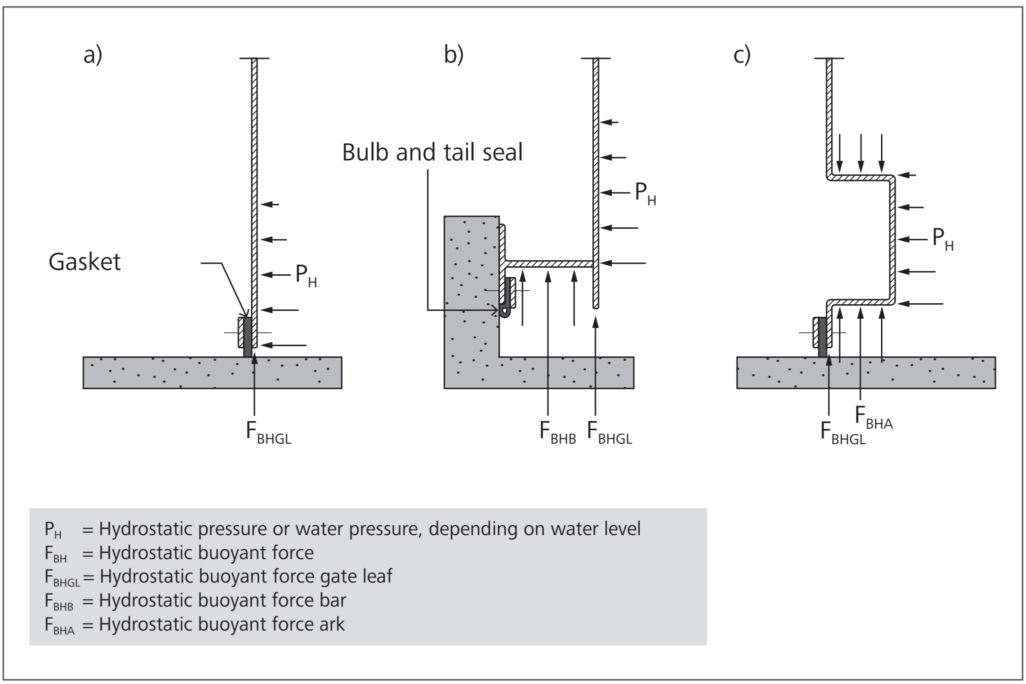


Figure 2.020: Buoyant force at a sluice gate: a) sluice gate with even gate leaf and gasket, b) sluice gate with sill sealed obturation, c) sluice gate with ark or trapezoidal profile

In example a), the vertical hydrostatic pressure and thus the hydrostatic buoyant force only acts below the gate leaf. The buoyant force is relatively low. If a gasket is mounted on the upstream side, no buoyant force type is available when the gate is closed.

Calculation to be made:

$$F_{BHGL} = A_{GL} * p_H$$

$$F_{BHGL} = l * w_{GL} * h * \rho_{Wat} * g$$

wherein
F_{BHGL} = Hydrostatic buoyant force of gate leaf
A_{GL} = Area of gate leaf
p_H = Hydrostatic pressure
l = Length of sluice gate
w_{GL} = Width of gate leaf, plate thickness
h = Height of water
ρ_{Wat} = Density of water
g = Gravitational force

In example b), in addition to F_{BHGL} the buoyant force acts across the complete bar width and length. The buoyant force considerably supports the opening operation. It is calculated for the total bar as follows:

$$F_{BHB} = l * w_{Bar} * h * \rho_{Wat} * g$$

wherein
F_{BHBar} = Hydrostatic buoyant force of bar
l = Length of sluice gate
w_{Bar} = Width of bar
h = Height of water
ρ_{Wat} = Density of water
g = Gravitational force

In turn, the buoyant force for the gate leaf is rather low.

In example c), larger differences in height are available for larger cross sections. Calculation is made as follows:

$$F_{BHA} = l * w_A * (h_{ALE} – h_{AUE}) * \rho_{Wat} * g$$

wherein
F_{BHA} = Hydrostatic buoyant force of ark
l = Length of sluice gate
w_A = Width of ark
h_{ALE} = Height of water of lower ark edge
h_{AUE} = Height of water of upper ark edge
ρ_{Wat} = Density of water
g = Gravitational force

Figure 2.021: In this case, tilting of sluice gates achieves an increase in buoyant force when opening for rear inflow

The buoyant force for larger dimensions and several bars should be verified. The buoyant force below the gate leaf can hereby be neglected.

Inclining the sluice gates which corresponds to an enlargement of the projected and thus the buoyed up area results in an increase of the hydrostatic buoyant force and consequently in a decrease of the tensile force required (**figure 2.021**). Automatic opening could be the result of excessive inclination.

In the examples above, the buoyed up areas $l * w_{GL}$, $l * w_{Bar}$ and $l * w_A$ were decisive, in this example it is $l * w_A$ (**figure 2.022**). Calculation to be made:

$$F_{BH} = l * w_A * h_v * \rho_{Wat} * g$$

$$h_v = h * \sin \alpha$$

After opening and subsequent water level equalisation, the hydrostatic buoyant force converts into buoyancy by water displacement.

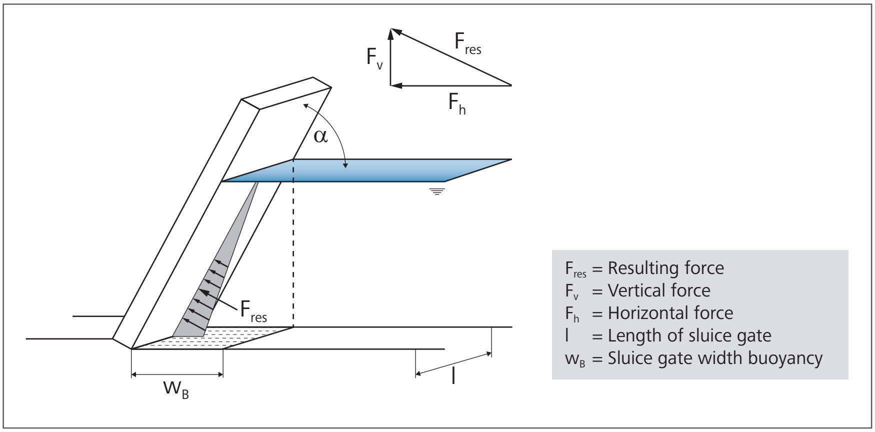


Figure 2.022: Buoyant force at a tilted sluice gate

2.1.1.4.2 Buoyancy by water displacement

Buoyancy by water displacement is the basis of navigation. According to Archimedes' principle, the upward buoyant force exerted on a body is equal to the weight of the fluid that the body displaces. It is independent of the depth of submersion. A precondition is that the submerged part of the closing element is completely surrounded by water. For the gate leaf, this means:

$$F_{BWD} = l * w_{GL} * h * \rho_{Wat} * g$$

wherein

F_{BWD} = Buoyancy by water displacement
l = Length of sluice gate
h = Height of water
w_{GL} = Width of gate leaf
ρ_{WAT} = Density of water
g = Gravitational force

The buoyancy for the stabilising bars is to be added.

The various construction types (refer to figure 2.020) are shown in closed position, i.e. not surrounded by water. There is no buoyancy by water displacement. The buoyancy by water displacement for the height of the water level equalisation of both sides of the closing element can only be calculated after the opening operation. Due to the low plate thickness, it can mostly be neglected.

2.1.1.4.3 Particularities of the folded plate

Today, a gate leaf shaped as folded plate is at the forefront (**figure 2.023**). This set-up can replace the more costly bar construction. Since the welded seams are predominantly

Figure 2.023: Folded plate gate

located at the front edges, anti-corrosion protection is quite easy. Moreover, it is more resistant to wear and tear. This considerably reduces maintenance costs. Use as by-pass or inspection gate is quite feasible. In turn, lift gates can generate significant high output forces – pull – due to undercurrent. When using gate leaves in mitre gates, sealing of the drive shaft running vertically through the folded plate is quite problematic. For this reason, by-pass gates are preferable [6].

Figure 2.024 shows the generation of the resulting forces depending on the type of installation of the folded plate. Thanks to the hydrostatic buoyant force in locks, the pivot bearings of closed mitre gates can be relieved for the opening operation. Sluice gates receive support for lifting (refer to figure 2.020). However, in insufficiently dimensioned systems, ballast had to be added for closing in order to overcome the unexpectedly large buoyant force.

2.1.1.5 Water load to be lifted

If a sluice gate is lifted or lowered, the water is displaced sideways across the bar edges. A so-called "head of water" is not transported. Not even for submerged gates. For determination of the hydrostatic force or the frictional force (dependent on the head of water), the water surrounding the construction is considered.

The water load to be lifted is only an issue, if a bar filled with water is lifted above the water surface. Therefore, constructors for hydraulic steel structures will mount profile steels openly downward and vertically drill bars which are open at the upper edge to avoid unnecessary water transport during the lifting operation. The water will virtually drain completely through the drain holes. Since pollution may occur and trapped water cannot

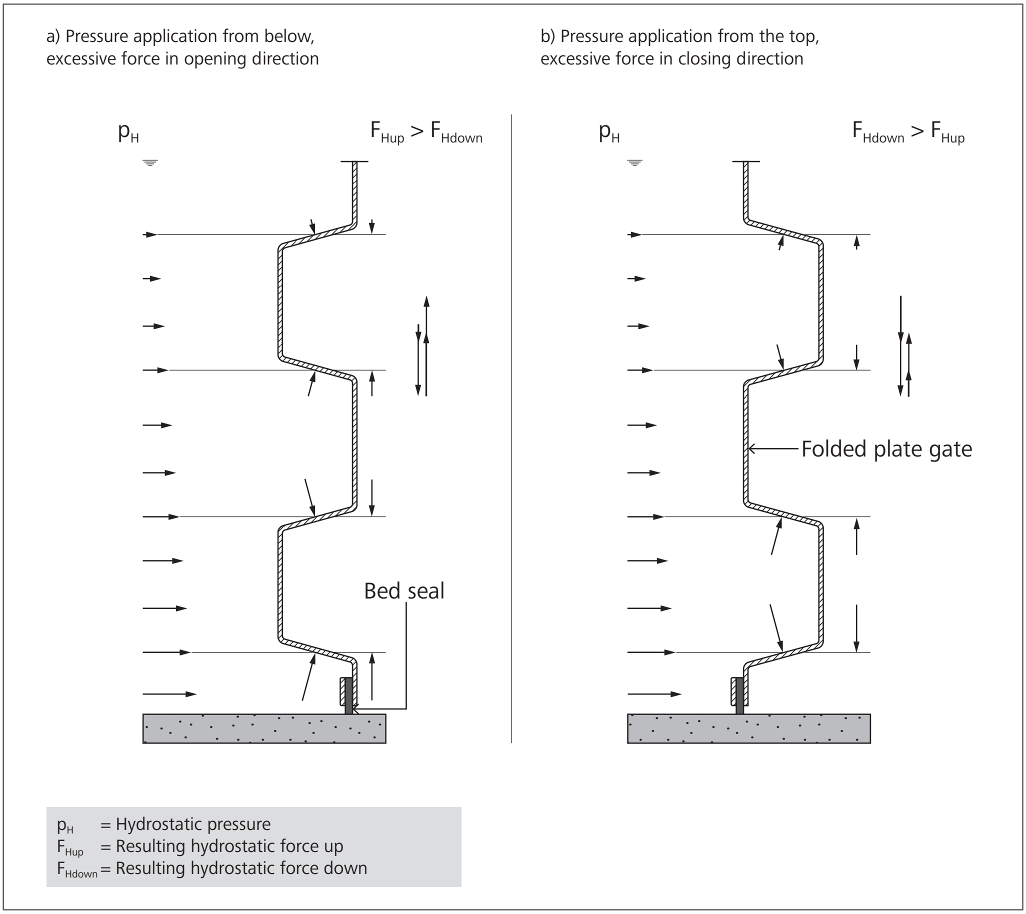


Figure 2.024: Hydrostatic pressure and resulting buoyant force at a folded plate gate

be securely drained, the potential water load for the bars should be calculated. Depending on the type of construction or the number of bars and the evaluation of the necessity, the multiple has to be inserted:

$$F_{WL} = l * h_{Bar} * w_{Bar} * \rho_{WAT} * g$$

wherein

F_{WL} = Water load
l = Length of sluice gate
h_{Bar} = Height of bar
w_{Bar} = Width of bar
ρ_{WAT} = Density of water
g = Gravitational force

In general, ten percent additional dead load of the water to be transported must be considered.

2.1.1.6 Applied ice load

Residual ice must be considered by ten percent additional dead load (DIN 19704-1/5.1). Applied ice load can be treated as additional dead load increase by the customer. If in winter a weir is used for headrace flood drainage, it might have to lift a compact ice load until it breaks away (**figure 2.026**), apart from the applied ice load on cross bars (**figure 2.025**). Prior to starting operation, experienced staff will relieve the gate of the ice load using a pickaxe or a chainsaw.

Due to overflow, the rear part of the closing element, which is not submerged, might be massively ice crusted at the sides of the guide bars (**figure 2.027**).

Calculation to be made:

$$F_{EL} = l_{Ice} * h_{Ice} * w_{Ice} * \rho_{Ice} * g$$

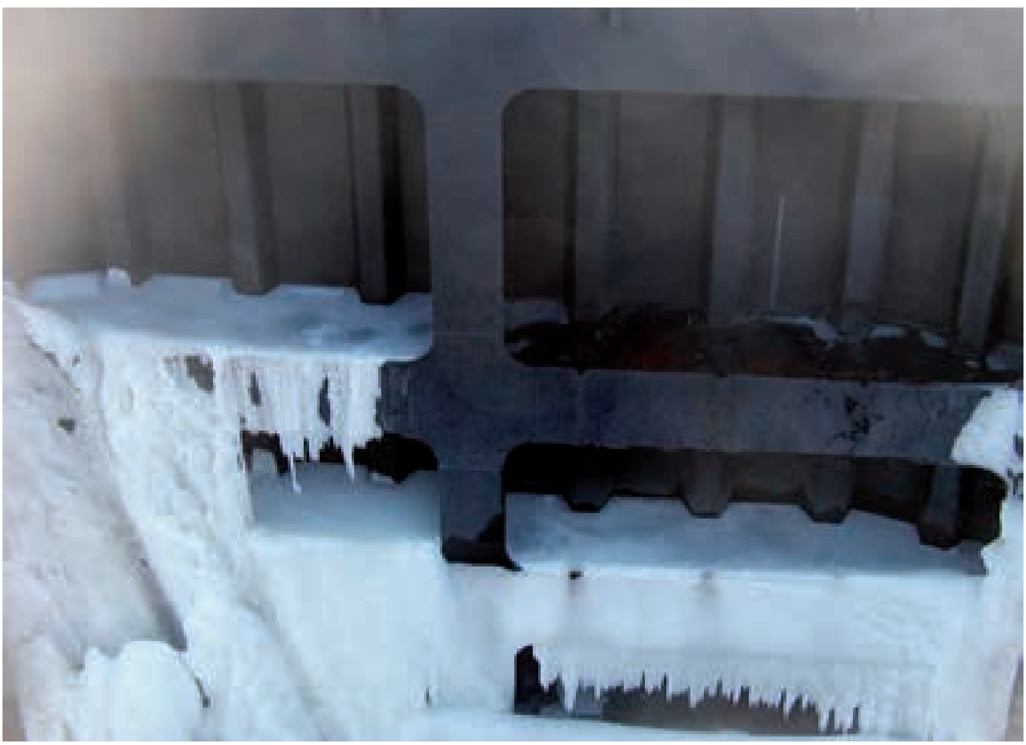


Figure 2.025: Applied ice load on cross bars

Figure 2.026: Headwater side closed ice crust apart from the weir

Figure 2.027: Applied ice load on the weir rear

Figure 2.028: Sluice gate weir with cable drums for cables of slide and sealing rail heaters

wherein

F_{IL} = Applied ice load
l_{Ice} = Length of ice
h_{Ice} = Height of ice
w_{Ice} = Width of ice
ρ_{Ice} = Density of ice
g = Gravitational force

As a measure of precaution, sliding and sealing rail heaters are installed. In a first step to facilitate sliding and above all to prevent tear of frozen seals. According to DIN 19704-2/7.1, ice formation should be avoided until reaching an air temperature of –20 °C. The heaters are installed at the rear of the carrier for the slide and sealing strips. Drums allow reeling and unreeling supply cables together with the sluice gate movement (**figure 2.028**).

2.1.1.7 Ice force

If the water surface is completely frozen in stagnant water, the ice pressure and consequently the ice force might additionally strain the frame and the sluice gate (**figure 2.029**).

Figure 2.029: Ice force applied onto a weir

Often the ice force acts from both sides. If applied at different levels, the torque acting on the frame and the sluice gate can lead to deformation (**figure 2.030**).

Figure 2.030: Impact ice force onto an inspection gate from both sides at different heights

Figure 2.031: A longitudinal cut into the ice brings relief for the ship lift at Niederfinow and the subsequent canal bridge. One man acts as security officer.

In civil engineering constructions for water applications it is often required to relieve the ice pressure on walls by separating cuts (**figure 2.031**).

The impact of ice force can be compared to that of the hydrostatic force. The ice force must only be considered for dimensioning actuators if the structure is to be equipped for winter operation and therefore provided with ice prevention systems.

$$F_{Ice} = l_{Ice} * h_{Ice} * p_{Ice}$$

wherein
F_{Ice} = Ice force
l_{Ice} = Length of ice
h_{Ice} = Height of ice
p_{Ice} = Ice pressure

Agitators and air bubble system are used in locks to avoid ice formation. To redirect ice floes via the openings of downstream sluice gates (**figure 2.032**), the sluice gates of the upper

Figure 2.032: Lower lock gates with drains for ice and flood drainage

gate are operated. Flood water can also be redirected this way or through the additionally opened lower gate.

2.1.1.7.1 Friction force of support strips due to ice force

The ice force is considered in addition to the hydrostatic force. The frictional application force is calculated as follows:

$$F_{FAIce} = \mu * \mu_0/\mu * F_{Ice}$$

wherein

F_{FAIce} = Frictional application force due to ice force
μ_0 = Static friction
μ = Sliding friction
F_{Ice} = Ice force

2.1.1.7.2 Friction force of guide runners due to ice force

$$F_{GRIce} = x * F_{FAIce}$$

wherein
F_{GRice} = Guide runner friction force due to ice force
F_{FAIce} = Frictional application force
x = Percentage of F_{FAIce}

2.1.1.8 Pull

When opening a sluice gate, a small cross section is available at first. According to Bernoulli

$$p_H + p_{HD} = \text{const.}$$

$$p_H + 1/2\ \rho_{Wat} * v^2 = \text{const.}$$

The hydrodynamic pressure p_H and thus the flow velocity v is at the highest level at the moment of the lowest hydrostatic pressure – i.e. in case of gate undercurrent:

$$p_{HD} = 1/2\ \rho_{Wat} * v^2$$

wherein
p_H = Hydrostatic pressure
p_{HD} = Hydrodynamic pressure
ρ_{Wat} = Density of water
v = Flow velocity of water

Due to undercurrent, a pull effect is created at the lower gate edge. The actuator has to overcome the pull effect when leaving the closed position. In unfavourable conditions – depending on the shape of the lower gate edge for example – the pull effect can "be a multiple of the dead load" [9]. When lifting, the highest value will be reduced and at one stage, the buoyant force will prevail. Due to major uncertainties caused by different basic conditions, DIN 19704-1/5.2.2 recommends using model tests (**figure 2.033**).

Due to the complexity of the pull load effect, a simplified calculation can be selected to achieve reasonable results. These results are already assumed by plant consultants in comparable values and determined by further calculations. Calculation to be made:

$$F_{Pull} \approx l * w_{Bar} * p_{Pull}$$

wherein
F_{Pull} = Pull force
l = Length of sluice gate
w_{Bar} = Width of bar
p_{Pull} = Pull (negative pressure)

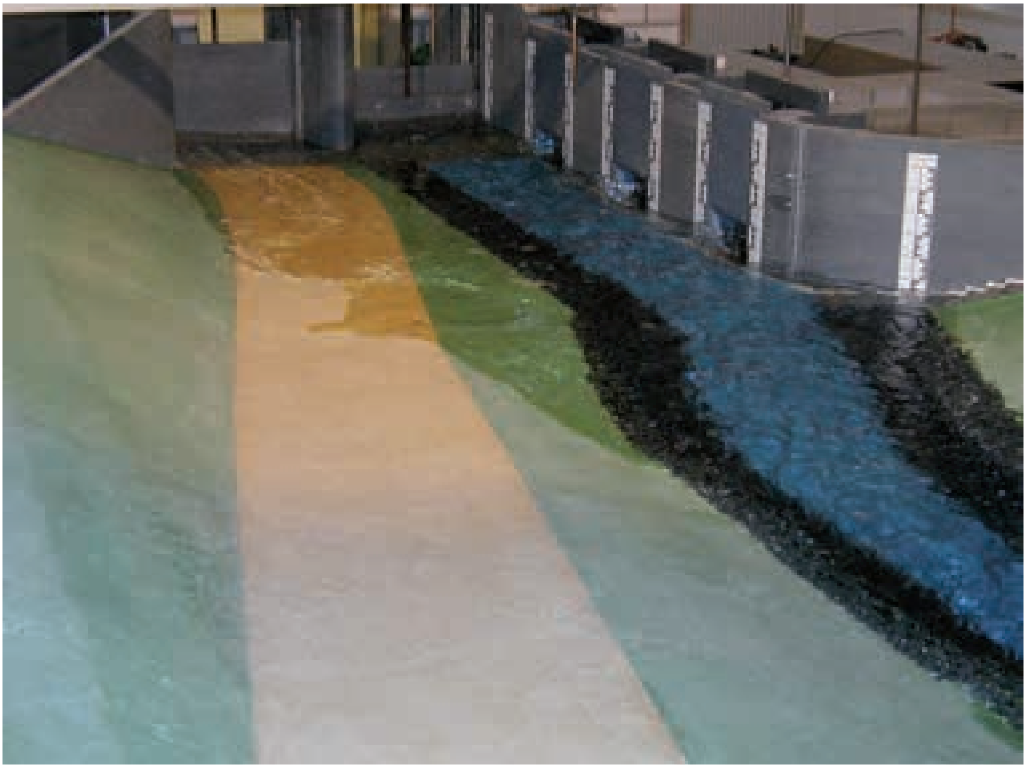


Figure 2.033: Water distribution model (1:10), from left: sluice gate weir, double gate weir, 3 side weirs

The down-pull effect is the dreaded interaction between pull and buoyant force. After opening, the sluice gate can adopt positions in which it is pulled down or pushed up in turns. The fluctuating flow velocity and the resulting friction change can be identified as causes. Furthermore, alternating swirls occur behind the closing element. Noise is caused by vibration which might even be audible at long distances. However, structural damage and increased wear of parts are considered as severe consequences.

Since low opening widths of sluice gates are connected with high flow velocities, plant consultants request a minimum opening size, i.e. to operate the first 30 cm of a total of 3 m without interruption. Insiders usually apply this rule of thumb of 10 %. By design, hollow spaces or suspensions are provided to fit weights at a later date to settle movement. Companies in this sector promote the term **vibration damping**. Inclined gate leaves and defined breakaway edges, e.g. as gasket placed inverted to the flow direction, could be considered as favourable.

The before mentioned vibration also occurs for overflowing of closing elements. The formation of alternating pulsating air cushions and vacua are the consequence. Flow deflectors avoid formation of underpressure for the most part. It is advised to avoid simultaneous overflow and undercurrent when controlling weirs.

2.1.1.9 Wind force

According to DIN 19704-1/5.2.12., wind impact does not need to be considered for closing element dimensioning. However, wind force can have an impact on the driving force. The impact of **wind force** is comparable to the hydrostatic force and ice force. Calculation is based on:

$$F_W = c_f \cdot * q * A$$

wherein
F_W = Wind force
c_f = Aerodynamic force coefficient
q = Dynamic pressure
A = Area of projection

2.1.1.9.1 Friction force of support strips due to wind force

The friction force of support strips due to wind power results from:

$$F_{FAW} = \mu * \mu_0/\mu * F_W$$

wherein
F_{FAW} = Frictional application force due to wind
μ_0 = Static friction
μ = Sliding friction
F_W = Wind force

Wind usually changes direction. It may press onto the gate leaf from different angles and may even act against the hydrostatic force, thus having a relieving effect.

Wind may cause pressure and pull.

According to DIN 19704-1/8.1, calculation must be made at its highest force acting in the most unfavourable position of the closing element.

2.1.1.9.2 Friction force at guide runners due to wind force

$F_{GRW} = x * F_{FAW}$

wherein

F_{GRW} = Guide runner friction force due to wind
F_{FAW} = Frictional application force due to wind
x = Percentage fraction of F_{FAW}

2.1.1.10 Contact force bed seal

Under normal conditions, water is blocked by operating a gate leaf perpendicularly downward against a steel beam fixed in the water bed. This could be for example a T beam or a double T beam placed in a concrete bed. Typically, a **gasket** (dimensions e.g. 80 mm x 10 mm) is secured in its total length upward by a welded rail and laterally mounted by means of a clamping strip to the lower gate edge, ensuring an overlap of 3 mm to 5 mm [3]. Under load, this small strip is compressed and has a sealing effect (**figure 2.034**).

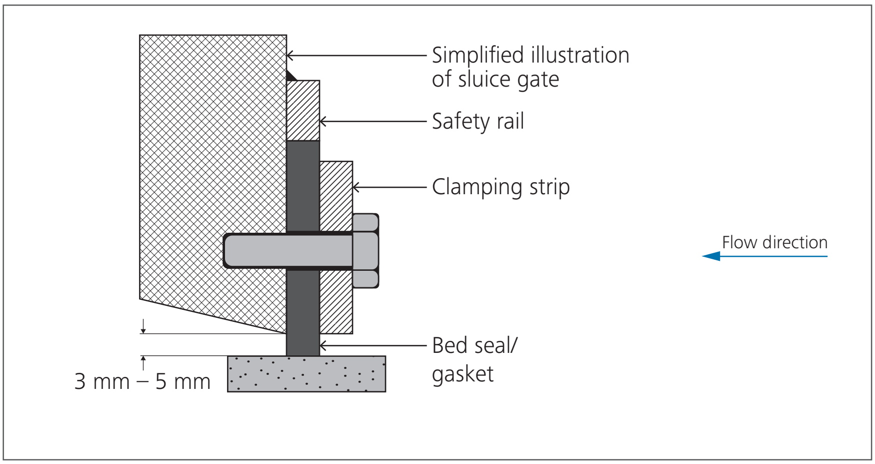


Figure 2.034: Bed seal

In this sealing position, the end position switch within the actuator is set. For reasons of safety, the torque switch with its set torque remains as second function switch in waiting position. In the event of sluggishness occurring prior to end position tripping, the same torque switch will cut off the actuator while signalling a fault.

In accordance with DIN 19704-1/7.6.3, the closing force for gaskets results from all vertical forces. It must at least be equal to the required **bed seal contact force**:

$F_{CB} = l * p_{CLB}$

wherein

F_{CB} = Bed seal contact force
l = Gate length
p_{CLB} = Bed seal closing pressure

Verifying the closing force is part of the **proof of usability**.

2.1.1.11 Resulting force

Figure 2.035 shows a schematic representation of all possible impacts on the closing element. Opening and closing forces can be calculated as follows:

$$F_O = F_C = \pm F_{DL} \pm F_{FAH} \pm F_{GRH} \pm F_{BH} \pm F_{BWD} \pm F_{WL} \pm F_{IL} \pm F_{FAIce} \pm F_{GRice} \pm F_{Pull} \pm F_{FAW} \pm F_{GRW} \pm F_{CB}$$

Once all component forces have been recorded as described above, decision has to be made on occurrence and algebraic sign in accordance with closing element position and

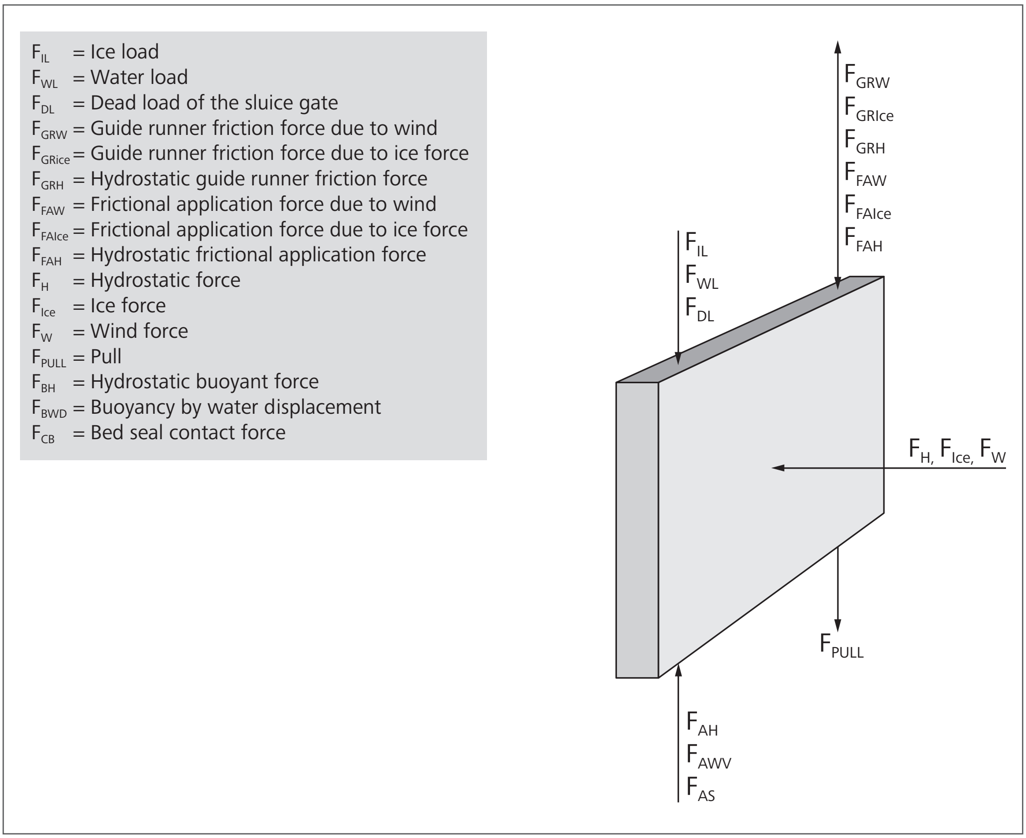


Figure 2.035: Significant impacts onto the closing element

direction of movement: I.e. the force in question is either considered as load (+) or support (-). In case of double suspension of the sluice gate, the result divided by 2 is the opening or closing force per connecting element.

2.1.2 Tender for an example

A weir is used to control the water level providing a multitude of applications. Years ago, a weir was built at the same location. Today, it no longer meets the requirements. A more solid solution is required. Or the environmental, working and living conditions have changed, creating new demands. New civil engineering constructions for water applications will have to interfere with nature. The customer will comment on this.

Who is the main contractor? There are several competent authorities such as the National Inland Waterways Authority, Waterways and Shipping Authority, River and Dam Authorities, Flood Protection Authorities, Environmental Protection Authorities. Now, the customer looks for a consultant – the one which is close and whom they have known for years? No, a proper tender is drawn up. **A consultant was found**. Not the one next door, but one with expert knowledge as is shown by the references. One that is known for thorough work. He or she discusses characteristics which are part of the **product requirements**.

2.1.2.1 Marginal conditions for the product requirements

Standards, design requirements as well as loads and functions, used as foundation for planning and construction, are agreed on and specified in detail for calculation and planning criteria. DIN 19704 – Hydraulic steel structures is ranking top. With reference to the sea level, the extreme high water EHW is assumed at 53.00 m normal height null (NHN) plus 0.20 m overflow. Statistically, this might occur once in a century, according to former definitions.

Incidentally, it should be possible to completely open sluice gates and close them again immediately if required. This corresponds to one **cycle** within actuator technology. The **measured water level**, the highest calculated flood MHW amounts to 52.80 NHN. The sluice gate is cleared infrequently – 6 times a year – and closed again. The target supply or **full supply level** amounts to a mean water level MW at 52.50 m NHN. In case of differing hydrostatic pressure, the water level is increased or reduced several times a year by 0.50 m and more frequently during the week by 0.30 m and 0.20 m. Exact figures are available. This corresponds two 24 double strokes. Altogether, 30 double strokes must be executed per year.

> ***These references must be realistic! Exaggerated requirements lead to excessive dimensioning.***

These references are subsequently laid down in the weir log. The bed beam is positioned at 50 m NHN.

Furthermore, definitions on ice, wind and occurring bed load are made. The sluice gate of 4 m should warrant for a **rough reach**. According to the available references, the sluice gate strain is at **medium level**.

And moreover, the following is often neglected: Definition of the **operating capacity** of the system. At **extreme high water**, the sluice gate should be capable of being operated at least two strokes without longer interruption! Hereby, the **operating speed** should be v = (200 … 400) mm/min.

Of course, assumptions are made about the specification of the system. The decision could be in favour of a **sliding gate**. This is known ground to consultants. There are widely spread, and therefore they must be the best choice. Stop, has the consultant truly checked the requirements? They know that **roller gates** have better coefficients of friction and can work with lower driving power. However, sliding gates are easier to build at lower cost requiring considerably fewer maintenance interventions. However double sluice gates and a shutter weir are also included into the consideration of feasible variants. Other options are not considered for functional reasons.

2.1.2.2 Tender specification

Once these basic issues have been clarified – on the basis of the product requirements – consultants compose the **tender specification** which will be acknowledged by the customer and sent or requested by potential bidders. According to the experience of the author, a tender offer – **construction specification** and **service specification** – can include a few lines or even many pages. The tone of the document can be casual or technical. The objective is to allow bidders or potential contractors to provide a detailed and realistic service specification while indicating best feasible pricing. In turn, the customer must be in possession of comparable offers to allow selection of the best economical solution.

Besides the references contained within the product requirements, the consultant will add the following indications: "The sluice gate, a **dam gate with low overflow** and **headrace water regulation** shall have a length of 4 m and a height of 3 m. On the basis of **statistical initial sizing**, the sluice gate requirements are: headwater side gate leaf of 12 mm made of steel – which is the minimum steel thickness for a safety lock according to DIN 19704-2/4.2 – as well joists, cross members and side members designed as bars made of sectional steel for stabilisation purpose. A sliding gate is to be provided. The gate laterally slides along side guides embedded in the weir recesses on guide blocks and slide strips made of UHMWP (ultra-high-molecular-weight polyethylene) and sealed with rubber profiles. Besides the 10 % supplement specified by DIN 19704-1/5.1 for coatings, water, pollution and applied ice load have to be considered according to the customer's experience with 15 % applied ice load on 15 % dead load. A sealing surface heating must be considered, since the sluice gate must be operated even during slight frost periods. The ice force can be neglected since operation will be suspended during this period. A **gasket** has to be provided at the lower end position.

Lifting and lowering of sluice gates are implemented via two rising trapezoidal threaded spindles. The actuator system must be self-locking and single-sided fixing to be provided for two-side driven closing elements. The connecting elements must be calculated for buckling. Closed gearings are also to be deployed for the output stages. The spindles and rotating shafts must be protected over their complete length against outside impacts. The electro-mechanical movement is based on local operation and by a central control room via remote data transmission e. g. via GSM and Profibus. Manual emergency operation must be additionally provided.

2.1.3 Example for calculating the resulting impact on actuator sizing

According to the sequence of typical impacts on the closing element indicated in 2.1.1 and complying as far as possible with DIN 19704-1/5, the overall forces acting on the sluice gate weir required for actuator and gearbox sizing subject to tender in 2.1.2 are to be calculated: Parameters were selected following discussions with experts for civil engineering constructions for water applications and considering available systems. The calculation should be as thorough as possible. All potential impacts shall be considered and assessed. The objective is to find the optimum sizing for this particular requirement. We shall start with the "example calculation" of the dead load.

2.1.3.1 Dead load

The dead load of the gate leaf is calculated as follows:

$$F_{GL} = l_{GL} * h_{GL} * w_{GL} * \rho * g$$
$$= 4\ \mathrm{m} * 3\ \mathrm{m} * 0.012\ \mathrm{m} * 7.85\ \mathrm{kg/m^3} * 9.81\ \mathrm{m/s^2}$$
$$F_{GL} = 11\ \mathrm{kN}$$

wherein
F_{GL} = Dead load of gate leaf
l_{GL} = Length of gate leaf
h_{GL} = Height of gate leaf
w_{GL} = Width of gate leaf
ρ = Density of steel
g = Gravitational force

The mounting elements of the closing element are to be calculated and added according to the same procedure considering the respective density:

Horizontal struts:	5.0 kN
Vertical struts:	3.5 kN
Lashing eyes, bracings, welded seams:	1.5 kN
Slide strips (PE), sealing strips (rubber):	0.5 kN
2 spindles:	0.5 kN

Since the definite spindle selection could not yet been made, the weight of a comparable system is considered which might be subject to later correction. As an alternative to the spindle weight, the weight of 2 lantern gears could be considered.

Submerged gates preferably use spindles – this is mainly due to the fact that sealing a rounded roof penetration is straightforward. Due to the required length, the spindle weight to be pulled and maintained can be a multiple of the sluice gate weight.

F_{DL} = 22 kN
F_{DL} = Dead load of sluice gate

According to DIN 19704-1/5.1, a supplement of 10 % has to be calculated. Consequently, the dead load component of the total opening force amounts to:

F_{DL} = 24 kN

2.1.3.2 Hydrostatic force

The hydrostatic force is calculated assuming the highest water level of the upper gate leaf edge h = 3 m as follows:

$$\begin{aligned} F_H &= l * h * h/2 * \rho_{Wat} * g \\ &= 4\text{ m} * 3\text{ m} * 1.5\text{ m} * 1 * 10^3\text{ kg/m}^3 * 9.81\text{ m/s}^2 \\ F_H &= 180\text{ kN} \end{aligned}$$

wherein
F_H = Hydrostatic force
l = Length of sluice gate
h = Height of water
ρ_{Wat} = Density of water
g = Gravitational force

Low water levels accordingly reduce the hydrostatic force F_H. No consideration is made of the impact of up to 20 cm water level at the sluice gate by overflow in the event of extreme high water (EHW) due to the negligible frequency of occurrence. On the other hand, the calculated downstream relief is also excluded. In this case, F_H has an impact as horizontal linear distributed load of 1/3 of sluice gate height (2.1.1.2).

2.1.3.3 Friction forces from hydrostatic force

2.1.3.3.1 Friction force of support strips from hydrostatic force

For the combination: Stainless steel/polyethylene (UHMWP) μ_0/μ = 1.2 is applicable for the ratio between static friction and sliding friction coefficients according to DIN 19704-1/tab. 3. In this instance, μ_{max} = 0.2 is used for µ. For this combination, the supporting slide strips made of UHMWP also adopt the sealing function. An additional consideration of sealing friction can be omitted. For the simple case, the following applies:

$$\begin{aligned} F_{FAH} &= \mu_0/\mu * FH \\ &= 0.2 * 1.2 * 180\text{ kN} \\ F_{FAH} &= 43.2\text{ kN} \end{aligned}$$

wherein
F_{FAH} = Hydrostatic frictional application force
μ_0 = Static friction
μ = Sliding friction
F_H = Hydrostatic force

Since the consultant wants to be on the safe side and places a rubber strip protruding by 3 mm with bulb and tail profile [3] at a significantly lower friction coefficient ($\mu = 1.0$) but superior sealing properties next to the rectangular polyethylene profile, components like guide runners and seals shall be considered separately relating to the hydrostatic frictional application force (**figure 2.036**).

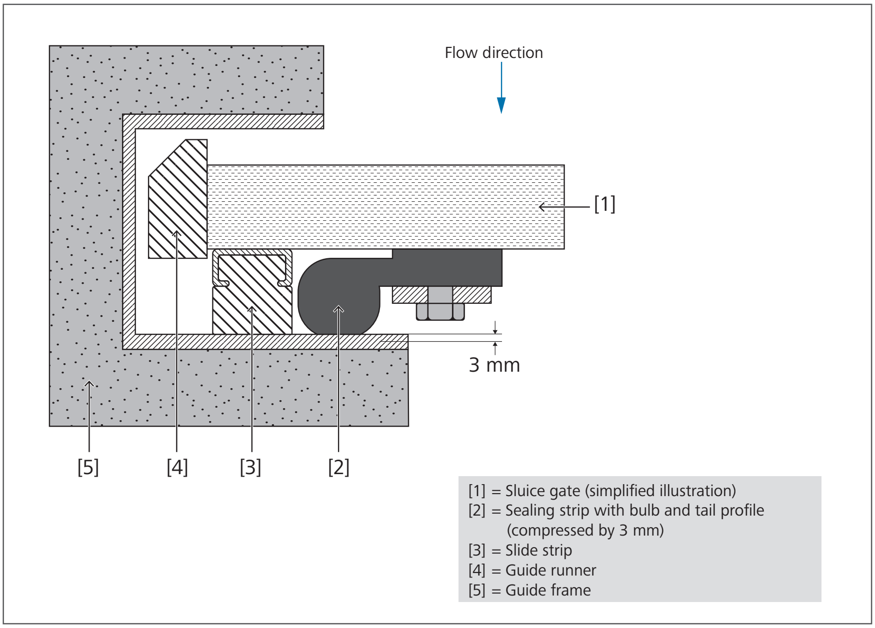


Figure 2.036: Slide, sealing and guide strips at the sliding gate (schematic diagram)

The fraction of seal friction can be determined by means of the seal preload. For example, the seal supplier will communicate the values for the different sealing materials and shapes by means of a deformation force diagram, within the framework of the documentation). The hydrostatic pressure leads to seal compression to the level that the slide strips establish contact. In our example the value is 3 mm. To compress the seal by 3 mm, 950 N per metre are required according to the manufacturer's specifications.

$$\begin{aligned} F_{SP} &= F_{SP/l} * l_S \\ &= 950 \text{ N/m} * 6 \text{ m} \\ F_{SP} &= 5.7 \text{ kN} \end{aligned}$$

wherein

F_{SP} = Seal preload force

$F_{SP/l}$ = Seal preload

l_S = 2 * h = seal length in water

The fraction of hydrostatic force adopted by the seal compression is included within the calculation considering the high friction coefficient of rubber. The slide strip must no longer transmit this fraction to the construction. Safety can be increased if in practical applications this low fraction is not deducted.

$$
\begin{aligned}
F_{FAH} &= F_{FAH/slide\ strip} + F_{FAH/seal} \\
&= \mu * \mu_0/\mu * (F_H - F_{SP}) + \mu * F_{SP} \\
&= 0.2 * 1.2 * (180\ kN - 5.7\ kN) + 1.0 * 5.7\ kN \\
F_{FAH} &= 48\ kN
\end{aligned}
$$

wherein
F_{FAH} = Hydrostatic frictional application force
μ_0 = Static friction
μ = Sliding friction
F_H = Hydrostatic force

2.1.3.3.2 Friction force of guide runners from hydrostatic force

Friction force of lateral guide runners were defined to 5 % of F_{FAH} with the consent of the customer in accordance with DIN 19704-1/10.18.

$$
\begin{aligned}
F_{GRH} &= 0.05 * F_{FAH} \\
&= 0.05 * 48\ kN \\
F_{GRH} &= 2.4\ kN
\end{aligned}
$$

wherein
F_{GRH} = Hydrostatic guide runner friction
F_{FAH} = Hydrostatic frictional application force

2.1.3.4 Buoyant force

2.1.3.4.1 Hydrostatic buoyant force

The hydrostatic buoyant force amounts to (refer to figure 2.020 “Buoyant force at sluice gate“ a)):

$$
\begin{aligned}
F_{BH} &= l * w_{GL} * h * \rho_{Wat} * g \\
&= 4\ m * 0.012\ m * 3\ m * 1 * 10^3\ kg/m^3 * 9.81\ m/s^2 \\
F_{BH} &= 1.\ 4\ kN
\end{aligned}
$$

wherein
F_{BH} = Hydrostatic buoyant force
l = Length of sluice gate
h = Height of water
w_{GL} = Width of gate leaf, plate thickness
ρ_{Wat} = Density of water
g = Gravitational force

2.1.3.4.2 Buoyancy by water displacement

The buoyancy by water displacement F_{BWD} results from the volume of the submerged gate leaf and the bar. Assuming that the volumes are relatively identical, the result is as follows:

$$\begin{aligned} F_{BWD} &= 2 * l * w_{LG} * h * \rho_{Wat} * g \\ &= 2 * 4\ m * 0.012\ m * 3\ m * 1 * 10^3\ kg/m^3 * 9.81\ m/s^2 \\ F_{BWD} &= 2.8\ kN \end{aligned}$$

wherein
F_{BWD} = Buoyancy by water displacement
l = Length of sluice gate
h = Height of water
w_{GL} = Width of gate leaf
ρ_{WAT} = Density of water
g = Gravitational force

Altogether, the buoyant force is relatively low which is the objective of constructions in civil engineering for water applications to ensure perfect closing. The buoyancy by water displacement is often balanced with the water load or even neglected and merely considered to favour the lifting procedure. We assign both buoyant force types to the positions of the closing elements in question.

2.1.3.5 Water load to be lifted

The water load is to be calculated for the lower U profile which is open toward the top. The following applies:

Height of bar $h_{Bar} = 0.08$ m and
Width of bar $w_{Bar} = 0.20$ m

$$\begin{aligned} F_{WL} &= L * h_{Bar} * w_{Bar} * \rho_{Wat} * g \\ &= 4\ m * 0.08\ m * 0.2\ m * 1 * 10^3\ kg/m^3 * 9.81\ m/s^2 \\ F_{WL} &= 0.6\ kN \end{aligned}$$

wherein
F_{WL} = Water load
l = Length of sluice gate
h_{Bar} = Height of bar
w_{Bar} = Width of bar
ρ_{WAT} = Density of water
g = Gravitational force

Since the water load of 0.6 kN is significantly below the supplement of ten percent to the dead load according to DIN 19704-1/5.1, it can generally be neglected.

2.1.3.6 Applied ice load

The experience and assessment of the customer shows, that ice plating can amount to 15 % of the dead load – irrespective of the 10 % supplement according to DIN 19704-1/5.1.

$$F_{IL} = F_{DL} * 0.15$$
$$= 24 \text{ kN} * 0.15$$
$$F_{IL} = 3.6 \text{ kN}$$

wherein
F_{IL} = Applied ice load
F_{DL} = Dead load

Otherwise, calculation can be made according to the variables determined in 2.1.1.6.

$$F_{IL} = l_{Ice} * h_{Ice} * w_{Ice} * \rho_{Ice} * g$$

wherein
F_{IL} = Ice load
l_{Ice} = Length of ice
h_{Ice} = Height of ice
w_{Ice} = Width of ice
ρ_{Ice} = Density of ice
g = Gravitational force

2.1.3.7 Ice force

According to DIN 19704-1/5.2.5, ice force can be considered as horizontally acting surface load in inland areas with p_{Ice} = 150 kN/m². The minimum ice thickness is assumed with h_{Ice} = 0.3 m. Ice force calculation is as follows:

$$F_{Ice} = l_{Ice} * h_{Ice} * p_{Ice}$$
$$= 4 \text{ m} * 0.3 \text{ m} * 150 \text{ kN/m}^2$$
$$F_{Ice} = 180 \text{ kN}$$

wherein
F_{Ice} = Ice force
l_{Ice} = Length of ice
h_{Ice} = Height of ice
p_{Ice} = Ice pressure

2.1.3.7.1 Friction force of support strips due to ice force

$$
\begin{aligned}
F_{FAIce} &= F_{FAIce/slide\ strip} + F_{FAIce/seal} \\
&= \mu * \mu_0/\mu * (F_{Ice} - F_{SP}) + \mu * F_{SP} \\
&= 0.2 * 1.2 * (180\ kN - 5.7\ kN) + 1.0 * 5.7\ kN \\
F_{FAIce} &= 48\ kN
\end{aligned}
$$

wherein

F_{FAIce} = Frictional application force due to ice force
μ_0 = Static friction
μ = Sliding friction
F_{Ice} = Ice force
F_{SP} = Seal preload force

2.1.3.7.2 Friction force of guide runners due to ice force

$$
\begin{aligned}
F_{GRIce} &= x * F_{FAIce} \\
&= 0.05 * 48\ kN \\
F_{GRIce} &= 2.4\ kN
\end{aligned}
$$

Figure 2.037: Out of order!

wherein
F_{GRice} = Guide runner friction due to ice force
F_{FAIce} = Frictional application force

In our example, the ice force is not important since the closing element is operated during frosty periods with applied ice load but not when the water surface is frozen solid (**figure 2.037**). Stability of frame and closing elements is not included within our range of consideration.

2.1.3.8 Pull

When assuming a medium flow rate for watercourses, an underpressure of 300 mbar = 30 kN/m^2 can be assumed to calculate the pull for 3 m sluice gate or water height immediately after sluice gate opening. The value is easy to understand since a vacuum is created at 1,000 mbar. Respective coefficients can be introduced for watercourses with different flow rates. This successfully applied and wide spread solution was selected, although not mentioned in the DIN standard, since practical calculation possibilities do not exist. The complete bar width leading to the highest result was implemented due to the uncertainties as mentioned in 2.1.1.8.

$$
\begin{aligned}
F_{Pull} &\approx l * B_{Bar} * p_{Pull} \\
&\approx l * w_{Bar} * h * \rho_{Wat} * g \\
&\approx 4\ \text{m} * 0.2\ \text{m} * 3\ \text{m} * 1 * 10^3\ \text{kg/m}^3 * 9.81\ \text{m/s}^2 \\
F_{Pull} &\approx 24\ \text{kN}
\end{aligned}
$$

wherein
F_{Pull} = Pull force
l = Length of sluice gate
w_{Bar} = Width of bar
h = Height of water
p_{Pull} = Pull (negative pressure)
ρ_{Wat} = Density of water
g = Gravitational force

Pull is not available when the gates are closed.

2.1.3.9 Wind force

According to DIN 19704-1/5.2.12, wind load or force calculation for wind impact on open landscape is based on coefficient c_f = 1.3, for wind protected areas like locks on coefficient c_f = 0.5. The customer must specify the dynamic pressure q in accordance with the applicable wind zone (DIN 1055-4). 0.5 kN/m^2 are specified for our application.

The area of projection A = 4 m * 3 m = 12 m^2.

$$F_W = c_f * q * A$$
$$= 1.3 * 0.5\ kN/m^2 * 12\ m^2$$
$$F_W = 8\ kN$$

wherein

F_W = Wind force
c_f = Aerodynamic force coefficient
q = Dynamic pressure
A = Area of projection

2.1.3.9.1 Friction force of support strips due to wind power

The frictional application due to wind is calculated in accordance with the frictional application force out of the hydrostatic force:

$$F_{FAW} = F_{FAW/slide\ strip} + F_{FAW/seal}$$
$$= \mu * \mu_0/\mu * (F_W - F_{SP}) + \mu * F_{SP}$$
$$= 0.2 * 1.2 * (8\ kN - 5.7\ kN) + 1.0 * 5.7\ kN$$
$$F_{FAW} = 6.\ 2\ kN$$

wherein

F_{FAW} = Frictional application force due to wind
μ_0 = Static friction
μ = Sliding friction
F_W = Wind force
F_{SP} = Seal preload force

2.1.3.9.2 Friction force at guide runners due to wind power

The guide runner friction due to wind meaning 5 % out of F_{FAW} amounts to:

$$F_{GRW} = 0.05 * F_{FAW}$$
$$= 0.05 * 6.2\ kN$$
$$F_{GRW} = 0.3\ kN$$

wherein

F_{GRW} = Guide runner friction force due to wind
F_{FAW} = Frictional application force due to wind

2.1.3.10 Contact force bed seal

According to DIN 19704-1/7.6.3, the bed seal closing pressure must amount to minimum p_{CLB} = 5 kN/m. The contact force to be applied at the bed seal amounts to:

$$F_{CB} = l * p_{CLB}$$
$$= 4\ m * 5\ kN/m$$
$$F_{CB} = 20\ kN$$

wherein

F_{CB} = Bed seal contact force
l = Sluice gate length
p_{CLB} = Bed seal closing pressure

2.1.3.11 Resulting force

In order to have a clear overview when determining the resulting impacts, it is recommended to list the individual forces in a table (**table 2.01**).

Tab. 2.01: Impacts onto the closing element

Pos.	Impacts	Force in kN at EHW 3.0 m	Direction of impact when lifting	Direction of impact when lowering
1	Dead load	22.0	↓	↓
2	Hydrostatic force			
3	Friction forces from hydrostatic force			
3.1	Friction force of support strips	48.0	↓	↑
3.2	Friction force of guide runners	2.4	↓	↑
4	Buoyant force			
4.1	Hydrostatic buoyant force	1.4	↑	↑
4.2	Buoyancy by water displacement	2.8	↑	↑
5	Water load to be lifted	0.6	↓	↓
6	Applied ice load	3.6	↓	↓
7	Ice force	n.c.		
7.1	Friction force of support strips		↓	↑
7.2	Friction force of guide runners		↓	↑
8	Pull	24.0	↓	↓
9	Wind force			
9.1	Friction force of support strips	6.2	↓	↑
9.2	Friction force of guide runners	0.3	↓	↑
10	Bed seal contact pressure	20.0		↑

n.c. = not considered

For actuator technology sizing, it is important to identify the positions of maximum and minimum load for the opening and closing directions and to achieve average values for long-term applications using diagrams. Customer and consultant agree on the evaluation as shown in **figure 2.038**.

In spite of all similarities with comparable installations, the specific applications need to be considered individually.

Opening and closing forces are calculated by summarising the respective load positions of the occurring individual forces.

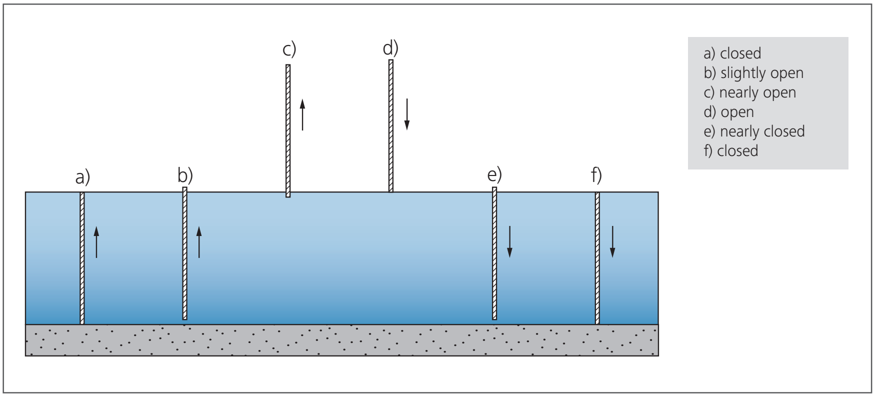


Figure 2.038: Closing element positions to be considered

Opening force

The dead load includes a supplement of 10 % (DIN 19704-1/5.1). In addition, according to the customer request, the aplied ice load of 15 % is calculated.

a) closed

$$
\begin{aligned}
F_O &= 1 + 3.1 + 3.2 - 4.1 + 6 \\
&= 24.0\text{ kN} + 48.0\text{ kN} + 2.4\text{ kN} - 1.4\text{ kN} + 3.6\text{ kN} \\
F_O &= 76.0\text{ kN}
\end{aligned}
$$

b) slightly open

$$
\begin{aligned}
F_O &= 1 + 3.1 + 3.2 - 4.2 + 6 + 8 \\
&= 24.0\text{ kN} + 48.0\text{ kN} + 2.4\text{ kN} - 2{,}8\text{ kN} + 3.6\text{ kN} + 24.0\text{ kN} \\
F_O &= 100.0\text{ kN}
\end{aligned}
$$

Contrary to expectations, the result of case b) is higher. The pull acts immediately and after opening with full power, fading shortly afterwards (**figure 2.039**). The friction is also decreasing and the buoyancy by water displacement (although low) also appears [4].

Case b) could even be rationalised and is often neglected due to the uncertainties mentioned above. Up to now, these statements could only be confirmed by model tests. But the result should be selected since on the one hand, the problem of the excessive unseating torque due to long standstill of case a) is well known and gives reason for vague supplements. On the other hand, in position b), it happens from time to time that sudden torque seating takes place.

Since the sluice gate is suspended at two points, the resulting force must be divided by two to obtain the load of the connecting element.

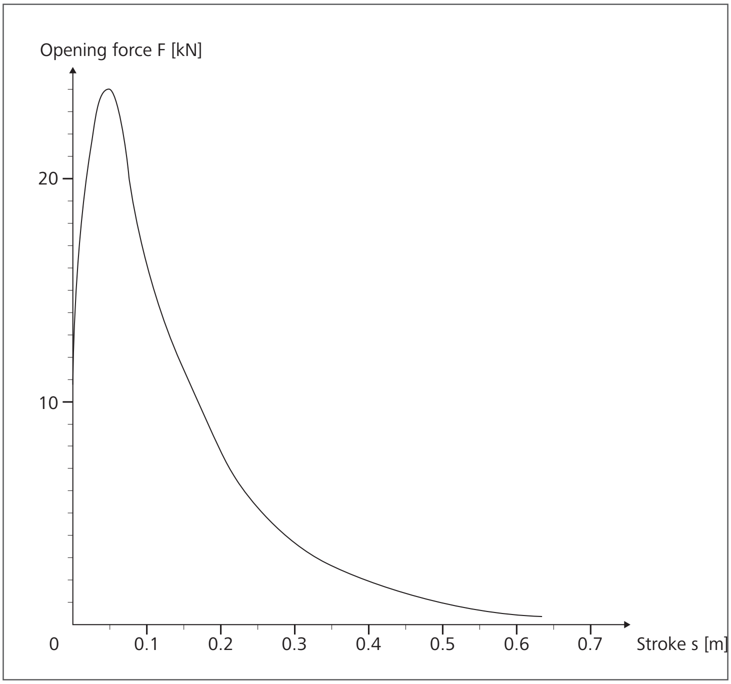


Figure 2.039: Pull impact at a sluice gate weir

$$F_{OCE} = \frac{F_O}{2} = \frac{100\ \text{kN}}{2}$$

$$F_{OCE} = 50.0\ \text{kN}$$

wherein
C_E = Connecting element
F_{OCE} = Opening force per connecting element

c) nearly open

$$F_O = 1 + 5 + 6 + 9.1 + 9.2$$
$$= 24.0\ \text{kN} + 0.6\ \text{kN} + 3.6\ \text{kN} + 6.2\ \text{kN} + 0.3\ \text{kN}$$
$$F_O = 35.0\ \text{kN}$$

Now, the essential impact is due to dead load and extra loads. The increasing friction due to wind force is relatively low compared to the friction due to water force.

Closing force

The helpful supplements for encrustation of fouling matter, water and ice become obsolete, since they are not always present.

d) open

The wind has an impact. However, the higher hydrostatic friction forces are missing.

$$\begin{aligned} F_C &= -1 + 9.1 + 9.2 \\ &= -22.0\text{ kN} + 6.2\text{ kN} + 0.3\text{ kN} \\ F_C &= -15.0\text{ kN} \end{aligned}$$

e) nearly closed

The pull is at its maximum immediately prior to closing.

$$\begin{aligned} F_C &= -1 + 3.1 + 3.2 + 4.2 - 8 \\ &= -22.0\text{ kN} + 48.0\text{ kN} + 2.4\text{ kN} + 2.8\text{ kN} - 24.0\text{ kN} \\ F_C &= +8\text{ kN} \end{aligned}$$

f) closed

The gasket has made contact. Consequently, there is no more pull. The buoyancy by water displacement is replaced again by the lower hydrostatic buoyant force. Friction due to hydrostatic force is not yet completely effective. The bed seal contact force F_{CB} = 20 kN must be considered here.

$$\begin{aligned} F_C &= -1 + 3.1 + 3.2 + 4.2 + 10 \\ &= -22.0\text{ kN} + 48.0\text{ kN} + 2.4\text{ kN} + 1.4\text{ kN} + 20\text{ kN} \\ F_C &= +50\text{ kN} \end{aligned}$$

To ensure safe closing, the actuator/gearbox combination must provide a compression force of minimum 50 kN.

As for the opening force, the following applies for the closing force per connecting element:

$$F_{CCE} = 25\text{ kN}$$

wherein
F_{CCE} = Closing force per connecting element

Hence, the closing force is significantly lower than the opening force. This is the reason why the required gearbox is specified considering the opening force. However, there are applications when the required closing force is higher than the opening force and consequently, the closing force is then used for selection.

2.1.3.12 Evaluation

The title of this section deals with the calculation of the resulting impacts on the machine design using an example. One out of many possible examples. It is quite normal that the general conditions differ like the weighing of impacts on the closing elements by customer and consultant. Utmost diligence should be applied. This means that all characteristic impacts should be considered, evaluated and combined (DIN 19704-1/tab.5). As a result, it appears that the issue is so diversified and cannot be definitely solved without model compositions as well as further investigations using computer-aided measurements and calculations. The ensuing accuracy would however not justify the expenditure and effort. These days, for many reasons, expenses should be closely monitored. Even if the objective is to win a contract. On the other hand, austerity is not always the solution. With current possibilities and the justified expenditure, it is possible to exactly calculate the requirements for installations at reasonable cost. There is no benefit if installations break down after 10 or 15 years. Narrow calculation is opposed to field experience as follows:

- Jamming will always be an issue in installations
- Temperature differences of 0 °C and less on the water side of the closing element and simultaneously 30 °C on the dry and sunny side lead to distortions like welding work.
- Two spindles never run synchronously to 100 %. This is not remedied by resigning to weld the spindle suspension in favour of screw connections through oblong holes to allow adjustments after assembly.

Therefore, the following motto must be heeded:

Calculate as accurately as required.

On the basis of this knowledge, the calculation was performed as closely as possible to DIN 19704-1/8.1 taking into account the highest load profile and thus in favour of safety.

To minimise calculation efforts, many users work with equivalent supplements based on their experience. In former times, the rule of thumb leading to success was applied:

Required driving force is the sum of dead load plus friction multiplied by two.

This rule resulted in oversized installations: high forces, high torque, large spindles, large gearboxes, large shafts, large actuators, high current consumption.

A regular pattern for driving torque determination was presented in section 2.1.1. However, the evaluation of the effective components in combination with the requirements of an installation is always required. This results generally in unique solutions and even small series are quite unlikely.

The weir selected for our calculation example might be a marginal case. In general, retention of flood water is mainly operated in the headrace section. The calculations did not take into account the low overflow for extreme high water (EHW). However, it offers the possibility to easily demonstrate and calculate all significant impacts on a closing element. If headrace or tailrace operation is required, selection of minimum one double sluice gate would be appropriate. We will go into more detail later.

Following the sluice gate weir, no further type of closing element shall be calculated since in principle, no new new findings would ensue. However, there are quantitative deviations in terms of impacts. The following applies:

- The complete connecting element load for submerged gates must always be considered. In turn, this does not apply to wind load.
- The ice force for sluice gates can generally be neglected. In turn, this does not apply to shutters.

However, the procedure shown for the following example always applies.

2.1.4 Sizing spindle actuators

The required actuator technology must be based on the forces or torques required for sluice gate operation. Calculation is prepared and implemented considering all relevant details.

2.1.4.1 Electric actuators

Actuator manufacturers offer a complete range of sizes covering small to high torques. Seen from the outside, the varying size is clearly visible. This of course due to the dimensions of required gearings, spring stacks and motors. Within a specified range, torques can be freely adjusted. Torque ranges are overlapping, making sure that end of ranges are never reached and sufficient reserves are available.

AUMA actuators comprise eleven sizes from SA 07 to SA 48, this corresponds to a torque range between 10 Nm and 32,000 Nm. For civil engineering constructions for water applications, AUMA has created selection and assignment tables.

SA 07.2 stands for:
SA = actuators in type of duty S2
07 = Flange sizes F07 according to EN ISO 5210
and simultaneously the size
2 = second generation

Besides the maximum torque, lifetime [1] is a further decisive factor for actuator selection. The basis for lifetime determination, this means for defining the possible hollow shaft or output revolutions, represents the lifetime formula for ball bearings according to ISO 281:

$$L = (C/P)^p$$

The quotient C/P is designated as **dynamic load rating**.

wherein
L = lifetime factor
C corresponds to run torque according to table
P corresponds to the calculated run torque
Exponent p > 3 for ball bearings or gearings

One of the major statements of this equation is that by reducing the run torque, the number of actuator drive shaft revolutions increase. Since a cube value is to be expected, reducing the permissible run torque by half, for example, results in the eightfold number of drive shaft revolutions. The result is the nominal lifetime.

However, an actuator is very complex and a bearing just one part. Other factors require consideration. This becomes particularly clear for larger sizes and requires experimental support. Such calculations and tests finally resulted in the assignment of run torque to maximum drive shaft revolutions.

2.1.4.2 Self-locking and self-braking

Static self-locking means that the closing element position at standstill is not changed by force effects. A gate or a sluice gate must not be lowered by the dead load. DIN 19704 does not refer to self-locking.

The advantage of single-stage worm gearings – in AUMA actuators of speeds 4 rpm to 90 rpm – is the self-locking properties of such gearings. The input shaft can be rotated either via handwheel or motor. But not the output drive side. However, worm shaft and worm wheel are always under load. Acting forces are quite high and might damage gearbox and actuator. The manufacturer declares:

"Single-stage worm gearboxes are self-locking when at standstill. Movements and vibration may cancel the self-locking properties, in particular within the following efficiency range $\eta \approx 0.45$ to 0.55."

Whereby self-locking is higher for smaller gearboxes than for larger ones. For specific applications, it is advised to contact the manufacturer to be on the safe side. In general, multi-start gearboxes are not self-locking, their efficiency equals $\eta > 0.55$. AUMA actuators are equipped with double-stage – non self-locking worm gearboxes – for the speeds 125 rpm and 180 rpm.

Since single-pitch spindles with pitches up to approx. 10° are self-locking, all other spindle thread pitch can be used in combination with non self-locking elements.

If a closing element is operated – opened or closed – it should be at standstill, once the driving force is no longer applied. This is called **self-braking** (dynamic). The braking torque of the actuator or the actuator/gearbox combination must correspond to at least the maximum output torque.

2.1.4.3 Procedure to determine the required forces and torques

For gates like weirs and lock gates, the same electric actuator technology is used as for valves like globe valves and gate valves. For this reason, we should first of all clarify that the mission of actuators designed for the valve industry is to open and close closing devices. However, in the example of gate valves, they must only provide maximum torques of up to T_{max} for "opening" and "closing" in end positions (EN 15714-2 Electric actuators for industrial valves).

Significantly lower torques are provided for travel by limit seating into the end position "open" and back. For gate valves, they amount to 20 % to 30 % of T_{max}. The actuators can settle during this phase. A common designation for the torque curve from opening at $\approx T_{max}$ via movement at no load torque until end position "open" and back to closing at $\approx T_{max}$ is the **bathtub graph**.

Weirs show different torque curves. The complete stroke of a weir is called clearing, when operating at a torque corresponding to the applied water pressure. Ideally, the sluice gate weir curve for opening and closing operations result in a **V-graph**. The applied torque decreases when lifting and increases when lowering.

One calculation option for the curve is as follows:

The customer will specify frequent water level movements within the product requirements (DIN 4049, part 3) (**figure 2.040**). Specific terminology applies for weirs and retention installations. Heights correspond to heights above sea level or for reservoirs the indications

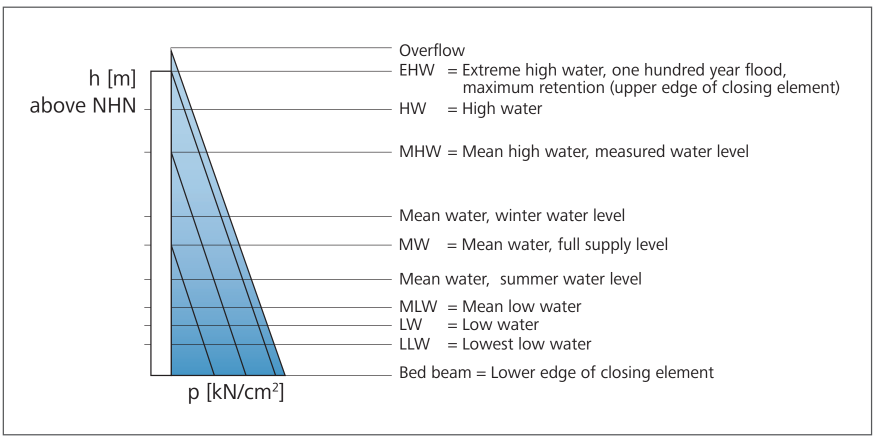


Figure 2.040: Hydrostatic pressure triangles for important water levels

within the network. Tenders specify for example the highest known flood (EHW – extreme high water) with 3.00 m plus 0.20 m overflow. In absolute figures, referring to the sea level, this amounts to 50 m bad height, 53.20 m normal height null (NHN). At times, the flow rate is indicated like the mean flow rate MQ [m^3/s] or the flow rate for one-hundred-year floods HQ100 [m^3/s].

The hydrostatic force can be determed in steps of 5 cm or 10 cm for example:

$$F_H = l * h * h/2 * \rho_{Wat} * g$$

wherein
F_H = Hydrostatic force
l = Length
h = Height of water
ρ_{Wat} = Density of water
g = Gravitational force 9.81 m/s^2
p = Hydrostatic pressure

Together with the continuous impacts (2.1.2), the calculated changing impacts can be integrated into a table (**table 2.02**). Since the tender specifies a sliding gate as closing element, static friction and sliding friction factors are integrated within the table. The table comprises a force reserve, since the ratio of static friction to sliding friction is always present $- \mu_0/\mu = 1.2$ – although operation is mostly not a partial but a complete travel.

Tab. 2.02: Sizing results for lifting and lowering, closing element positions 2.1.3.11

Closing element position	Water level at sluice gate	Dead load F_{DL}	Hydrostatic force F_H	Sliding friction μ	Static/sliding friction μ_0/μ	Hydrostatic frictional force, ice, wind – application F_{FA}	Hydrostatic frictional force, ice, wind – guide runner F_{GR}	Buoyancy F_B	Water load F_{WL}	Applied ice load F_{IL}	Pull F_{Pull}	Contact pressure bed seal F_{CB}	Opening force F_O	closing force F_C	Opening torque T_O	Closing torque T_C
	m	kN	kN			kN	kN	kN	kN	kN	kN	kN	kN	kN	Nm	Nm
a	3.00															
b	2.95															
...																
d	0.00															

The following resulting forces result from the respective position of closing element:

$$F_O = F_C = \pm F_{DL} \pm F_{FAH} \pm F_{GRH} \pm F_{BH} \pm F_{BWD} \pm F_{WL} \pm F_{IL} \pm F_{FAIce} \pm F_{GRice} \pm F_{Pull} \pm F_{FAW} \pm F_{GRW} \pm F_{CP}$$

This results in the force characteristic for a sluice gate weir, which might look like in **figure 2.041**.

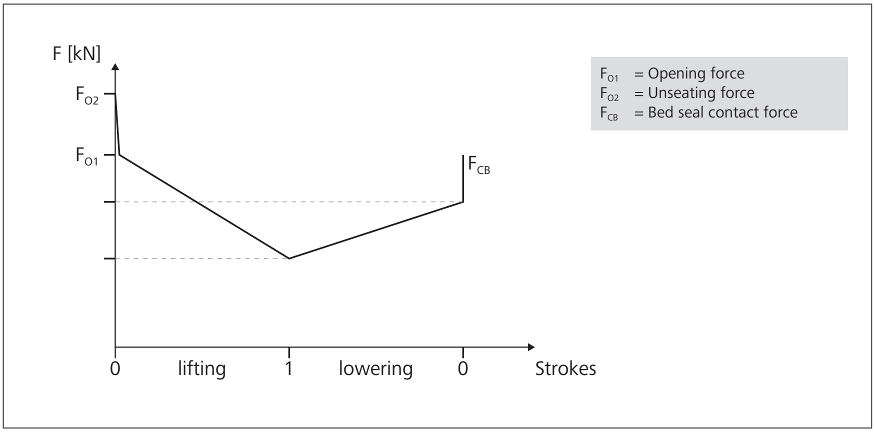


Figure 2.041: Typical force cycle at sluice gate across one load cycle "lifting – lowering" for EHW, schematic diagram

F_{O1} is the required opening force for opening a sluice gate from 0 m and at maximum water level. Depending on the period of standstill at ambient conditions (**figure 2.042**), the required opening force can increase from F_{O1} to F_{O2}. For the range exceeding the

Figure 2.042: Strongly polluted and sluggish mitre gates

initially required force, it is also called **unseating force**. This also applies for other closing element positions. At the end of each lowering process, the bed seal contact force applies.

Respective factors can be included within table 2.02 to achieve a torque curve identical to the force curve. The mean value can be determined for the V-shape part of this graph using for example the **surface integral** which will always be decisive for actuator technology selection.

Typically, spindles for civil engineering constructions for water applications are designed as trapezoidal threaded spindles (DIN 103) (**figure 2.043**).

α = rising pitch angle ≈ 3° to 5.5° for single-pitch threads [1]
ρ = friction angle Rho ≈ 6° to 12° [1]

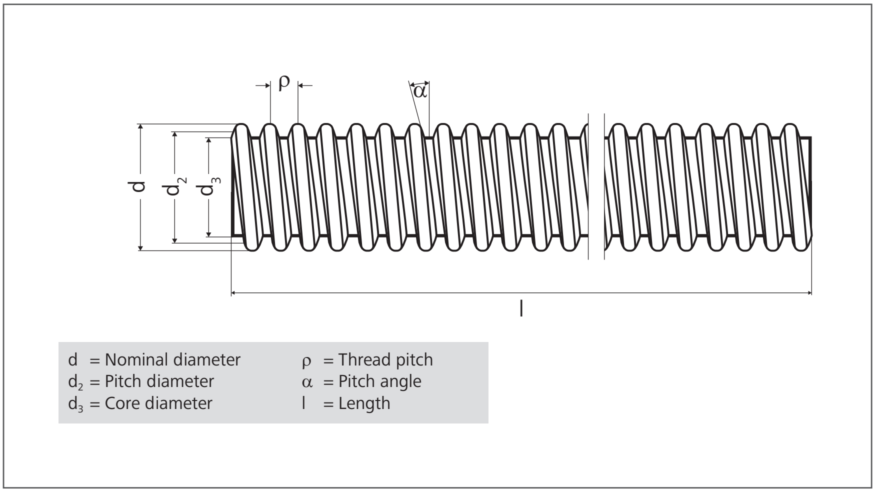


Figure 2.043: Parameters of a trapezoidal threaded spindle

Friction angle ρ is always positive. Thread pitch angle α however is always preceded by different algebraic signs, depending on the direction.

The following formulas allow calculation of opening and closing torques [1]:

Opening torque $T_O = F_O * \tan(\alpha + \rho)\, d_2/2$
Closing torque $T_C = F_C * \tan(-\alpha + \rho)\, d_2/2$

wherein
F_O = Opening force
F_{O1} = Opening force when new
F_{O2} = Opening force after ageing, unseating force
F_C = Closing force

Self-locking available if $\alpha < \rho$.

2.1.4.3.1 Starting torque

T is the total torque required to operate a closing element. T_S corresponds to the starting torque obtained by division through the gear factor. T_{S1} is the starting torque of an actuator when the installation is new. The **unseating torque** T_{S2} which is increasing with ageing is calculated on the experience of consultants with an excessive static friction coefficient μ_0 according to (DIN 19704-1/tab.3). Since the starting torque depends on factors like ageing and environmental conditions can be at different levels, the following applies (DIN 19704-1/8.4): *The set torque should be at least 25 % above the required actuator torque, according to 8.3*. This means: The torque switch is set to $T_S \geq T_{S1+25\,\%}$! This is valid for lifting and lowering, irrespective of the lower run torques considered hereafter.

In this respect, the following has to be heeded: When dealing with electric motors, controlled by frequency converters, the nominal motor torque – for considering highest possible force transmission – must be selected at least 5 % higher than the required torque (DIN 19704-1/8.3).

2.1.4.3.2 Mean torque at extreme high water (EHW)

For sizing the actuator/gearbox technology, another factor besides the starting torque is the maximum gate torque curve $T_{\text{sluice gate-EHW}}$ across the cycle to be considered according to the tender.

> ***A cycle comprises the number of complete lifting and lowering operations without significant pauses.***

By means of the surface integral, the mean torque – **arithmetic mean** – $T_{ØEHW}$ is to be determined (**figure 2.044**). Among others this is a requirement for **fitness for purpose**. The starting or unseating torque applied for a very short period and for just a few centimetres as well as the head of water above the sluice gate which might amount to up to 20 cm must not be considered for calculating the following types of duty and lifetime.

> ***The mean torque $T_{ØEHW}$ for extreme high water must not exceed the torque with which the actuator and gearbox (according to EN 60034-1) can reach the run time depending on their type of duty – for example 15 min for S2 - 15 min or 30 min for S2 - 30 min. Otherwise, there would be little reserve for the combination resulting in fast overheating, thermal actuator tripping and rapid gearbox wear.***

Simultaneously, the actuators and gearboxes working at the required torque are prepared for a higher maximum setting torque, overcoming high starting peaks.

Depending on the requirements of a curve across a stroke, $T_{ØEHW}$ can take two, three and consecutive more strokes using the surface integral (**figure 2.045**).

When lifting and lowering are considered within one cycle, this favours a more realistic averaging of the mostly higher variables for lifting and lower variables for lowering. If all

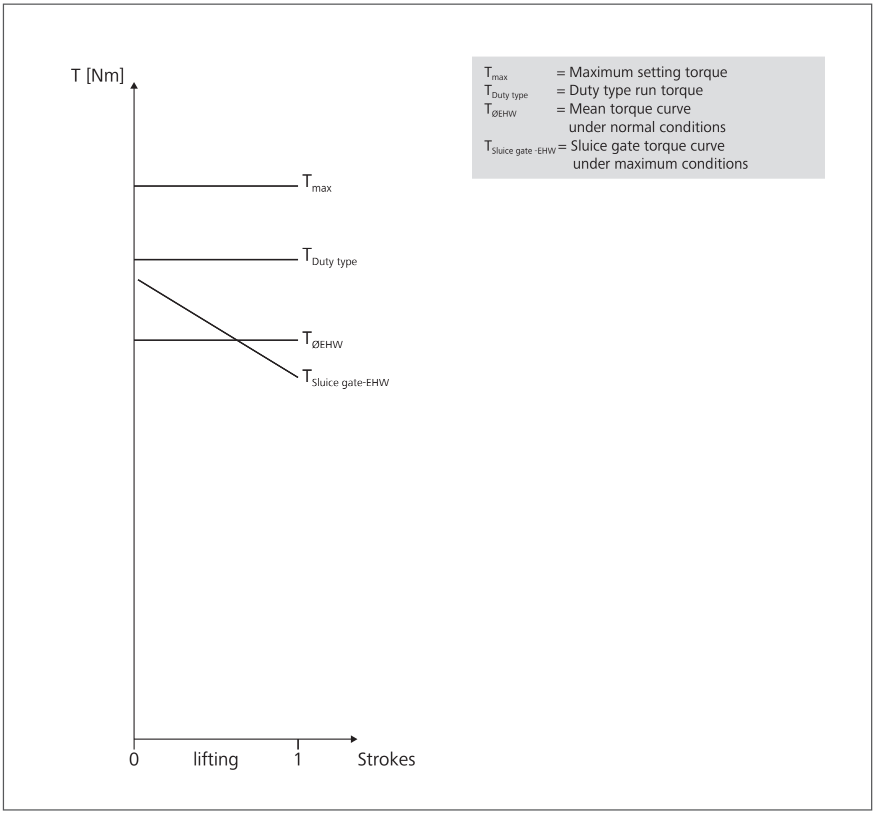


Figure 2.044: Torque characteristic for EHW, lifting, schematic diagram

strokes are assumed as lifting processes, a safety supplement is automatically to be added. Often, considerations are made to select larger actuators to avoid subsequent and expensive retrofitting.

However, in the end, one must not forget to add the many well-intentioned safety factors to evaluate the result. Since at times, excessive closing forces might lead to spindle buckling (refer to 2.1.8 Buckling safety).

The mean torque at extreme high water must be considered for selecting the type of duty.

2.1.4.3.3 Mean torque for mean water level

Actually, most sluice gate weirs are not even once pulled across the total stroke at maximum hydrostatic pressure. It happens more frequently that the closing device is operated to one or the other direction in small steps at low pressure like for MHW or MW.

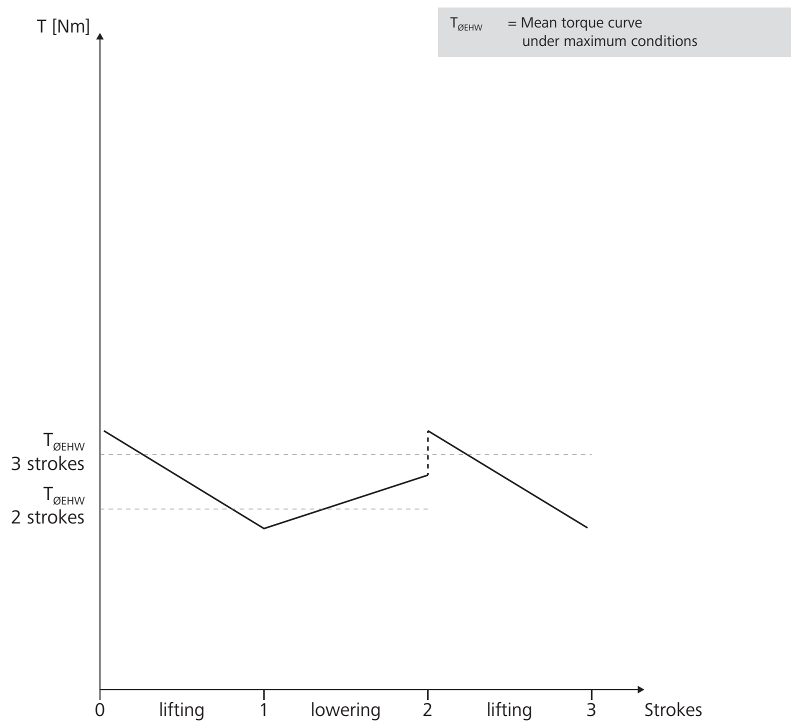


Figure 2.045: Determination of mean torque characteristics across 2 and 3 strokes at EHW, schematic diagram

The sum of operations at different water levels finally decides on the total running time of mechanics. Now, not only the arithmetic calculated force used to calculate EHW – it allows to make a statement for implementing the desired type of duty – is required but the **equivalent force** [12].

If for constant speed and identical load direction, the load continuously changes at a certain interval between a lowest value $F_{min.}$ and a highest value F_{max}, the equivalent force results from [1]:

$$F_E = \frac{F_{min} + 2\,F_{max}}{3}$$

wherein
F_E = Equivalent force

The equivalent force is the force causing the same impact as the spectrum of occurring forces. The impact of higher forces on the lifetime is disproportionately high and is therefore taken into higher consideration.

Forces or torques can be reduced to a common denominator using the **Palmgren Miner rule** [8].

$$T_{\varnothing} = \sum_{i=1}^{k} \frac{n_1 * T_i}{N_i}$$

$$T_{\varnothing} = \frac{n_1 * T_1}{N_1} + \frac{n_2 * T_2}{N_2} + \ldots + \frac{n_k * T_k}{N_k}$$

wherein
$T_{\varnothing}$ = Mean torque
n_i = Partial cycles
N_i = Total cycles

The actuator opening torque T_{AO} is often relatively high, but only effective during a few seconds and can consequently be omitted for lifetime calculation. This also applies to the torque at extreme high water $T_{A\text{-}\varnothing EHW}$. According to applicable design specifications, it is only applicable once in a hundred years. Typically, the most influential loads occur due to the measured water level $T_{A\text{-}\varnothing MHW}$ and the full supply level $T_{A\text{-}\varnothing MW}$ (**figure 2.046**).

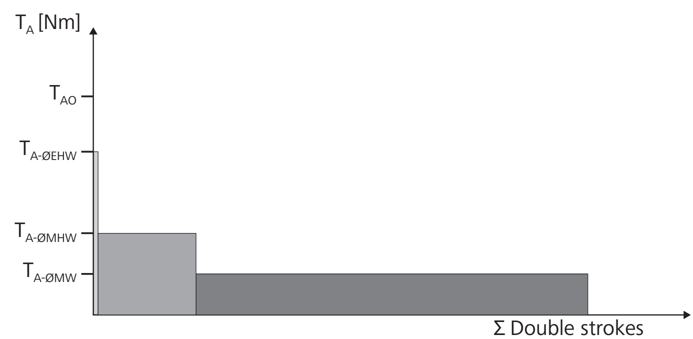


Figure 2.046: Combining the proportionate stress cycles

For easier calculation, shorter stress cycles stipulated in the product requirements of same water level type are combined as one complete lifting and lowering process linked with an optional pause time to result in a double stroke.

The objective is to determine a mean torque – for example: $T_{\varnothing MHW/MW}$ or $T_{A\text{-}\varnothing MHW/MW}$ – and the number of occurring actuator and gearbox rotations for a time of 35 years (DIN 19704-1/9.5.3). The results are the foundation for the **proof of operational stability**. In this instance, $T_{lifetime}$ always requested from the actuator/gearbox technology must be at least equal to $T_{\varnothing MHW/MW}$ (**figure 2.047**).

The mean torque at mean water level must be considered for the lifetime calculation.

2.1.4.4 Applications for hydraulic steel structures (HSS)

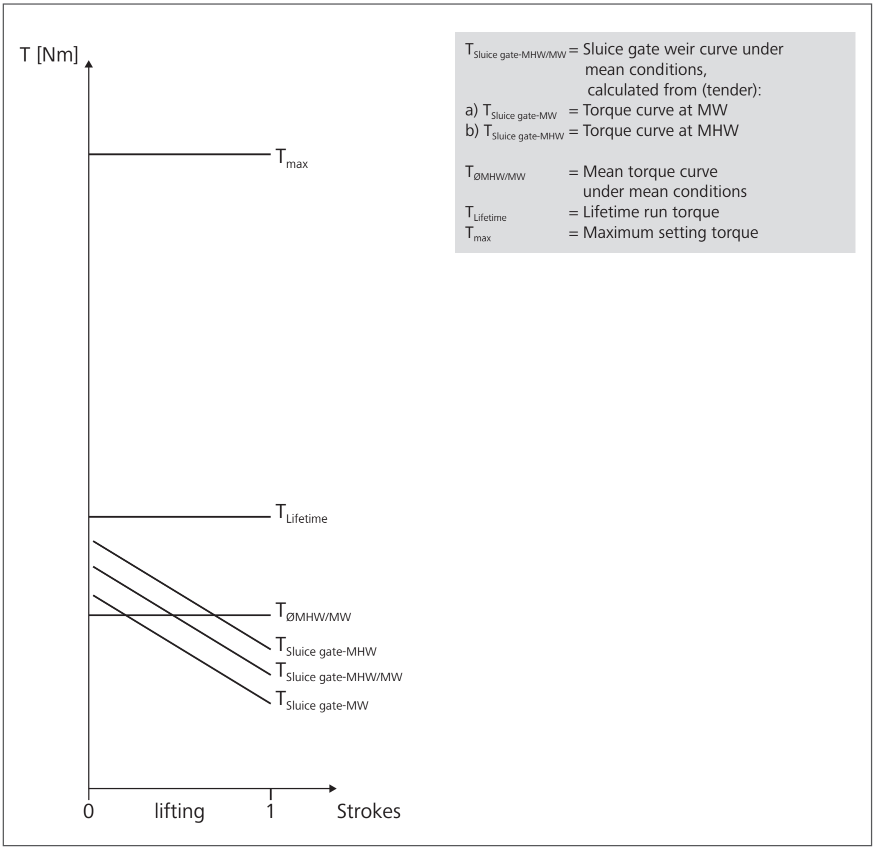


Figure 2.047: Torque characteristic for mean load, lifting, schematic diagram

Civil engineering constructions for water applications allow a multitude of applications requiring different torques, running times and pause times. Consequently, AUMA has created a classification. They are designed as applications for hydraulic steel structures of classes HSS 1, HSS 2 and HSS 3 and classify different loads (**table 2.03**). Hereby the duration of using machine parts and their electrical equipment with the exception of wear parts is generally 35 years within applications for hydraulic steel structures (DIN 19704-1/9.5.3.1).

Tab. 2.03: Selection criteria for actuators and gearboxes for civil engineering constructions for water applications – descriptive assignment

Application	HSS 1	HSS 2	HSS 3
Type of duty	- S2 - 15 min/30 min/60 min -		
Lifetime	35 years	35 years	35 years
Load	low	medium	high
Application profile: Locks	Touristic use during sommer months	Commercial use Ice-free months/12 h/d	Commercial yearly use 24 h/d
Application profile: Weirs	Retention of floodwater	Roughly maintaining the level	Retention of sluice gates

Of course, there are also peak times at locks for pleasure boats (**figure 2.048**), this means with highest lock density but relatively low usage during the nights and the winter months. The total operating time of 35 years is consequently relatively low.

Figure 2.048: Sluice gate for pleasure use

The classification described is for rough orientation only. Therefore, the number of possible output drive revolutions at lifetime no-load torque was assigned to the mentioned criteria (**table 2.04**). Calculations and experimental examinations were the prerequisites. The movements or the voltage cycles specified in the product requirements are basis for determining the required total revolutions of the actuator output shaft at mean torque for a duration of 35 years. This allows to determine the HSS application and the corresponding permissible run torque.

Tab.2.04: Selection criteria for AUMA actuators in civil engineering constructions for water applications – figure assignment

Number of possible output drive shaft revolutions of actuator			
Application	HSS 1	HSS 2	HSS 3
Type of duty	- S2 - 15 min/30 min/60 min -		
SA 07.2 – SA 14.2	1.2m	5.0m	10.0m
SA 14.6 – SA 16.2	0.9m	4.0m	8.0m
SA 25.2 – SA 30.2	0.6m	2.5m	5.0m
SA 35.2 – SA 40.2	0.3m	1.0m	2.0m

To meet the frequently high requirements for hydraulic steel structures,the manufacturer AUMA uses reinforced ball bearings, polished drive worms and shot-peened drive shafts when stating the **HSS** abbreviation.

HSS allows the actuators to achieve a longer and disturbance-free lifetime by at least 10 %.

With regard to long lifetimes upon actuator selection by consultants and builders, this special version must also be provided as well as an increased corrosion protection under certain circumstances. When neglecting HSS version and HSS run torque, this might require worm and worm wheel replacement after only a few years.

2.1.4.5 Run torques

What is the reason why an operator cannot operate the actuator and gearbox within the selected torque range as long and as often as thinking best?

2.1.4.5.1 Duty type run torque

Motors deployed in actuators cannot be permanently operated. The reason is on the one hand that customers request so-called pot-type motors, renown to be dust and watertight. They even allow for **enclosure protection IP67** and **IP68** for the overall actuator. IP (= Ingress Protection) characterises the protection against ingress of dust and water. The **degree of protection** describes the two figures 67 respectively 68.

The drawbacks of pot-type motors are that they are only capable of self-cooling or surface cooling and consequently need cooling-down intervals. If this is not respected, the typically integrated temperature monitor switches of the motor when reaching winding temperatures of 140 °C. Continuously running motors have obligatory cooling, working with a fan to prevent overheating. However, in standard version, the high degree of protection cannot be achieved.

Due to the possible running time, the useability of an electrical machine is characterised by the **type of duty**. The abbreviations **S2 - 15 min** or **S2 - 30 min** – on request, the type of duty **S2 - 60 min** can be agreed with AUMA – characterise an actuator capable of running during 15 min, 30 min or 60 min without interruption at the duty type run torque (**table 2.05**). Prior to operating the motor again, a cooling phase must be respected, ideally until reaching the defined initial temperature.

Tab. 2.05: Technical data Multi-turn actuators – Duty type – run torques, at 40 °C (extract)

Type	Torque ranges [Nm]		Speed ranges [rpm]	Duty types run torques [Nm]	
	in duty type			In duty type	
	S2 - 15 min	S2 - 30 min		S2 - 15 min	S2 - 30 min
SA 14.2	100 – 250	100 – 180	4 – 180	85	65
SA 14.6	200 – 500	200 – 360	4 – 180	175	125
SA 16.2	400 – 1,000	400 – 710	4 – 180	350	250
SA 25.2	630 – 2,000	630 – 1,400	4 – 180	700	500

If an actuators is operated at its duty type run torque, it reaches the running time in accordance with the selected type of duty. This running time may suffice to execute one or several strokes. The duty type run torque refers to the actuator motor. It must be at least as high as the mean torque required at extreme high water.

Longer running times could be required if the duty type run torques fall bellow the values specified in the table. For this purpose, manufacturers provide appropriate tables.

The **stall torque** – the peak point on the torque characteristics – amounts to double the maximum adjustable torque.

However, gearboxes are closely linked to the duty type configuration of actuators. During their mission, they are also subjected to thermal load. This is the case for worm gearings within the actuators as well as separate worm, bevel and spur gearboxes. This is explained in more detail in the lifetime run torque (2.1.4.5.3).

2.1.4.5.2 Number of subsequent strokes

Both thermal items, thermal balance within the motor and indirectly also within the actuator gearing are decisive for actuator assignment. Therefore, it is of utmost importance, to define the strokes to be subsequently operated as well as the resulting running time prior to undertake further calculations. If this exceeds the assigned type of duty, the actuator is subjected to thermal overload. The motor protection – themoswitch or PTC thermistor – can trip and stop the actuator for a certain time, depending on its heat store capacity.

Mention was already made that the sluice gates are only operated in steps of 10 cm including pauses of hours or even days, leading to the assumption that the sizing of the actuator can be relatively low. Experts do not agree with this assumption:

- For safety reasons, a sluice gate must be capable of travelling for **at least one complete stroke**. It must be possible to close an open weir without delay in the event of an arriving flood wave.
- Errors can always be made. Operator faults made in remote control stations where only end position feedback is provided and possibly not even allowing visual checks. In this instance, corrections must be made. Here, we are dealing with **two complete strokes**. They are required during commissioning anyway, according to operators who have to live with irritating downtimes when exceeding the type of duty ...
- Furthermore, incalculable loads can occur. Maybe a slip clutch must be readjusted. Let alone, difficult and unclear water level development. The consequence is alternating movement which might result into **three and more strokes**. If in between the operation, the actuator comes to a standstill, cooling times of one hour up to two hours might be required until operation can be resumed.

As a matter of fact, very high actuator reserves would have to be considered to implement all previous requirements. However, this can already be achieved by sufficient dimensioning of actuator technology and also by selecting high output speed and consequently higher gate operating speeds.

Affected customers will agree where protection is needed for potentially rapid flood waves. Whenever these obligations are not imperative, the following point of view could be represented: In plains, a sluice gate should be opened slowly, maybe at a velocity of v = 150 mm/min, to ensure absorbing capacity of the subsequent water section including the groundwater reservoir. In mountains where higher precipitation quantities are to be expected, the operation velocity should be defined to at least v = 300 mm/min for opening and closing.

Sizing demands can widely differ. The DIN standard does not stipulate any of these definitions. The decision is incumbent to the customer and consultant and their demands are included within the tender.

2.1.4.5.3 Lifetime run torque

Another physical challenge is the actuator gearing. Single-stage worm gearings are integrated in AUMA actuators for speeds of 4 rpm to 90 rpm and double-stage worm gearings for 125 rpm and 180 rpm, whereby the single-stage gearing is of particular importance. Often it is needed for self-locking. The relatively low efficiency of less than 45 % compared to minimum 55 % for double-stage units leads to heat development by friction, resulting in material stress and problems in heat dissipation. An appropriate actuator sizing is the solution. This fact determines the lifetime run torque (**table 2.06**) which should not be significantly exceeded in average when considering the desired service life.

If an actuator is operated in agreement with the lifetime run torque, it will achieve the number of drive shaft revolutions according to the application (refer to table 2.04). The lifetime run torque refers to the actuator gearing. It must be at least of the size as the mean torque at mean water levels.

Tab. 2.06: Technical data Multi-turn actuators – Lifetime run torques (extract)

Type	Speed ranges [rpm]	Lifetime run torques [Nm]		
	4 – 180	HSS1	HSS2	HSS3
SA 14.2	4 – 180	100	75	50
SA 14.6	4 – 180	175	135	100
SA 16.2	4 – 180	350	270	200
SA 25.2	4 – 90	700	550	400

Interpolation, in this particular case rising the potential output speed by falling below the permissible lifetime run torque, facilitates the graph in **figure 2.049**.

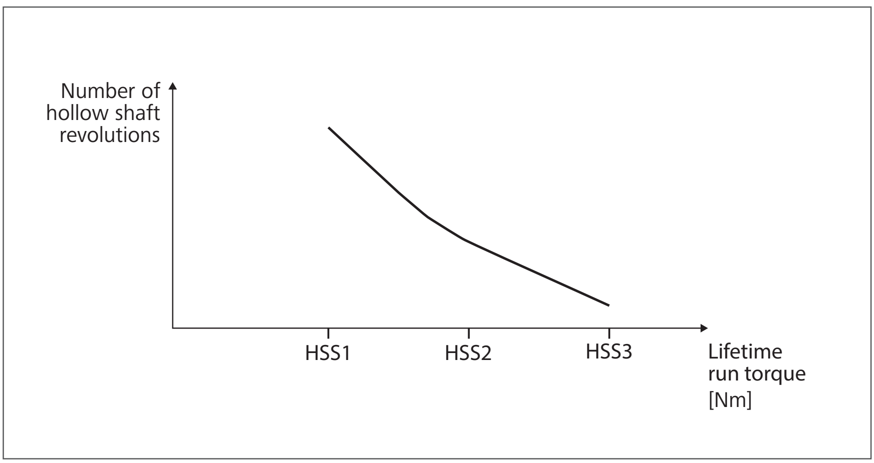


Figure 2.049: Possible drive shaft revolutions depending on Lifetime run torques

Like the actuator gearings, the separate gearboxes need special consideration. Please refer to the "Technical data Bevel gearboxes" by AUMA (**table 2.07**).

Tab. 2.07: Technical data Bevel gearboxes – Lifetime run torques (extract)

Type	Output torque perm. [Nm]	Reduction ratio i	Input torque[1] [Nm]	Factor[2]	Lifetime run torques [Nm]	Lifetime, number poss. output revol. $U_{GK\text{-}HSS}$ [million]
GK 16.2	1,000	4 : 1 5.6 : 1	278 198	f1 = 3.6 f2 = 5.0	350	0.9
GK 25.2	2,000	5.6 : 1 8 : 1	397 278	f1 = 5.0 f2 = 7.2	700	0.6
GK 30.2	4,000	8 : 1 11:1	556 404	f1 = 7.2 f2 = 9.9	1,400	0.3

1) For maximum output torque
2) Conversion factor from output torque to input torque to determine the actuator size. The following applies: $f = i * \eta$, i = reduction ratio, η = efficiency
3) Lifetime run torques for bevel gearboxes are reduced to 35 %
4) For higher lifetime requirements, the lifetime run torques are to be reduced by the lifetime formula $L = (T_{GK} / T_{GKØ})^3$ (2.1.5.6.2 Lifetime proof).

The gearbox **breaking torque** is typically double the maximum output torque. Damage or plastic deformation by maintaining large forces can however occur for lower values.

2.1.5 Sizing of spindle actuators – example calculation

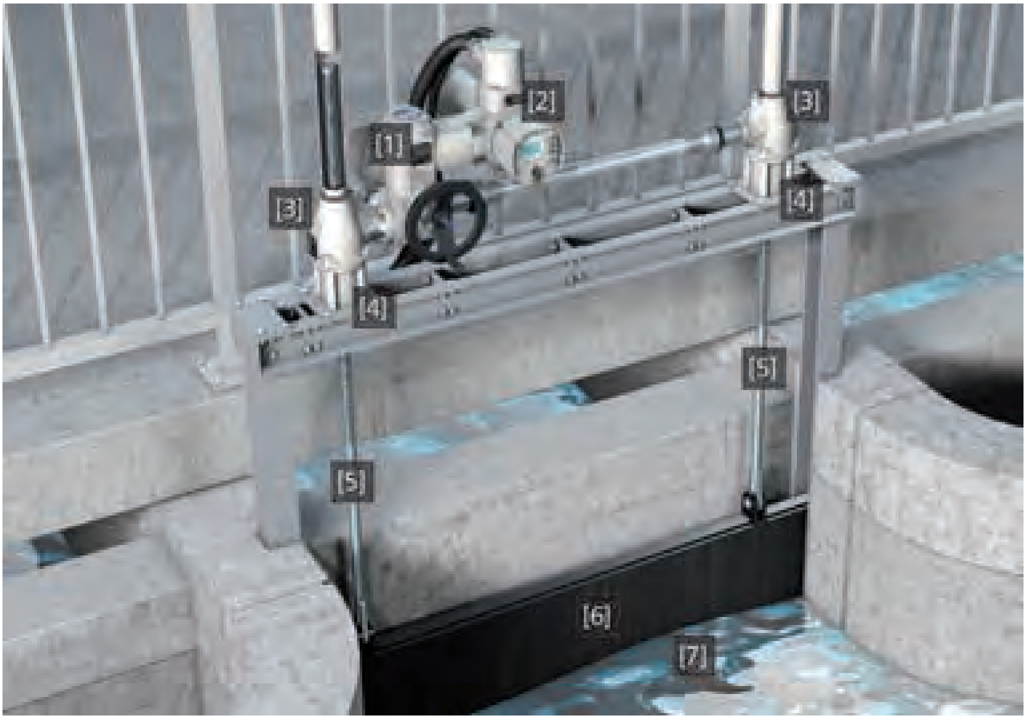
Consideration, calculation and evaluation must be made for the elements shown in (**figure 2.050**). The following was taken from the tender or the subsequent example calculation.

Given:

- Resulting forces from section 2.1.3.11 (refer to figure 2.038)
  - Position a) (lifting): 76 kN
  - Position b) (lifting): **100** kN
  - Position c) (lifting): 35 kN

  - Position d) (lowering): –15 kN
  - Position e) (lowering): 8 kN
  - Position f) (lowering): **50** kN

- Decisive water levels
- Number and length of sluice gate movement for assigned water levels
- Number of potential subsequent strokes: minimum 2 strokes
- Operating speed v = (200 … 400) mm/min



Actuator [1] with actuator controls [2], bevel gearboxes [3] and AK spindle nuts [4], connecting elements: spindles [5], closing elements: sluice gates [6], MW mean water [7]

Figure 2.050: Possible set-up of an actuator/gearbox combination

Wanted:

- For gearbox sizing: Forces and torques at gearbox outputs for unfavourable load distribution ($F_{OCE60\,\%}$ / $F_{CCE60\,\%}$, $T_{OCE60\,\%}$ / $T_{CCE60\,\%}$)
- For actuator sizing: Forces and torques at gearbox inputs or at actuator outputs (T_{AO} / T_{AC})

To be determined:

- Running times per stroke [min]
- Number of possible subsequent strokes
- Operating speed

Proof to be shown:

- Type of duty
- Lifetime
- Self-locking

To be considered:

- One-sided stop
- Exceptional impacts of actuators in case of an accident
- Buckling resistance

To be defined:
- Gearbox parameters
- Actuator parameters

2.1.5.1 Calculation of opening and closing force

Taken from the force table:

Opening: Opening force F_O = 100 kN for EHW or
Opening force per connecting element F_{OCE} = 50 kN

Closing: Closing force F_C = 50 kN at EHW or
Closing force per connecting element F_{CCE} = 25 kN

These are maximum values which should be sufficient for starting new installations at highest water level. It would however be very optimistic to believe that the same force requirement prevails for each side. Due to uncertainties like sluggishness caused by insufficient precision during assembly, temperature fluctuations, roughness caused by abrasion, but also natural ageing contribute to at least slight unfavourable load distributions for two-side operated closing elements (DIN 19704-1/8.2g). Consequently, highest asynchronous driving forces must be available for actuators with electrical or mechanical synchronous operation (DIN 19704-1/8.4). In practical applications, this is considered while the total load is not distributed at

50 % to 50 %, but at
40 % to 60 % or at a different ratio if necessary.

In our case, 40 % to 60 % means 40 kN to 60 kN.
For 60 % load $F_{OCE60\,\%}$ = 60 kN.

Both gearboxes must be selected.

Closing is processed at 40 % to 60 % = 20 kN to 30 kN:
$F_{CCE60\,\%}$ = 30 kN.

Due to the lower force for gearbox selection, this is insignificant in our example.

2.1.5.2 Determination of the output torque in opening direction

- **Stress proof**

After having determined the opening force per connecting element for unbalanced load, Tr 80 is the preliminary definition of the spindle thread. Prior to torque determination, an approximative calculation with stress proof (DIN 18 800-1/747), a multiple stress tolerance is proven:

$$\frac{\sigma_d}{\sigma_{R,d}} \leq 1$$

$$\frac{\frac{F_{Z,d}}{A}}{\frac{f_{Y,K}}{\gamma_M}} \leq 1$$

$$\frac{\frac{101\,kN}{37.4\,cm^2}}{\frac{65\,\frac{kN}{cm^2}}{1.5}} \leq 1$$

$$0.06 \leq 1$$

wherein

σ = Normal stress

σ_d = Rated value of σ

$\sigma_{R,d}$ = Limit normal stress

$F_{T,c}$ = Characteristic value of normal (tensile) force F_T
= $1.25 \cdot F_{OCE60\,\%}$

$F_{T,d}$ = Rated tensile force
= $F_{T,c} * \gamma_F = 1.25 * F_{OCE60\,\%} \cdot \gamma_F$
= $1.25 * 60\ kN * 1.35 = 101\ kN$

γ_F = Partial safety factor for impacts (loads), $\gamma_F = 1.35$

A = Core cross-section = $\frac{d_3^2 * \pi}{4}$

A_{TR80} = Core cross section TR80 $= \frac{6.9^2\ cm^2 * \pi}{4} = 37.4\ cm^2$

d_3 = Core diameter = 6.9 cm

$f_{Y,K}$ = Material yield strength 42CrMo4 = $65\ \frac{kN}{cm^2}$

γ_M = Partial safety factors for resistance variables (load capacity), for mechanical engineering $\gamma_M = 1.5$

After formula conversion

$$A \geq \frac{F_{Z,d} * \gamma_M}{f_{Y,K}}$$

The core cross-section required for lifting is easily compared to the core cross-section of a trapezoidal threaded spindle TR80. “Since this is easier to understand.”

Also the comparison:

$$A_{TR80} \geq A_{Trequired}$$

$$\geq \frac{F_{Z,d} * \gamma_M}{f_{Y,K}}$$

$$\geq \frac{101\ kN * 1.5}{65 \frac{kN}{cm^2}}$$

$$37.4\ cm^2 \geq 2.3\ cm^2$$

$A_{Trequired}$ = required core cross-section required for tensile strength

is clearly in favour of the selected spindle (refer to a.2.1.82.). Lifting is unproblematic when using the same material. If only lifting is significant, a smaller cross-section must be selected for the result. The verification of buckling safety will not be so obvious. Slenderness and buckling length of the spindle play an important role.

- Spindle pitch

The pitch P = 10 mm should be preferred according to (DIN 103). 12 mm, 14 mm and 16 mm can be used under special conditions.

A drawback lies with larger spindle pitches requiring higher torques. Smaller spindle pitches are therefore more significant for manual operation, since DIN 19704-1/8.3 specifies an upper limit value for the force to be applied.

P = 10 mm is selected. Pitch angle α is however below 2.5°. The spindle is self-locking (2.1.4.3).

You may directly start calculation using the force-torque conversion formula above (2.1.4.3). For frequent applications, it is worthwhile entering the data into a computer program.

- Opening torque for unfavourable load distribution

The following is available:

$$F_{OCE60\ \%} = 60\ kN$$

Depending on the procedure, the following spindle data is required:

n = 1
μ = 0.2 (material pairs steel/copper alloy)
d = 80 mm
d_2 = $d - P/2$ = 80 mm − 10 mm/2 = 75 mm
d_3 = 69 mm

n = Number of threads
μ = Friction coefficient
d = Nominal diameter
d_2 = Pitch diameter

Consideration must be made that dry spindles require significantly higher torques than lubricated spindles (DIN 19704-1/6 tab.4). If the calculation included a lubricated spindle, the motor might not be able to shut the closing element with a dry spindle.

To be on the safe side, a friction value of $\mu = 0.2$ was integrated into example calculation. A computer program supported the calculation of the torque required at the spindle:

$$T_{OCE60\,\%} = 550 \text{ Nm}$$

wherein
$T_{OCE60\,\%}$ = Opening torque for unbalanced load per connecting element

2.1.5.3 Gearbox selection

A bevel gearbox is to be selected for satisfying the opening torque requirement of 550 Nm per spindle. According to table 2.07, GK 25.2 with lifetime run torque of 700 Nm was selected.

2.1.5.4 Proof for actuator opening and closing torques

We had to consider a potential unbalanced load for gearbox definition. For actuators, normal loads apply: The total load amounts to 100 %. Consequently, the torque T_O = 920 Nm can be calculated using F_O = 100 kN.

To achieve a running time as short as possible, the smaller gear reduction ratio 5.6:1 should be selected, leading to a higher required input torque. The larger gear reduction ratio 8:1 with a lower input torque is in turn recommended if manual force should be kept as low as possible.

The required **actuator opening torque** results from integrating the selected gear factor with the pertaining reduction ratio (table 2.07):

$$\begin{aligned} T_{AO} &= \frac{T_O}{f1} \\ &= \frac{920 \text{ Nm}}{5.0} \\ T_{AO} &= 184 \text{ Nm} \end{aligned}$$

wherein
T_{AO} = Actuator opening torque
f_1 = Conversion factor from output torque to input torque

For actuators, a supplement of 25 % is considered here for the adjustment torque (DIN 19704-1/8.4). For actuators controlled by frequency converters, for example at lock gates and lock gate leaves, an additional supplement of 5 % has to be considered (DIN 19 704-1/8.3).

$$T_{AO+25\,\%} = 1.25 * T_{AO}$$
$$= 1.25 * 184\ \text{Nm}$$
$$T_{AO+25\,\%} = 230\ \text{Nm}$$

In our example, this means that 230 Nm should be the minimum value to be selected. For this application, the SA 14.6 is adapted even for a speed of 180 rpm in type of duty S2-30 min at T_{Amax} = 290 Nm. **Table 2.08** shows an extract of the manufacturer's technical data sheets.

All maximum permissible torques are sufficient for starting up since the torque rapidly decreases after the peak has occurred (refer to figure 2.041).

Tab. 2.08: Torques for SA 14.6 (extract)

Actuator type	Speed rpm	T_{Amax} for S2 - 15 min [Nm]	T_{Amax} for S2 - 15 min [Nm]	$T_{Duty\ type}$ for S2 - 15 min [Nm]	$T_{Duty\ type}$ for S2 - 30 min [Nm]	$T_{Lifetime}$ for HSS2 [Nm]
SA 14.6	45/125 180	500 400	360 290	175 175	125 125	135 135

The result for closing direction

$$F_C = 50\ \text{kN}$$

is converted by our computer program to:

$$T_C = 300\ \text{Nm}.$$

The actuator closing torque amounts to:

$$T_{AC} = \frac{T_C}{f1}$$
$$= \frac{300\ \text{Nm}}{5.0}$$
$$T_{AC} = 60\ \text{Nm}$$

or

$$T_{AC+25\,\%} = 1.25 * T_{AC}$$
$$= 1.25 * 60\ \text{Nm}$$
$$T_{AC+25\,\%} = 75\ \text{Nm}$$

Whereby:
F_C = Closing force
T_C = Closing torque
T_{AC} = Actuator closing torque
f_1 = Conversion factor from output torque to input torque

The actuator torque switch should be set to minimum $T_{AC+25\,\%}$ = 75 Nm. Simultaneously, the increased closing force has to be considered for verifying the buckling safety. If only an increased torque can be set for the selected actuator, the proof has to be made applying respectively high force.

2.1.5.5 Determination of the number of subsequently operable strokes and total actuator revolutions

- Determination of the number of subsequently operable strokes

According to the calculation objective, the operating speed shall be v = (200 ... 400) mm/min. Calculation is made on the basis of actuator speed n = 45 rpm. The following results occur for the selected GK 25.2 gearbox considering i = 5.6:

Running time/stroke

$$t = \frac{s * i_{GK}}{P * n}$$

$$= \frac{3{,}000\ \text{mm} * 5.6}{10\ \text{mm} * 45\ \text{min}^{-1}}$$

$$t = 38\ \text{min}$$

wherein
t = Running time/stroke
s = Travel of one stroke
i_{GK} = Reduction ratio of bevel gearbox
P = Thread pitch
n = Actuator output speed

Operating speed:

$$v = \frac{P * n}{i_{GK}}$$

$$= \frac{10\ \text{mm} * 45\ \text{min}^{-1}}{5.6}$$

$$v = 80\ \text{mm/min}$$

For further calculations, the following is needed:

Bevel gearbox revolutions per stroke:

$$R_{GK}/\text{stroke} = \frac{s}{P}$$
$$= \frac{3{,}000\text{ mm}}{10\text{ mm}}$$
$$R_{GK}/\text{stroke} = 300$$

Actuator revolutions per stroke:

$$R_A\ /\text{stroke} = R_{GK}\ /\text{stroke} * i_{GK}$$
$$= 300 * 5.6$$
$$R_A\ /\text{stroke} = 1{,}680$$

The calculated operating speed v = 80 mm/min does not comply with the tender specification.

The advantage of a higher speed is that a larger stroke can be achieved in the selected type of duty. It must be considered that no self-locking properties are available for 125 rpm and 180 rpm.

Comparing running time per stroke with the possible running time according to the type of duty, the number of subsequently operable strokes is obtained. Verification shall also be made of the actuator speed n = 90 rpm and n = 180 rpm (**table 2.09**).

Tab. 2.09: Number of possible strokes

n [rpm]	i	Run time/ stroke [min]	Strokes at S2 - 15 min	Strokes at S2 - 30 min	Operating speed [m/min]
45	5.6	38	0.4	0.8	0.08
90	5.6	19	0.8	1.6	0.16
180	5.6	9.5	1.6	3.2	0.32

In type of duty S2 - 30 min and 45 rpm speed, only 0.8 strokes can be achieved. In type of duty S2 - 30 and 180 rpm speed, the tender specification of “2 subsequently operated strokes” is significantly exceeded with 3.2 strokes. As a matter of fact, 180 rpm is not readily applied by everybody, “since excess torques could occur when approaching obstacles”. However, in our example this effect could be attenuated by considering the gearboxes and their type ranges. In addition, safety switches are provided for timely switching off and DIN 19704-1/5.5 requires in any case “to record the best possible impacts of actuator forces/ actuator torques .... onto the closing element...” for protecting the installation. Furthermore, in our example we have the possibility of selecting the higher gearbox reduction ratio 8:1.

The ensuing running time

$$t = \frac{s * i_{GK}}{P * n}$$

$$= \frac{3{,}000 \text{ mm} * 8}{10 \text{ mm} * 180 \text{ min}^{-1}}$$

$$t = 13.3 \text{ min}$$

wherein

i_{GK} = Reduction ratio of bevel gearbox
P = Thread pitch
n = Actuator output speed

leads to 2.2 strokes in type of duty S2 - 30 min.

The same number of strokes is offered alternatively when using 125 rpm speed. For this, the initially selection reduction ratio i = 5.6 must be applied. The running time per stroke amounts then to t = 13.4 min, the operating speed to v = 223 mm/min. The is the sizing selected at this point in time.

When calculating several options, the optimum solution can be identified.

- Determination of the total actuator hollow shaft revolutions

For the following proof, all closing element movements and all stress cycles “lifting and lowering” at MWH measured water level and full MW supply level were recorded according to the product requirements. For simplification, all tendered stress cycles of one type of water level were added and considered as one cycle of 3 m strokes or double strokes (**table 2.10**).

Clearing and water regulation result in ≈ 30 double strokes/year or ≈ 60 strokes/year ≈ 2,100 strokes/35 years.

Tab. 2.10: Number of double strokes “Open-Close” per year

Water level	Water height	Double strokes
EHW	3.00 m	0.01
MHW	2.80 m	6
MW	2.50 m	24

The total figure of actuator hollow shaft revolutions results from:

$$R_{A-35y} = R_A \text{ /stroke} * \text{strokes/year} * t_{tot}$$

$$= 1{,}680 \; R_A \text{ / stroke} * 60 \text{ strokes/year} * 35 \text{ years}$$

$$R_{A-35y} = 3.528\text{m}$$

According to table 2.04, for SA 14.6 with

$R_{A\text{-}35y} < R_{A\text{-}HSS2}$
$3.528m < 4.0m$

the HSS2 application for hydraulic steel structures is confirmed. Now, proof for the respect of the defined run torques and later operational stability of selected gearboxes must be provided.

If $i_{GK} = 8{:}1$ had been selected as gearbox ratio, the result would have been

$R_{A\text{-}35y} > R_{A\text{-}HSS2}$
$5.040m > 4.0m$

The respective run torque $R_{A\text{-}HSS2} = 4.0m$ would have to be reduced by interpolation using the lifetime formula $L = (C/P)^p$ (2.1.4.1) or using the graph (figure 2.049).

2.1.5.6 Proof of run torques

The run torques of a cycle consisting of lifting and lowering shall be confirmed according to (2.1.4.3.2) and (2.1.4.3.3). For our calculation, it is sufficient to consider the resulting forces on the respective closing element positions (2.1.3.11) (**figure 2.051**).

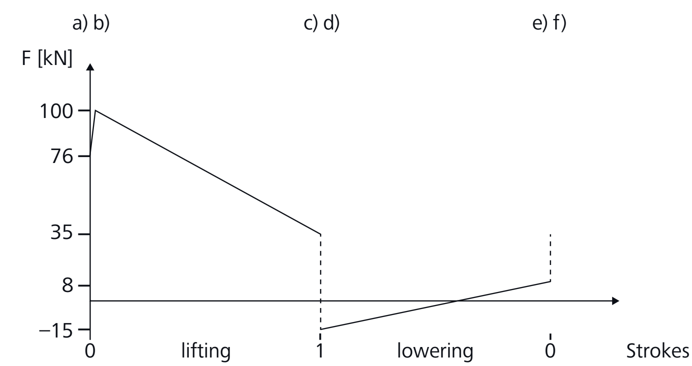


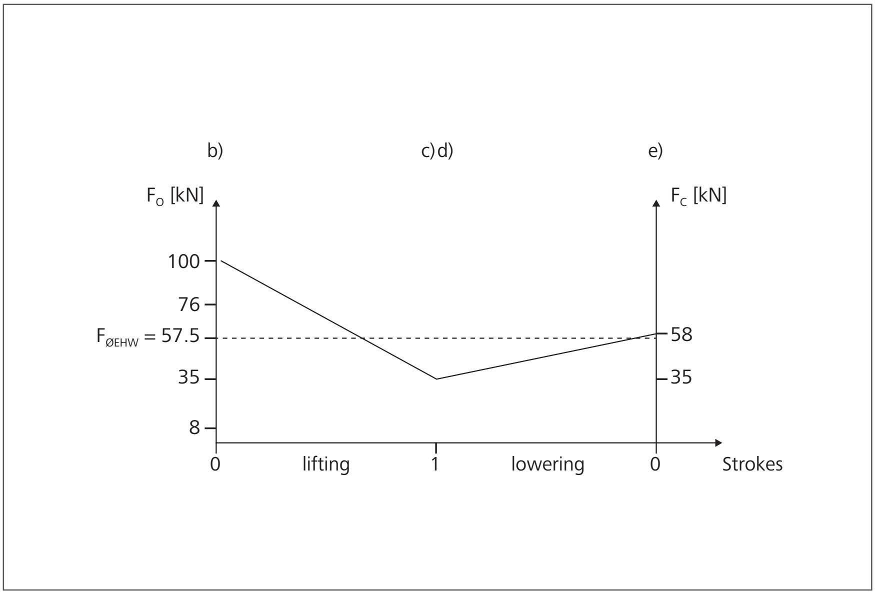
Figure 2.051: Calculated force cycle at sluice gate across one load cycle “lifting – lowering” for EHW

When lifting, in b) the particularity of the large pull opposes the opening torque a). Considering the subsequently increasing unseating torque – effectively between 76 kN to 100 kN – the force reduction is neglected while considering sections a) and b) for the calculation of the mean force impact. A tensile force of –15 kN results when starting lowering d). This means the sluice gate would operate autonomously in closing direction. The actuator would even have to brake. Once the friction force increases due to the hydrostatic pressure, pressure would have to be applied.

Since spindles are self-locking, at least the self-locking of spindle/spindle nut must be by-passed. The initially low force required rises up to the peak value in position 3). Following this, 50 kN in position f) is required to reach the necessary bed seal contact force.

Two possibilities are considered:

1. Due to investigations, it was noted that heating up of the motor occurs for both when braking while lowering and in no-load as well as during lifting. The driving power for lowering is difficult to define and therefore, in practical applications, the **sudden jump of the lifting/lowering graph** is neglected (**figure 2.052**). Upward parallel translation of line $F_d - F_e$ until F_d contacts the lifting line in F_c, a V-graph is obtained which is arithmetically averaged using the surface integral for the duty type proofs.



F_e – Force in position e)

Figure 2.052: Calculated force cycle at sluice gate across one load cycle "lifting – lowering" for EHW

$$F_{\varnothing EHW} = \frac{\frac{F_b + F_c}{2} + \frac{F_d + F_e}{2}}{2}$$

$$= \frac{\frac{100\ kN + 35\ kN}{2} + \frac{35\ kN + 58\ kN}{2}}{2}$$

$$F_{\varnothing EHW} = 57.5\ kN$$

This procedure has been proven appropriate in practical experience.

2. Due to the problems described above, many field experts replace the “double stroke” by “lift twice” to be on the safe side.

Calculation to be made:

$$F_{\varnothing EHW} = \frac{\frac{F_b + F_c}{2} + \frac{F_d + F_e}{2}}{2}$$

$$= \frac{\frac{100\ kN + 35\ kN}{2} + \frac{35\ kN + 58\ kN}{2}}{2}$$

$$F_{\varnothing EHW} = 67.5\ kN$$

The result is approximately 17 % above the force $F_{\varnothing EHW}$ = 57.5 kN, calculated for lifting and lowering. Excess is tolerable. When deciding in favour of this solution, the sum of all supplements must be taken into consideration.

2.1.5.6.1 Duty type proof of actuator

For duty type proof, the actuator must be capable of supplying a duty type run torque $T_{duty\ type}$ which is equal or higher than the mean torque $T_{A\text{-}\varnothing EHW}$ to be determined and this at EHW extreme high water. For providing proof, the average force

$F_{\varnothing EHW}$ = 57.5 kN

was converted to the corresponding mean torque using the computer program.

$T_{O\text{-}\varnothing EHW}$ = 527 Nm

Using the gear factor F1 = 5.0

$T_{A\text{-}\varnothing EHW}$ = 105 Nm

could be determined. According to table 2.05, the actuator in type of duty S2 - 30 min has a duty type run torque of $T_{A\text{-}S2\text{-}30\ min}$ = 125 Nm. Compared to

$T_{A\text{-}ØEHW} < T_{A\text{-}S2\text{-}30\ min}$

105 Nm < 125 Nm

the **fitness for purpose** is delivered.

2.1.5.6.2 Lifetime proof

- Lifetime proof of actuator

Lifetime or **proof of operational stability** must be provided on the basis of the mean equivalent torque for water levels to be considered. For MHW and MW water levels, like for EHW, the resulting forces within the respective closing element positions (2.1.3.11) were recorded (**table 2.11**). The graphs in **figure 2.053** were the result.

Tab. 2.11: Resulting forces for MHW mean high water and MW mean water (also refer to 2.1.3.11, figure 2.038)

Water level	MHW 2.8 m	MW 2.5 m	Water level	MMW 2.8 m	MW 2.5 m
Pos.	lifting [kN]		Pos.	lowering [kN]	
b)	95	80	d)	35	35
c)	35	35	e)	52	40

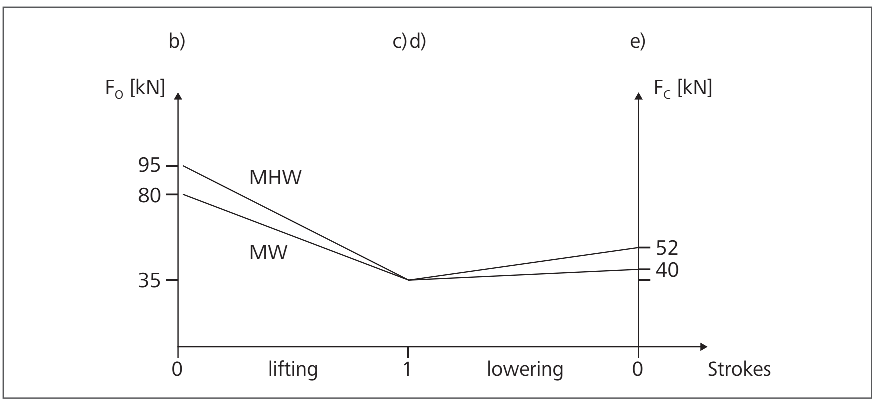


Figure 2.053: Calculated force cycle at sluice gate across one load cycle “lifting – lowering” for MHW and MW

With

$$F_E = \frac{F_{min} + 2F_{max}}{3}$$

F_E = Equivalent force (2.1.4.3.3)

results in equivalent force [12] for MHW:

$$F_{b)-c)} = \frac{35\text{ kN} + 2 * 95\text{ kN}}{3}$$

$$F_{b)-c)} = F_{\varnothing MHW} = 75\text{ kN}$$

and:

$$F_{d(-e)} = \frac{35\text{ kN} + 2 * 52\text{ kN}}{3}$$

$$F_{d(-e)} = 46\text{ kN}$$

For a double stroke of lifting and lowering results the mean force [1] for MHW:

$$F_{\varnothing MHW} = \sqrt[3]{\frac{F_{b)-c)}{}^{3} * s + F_{d)-e)}{}^{3} * s}{2 * s}}$$

wherein
$s = Stroke_{lifting} = Stroke_{lowering}$

$$F_{\varnothing MHW} = \sqrt[3]{\frac{F_{b)-c)}{}^{3} + F_{d)-e)}{}^{3}}{2}}$$

$$F_{\varnothing MHW} = \sqrt[3]{\frac{75\text{ kN}^3 + 46\text{ kN}^3}{2}}$$

$f_{\varnothing MHW} = 63$ kN

This corresponds to a torque of

$T_{\varnothing MHW} = 580$ Nm

Using the gear factor f1= 5.0, the following results:

$T_{A\text{-}\varnothing MHW} = 116$ Nm

For MW, the following is calculated:

$F_{b)-c)} = 65$ kN
$F_{d)-e)} = 38$ kN

$F_{\varnothing MW}$ = 54 kN
$T_{\varnothing MW}$ = 495 Nm
$T_{A\text{-}\varnothing MW}$ = 99 Nm

The mean torque was determined for both water levels according to the Palmgren Miner method. The number of double strokes from adding the proportional stress cycles within the product requirements resulted in:

T_{AO} remains unconsidered due to the low impact duration
$T_{A\text{-}ØHHW}$ with 0.01 double strokes/year is neglected
$T_{A\text{-}ØMHW}$ n_1: 6 double strokes/year,
$T_{A\text{-}ØMW}$ n_2: 24 double strokes/year,

Consequently:

$$T_{A-\varnothing} = \sum_{i-1}^{k} \frac{n_i * T_{Ai}}{N_i}$$

$$T_{A-\varnothing MHW/MW} = \frac{n_1 * T_{A-\varnothing MHW} + n_2 * T_{A-\varnothing MW}}{N_1 + N_2}$$

$$= \frac{6 * 116\ \mathrm{Nm} + 24 * 99\ \mathrm{Nm}}{6 + 24}$$

$$T_{A-\varnothing MHW/MW} = 102\ \mathrm{Nm}$$

With $T_{A\text{-}ØMHW/MW}$ = mean actuator torque at MHW/MW.

The conditions for lifetime proof is being fulfilled herewith (refer to table 2.06):

$T_{A\text{-}ØMHW/MH} < T_{A\text{-}HSS2}$
102 Nm < 135 Nm.

If the actual torque is lower than permitted run torque, heating up and wear are reduced allowing to increase the number of revolutions. Interpolation can be made (2.1.4.1 Lifetime formula or 2.1.4.5.3 Lifetime run torque figure).

- Lifetime test of bevel gearboxes

Possible gearbox revolutions must comply with the actuator hollow shaft revolutions specified for the respective application.

GK 25.2 bevel gearbox must

$$R_{GK-required} = \frac{R_{A-35y}}{i_{GK}}$$

$$= \frac{3.528\mathrm{m}}{5.6}$$

$$R_{GK-required} = 0.63\mathrm{m}$$

allow for 630,000 revolutions. According to the manufacturer's documents (refer to table 2.07), $R_{GK\text{-}HSS}$ = 600,000 revolutions are permitted at reduced torque T_{GK} = 700 Nm. If the actually required torque falls short of this value, lifetime can be respectively adapted to the ratio. An approximate calculation of the nominal gearbox lifetime is possible using the lifetime formula (2.1.4.1 Electric actuators).

$L = (C/P)^p$

C: T_{GK} = 700 Nm (refer to table 2.07)

$$\begin{aligned} P: T_{GKØ} &= T_{A\text{-}ØMHW/MW} * f1 \\ &= 102 \text{ Nm} * 5.0 \\ T_{GKØ} &= 510 \text{ Nm} \end{aligned}$$

wherein
L = lifetime factor
C corresponds to the permitted run torque T_{GK}, according to the application
P corresponds to the calculated mean run torque $T_{GKØ}$.
p = 3 applies to ball bearings or gearings

T_{GK} Lifetime run torque of bevel gearbox

$T_{GKØ}$ Mean torque at bevel gearbox
$T_{A\text{-}ØMHW/MW}$ Mean torque at actuator
f1 = Conversion factor from output torque to input torque (refer to table 2.07)

Required 510 Nm are opposed to permitted 700 Nm which means that the bevel gearbox still has reserves. The lifetime factor results in:

$$\begin{aligned} L &= (T_{GK}/T_{GKØ})^3 \\ &= (700 \text{ Nm}/510 \text{ Nm})^3 \\ L &= 2.57 \end{aligned}$$

The adjusted nominal run power is:

$$\begin{aligned} R_{GK\text{-nominal}} &= R_{GK\text{-HSS}} * l \\ &= 0.6\text{m} * 2.57 \\ R_{GK\text{-nominal}} &= 1.5\text{m} \end{aligned}$$

wherein
$R_{GK\text{-nominal}}$ = Bevel gearbox revolutions according to load
$R_{GK\text{-HSS}}$ = Bevel gearbox revolutions at run torque (refer to table 2.7)

The nominal value is rather generous (**table 2.12**). If this should be insufficient for extreme individual solutions, the manufacturer offers specific revision solutions. For example, a gearbox maintenance scheme should be specified in regular intervals of 10 years. Maintenance includes lubricant change and sealing ring replacement as well as verification of

Tab. 2.12: Revolutions of bevel gearboxes at the calculated spindle actuator

U_{GK}/stroke	300
$U_{GK\text{-HSS}}$	600,000
$U_{GK\text{-35y}}$	530,000
$U_{GK\text{-nominal}}$	1,500,000

gear wheels, worm shaft and worm wheel and their replacement if necessary. During this "status as new", further operation is not required. A viable alternative would be to decide in favour for the next higher gearbox size. The true necessity is subject to individual consideration of the application.

2.1.5.7 Proof of self-locking

For the present case and according to the tender, lifting and lowering of the closing elements are to be implemented by means of a trapezoidal threaded spindle. A combination of actuator and two bevel gearboxes is the ideal solution. For this, the bevel gearboxes are not self-locking at an efficiency around $\eta = 0.9$ and the selected actuator with speed n = 125 rpm and an efficiency $\eta > 0.55$. The spindle nut-spindle combination provides sufficient self-locking (2.1.4.2).

2.1.5.8 Evaluation of example calculation

The following torques apply for an actuator SA 14.6 in version 125 rpm – S2 - 30 min – in application HSS2:

Permissible torque

$T_{Amax.\ S2\text{-}30min}$	360 Nm

Opening torques:

$T_{A\text{-}OEHW}$	184 Nm
$T_{A\text{-}OEHW+25\ \%}$	230 Nm

Mean torque for EHW and type of duty:

$T_{A\text{-}ØEHW}$	105 Nm

Mean torque for MHW/MW, for lifetime:

$T_{A\text{-}ØMHW/MW}$	102 Nm

Closing torques

$T_{A\text{-}CEHW}$	60 Nm
$T_{A\text{-}CEHW+25\ \%}$	75 Nm

- Starting torque

With a speed of 125 rpm in type of duty S2 - 30 min, the SA 14.6 has a maximum torque of 360 Nm. Consequently, the opening torque required for a new installation of $T_{AO} = 184$ Nm and the supplement of 25 % as demanded by DIN 19704-1/8.4 are met. The unseating torque amounts to a value between $T_{A\text{-}OEHW}$ and $T_{A\text{-}OEHW+25\,\%}$ after ageing and resulting uncertainties. The lower closing torque did not have to be considered here.

- Duty type run torque

SA 14.6 in type of duty S2-30 min and a mean torque of $T_{A\text{-}ØEHW}$ = 105 Nm is suited for EHW extreme high water conditions.

- Lifetime run torque

The actuator allows for minimum 4.0m output drive revolutions during more than 35 with an average of $T_{A\text{-}ØMHW/MW}$ = 102 Nm in HSS2 applications. 3.528m revolutions are required. This corresponds to the nominal lifetime of the selected bevel gearbox.

- Self-locking

Sufficient self-locking is provided for the selected actuator/gearbox connecting element combination by the single-pitch spindles.

- Number of strokes

The actuators can operate 2.2 subsequent strokes at 125 rpm. Higher output speed selection is possible if more strokes are required. On request, type of duty S2-60 min can be used.

- Permissible operating speed

According to DIN 19704-2/9.2.2, the speed of the closing element should not exceed 0.1 through 1.0 m/min when reaching the end position. The tender specified the operating speed at v = (200 … 400) mm/min. The calculated operating speed v = 223 mm/min is permissible.

Consultants increasingly demand not only the specification of calculated torques but also the specification of definite actuator and gearbox sizes to avoid sizing errors.

2.1.6 Proofs for exceptional loads

Contrary to the previously discussed dynamic procedures, **static stress** is also an issue. Load is applied without movement; no wear can be detected; there is no impact on the lifetime (operational stability). However, the safety factor 2 – or even rather 1.5 – of the rated torque should not be exceeded to avoid plastic deformation, cracks or breaking.

2.1.6.1 One-sided stop for two-sided operated closing elements

In case of accidents, this means interruption of connecting lines, the respectively other line – besides single or provisional driving which has to be specified in the tender – must guarantee **one-sided stop** of the closing element (DIN 19704-1/5.4.1a). The holding forces or the holding torque of the appropriate elements like actuator, gearbox and connecting elements must be sufficiently high. In the present case, the spindle-spindle nut connection can assume this task. Consequently, the gearbox is not subjected to load.

For non self-locking connecting elements, the gear wheels of the subsequent gearbox are subjected to load. In our example, we are dealing with a non self-locking GK 25.2 bevel gearbox. With the permissible torque $T_{maxGear}$ = 2,000 Nm and the safety factor to S = 2 this results in:

$T_O = 920 \text{ Nm} \leq T_{maxGear} = 4{,}000 \text{ Nm}$

Consequently, the gearbox with its geared wheels is suitable for this load. However, holding must be adopted by a self-locking member – actuator, brake motor or anti-backdrive device. A self-locking worm gearbox is subjected to identical load – worm shaft/worm wheel – but can take over the holding task.

Bevel, spur, worm gearboxes and actuators provide double break resistance irrespective of the available self-locking capability.

2.1.6.2 Exceptional impacts of actuators in case of an accident

Besides the considerations made previously, consultants for civil engineering constructions for water applications must bear in mind the following issues:

According to DIN 19704-1/5.5, "In case of a one-sided impact of an obstruction, for example caused by **blocking of the closing element** due to jamming or freezing, the complete driving power is adopted by this line. As a consequence, the highest possible impacts of the actuator driving forces/torques onto the closing element .... must be recorded". This applies to the opening and closing directions.

"**Actuator impacts**, like for example the motor stall torque, the break torque or the setting values (which must be sealed against inadvertent modification) are considered as **controlled changeable impacts**. Their values must be included into dimensioning as characteristic values of highest force transmission. The repercussions .... are required for providing proof of structural safety within all essential machine parts and must be observed until final connecting to the steel construction" (DIN 19 704-1/8.4). Possibly an excessive torque is applied to the hydraulic steel structures like screws, bolts, bearings, welded seams, fastenings, connecting elements including shafts with their output drive sleeves and parallel keys which could be subject to damage.

Parallel keys of different qualities are available. The degree of damage is the indicator for load impact. However, any failure must be excluded.

For this, the often higher torque in opening direction is to be considered. Section 2.18 "Buckling safety" provides information on buckling safety for connecting elements caused by the closing torque.

"When dealing with electric motors, controlled by frequency converters, the driving torque – for considering highest possible force transmission – must be selected at least 5 % higher than the required driving torque (DIN 19704-1/8.3).

$T_A = 1.05 * T_{Amax}$

According to DIN 19704-1/8.4, the required driving torque should be increased by at least 25 % upon setting:

$T_{ASetting} = 1.25 * T_A$

The torque setting values are to be increased by supplements – partial safety factors γ_F. The standards (DIN 19704-1/tab.5/no.14 and tab.6) comprise safety factors for actuator impacts on various constructions or potential loads. For example, the impact of the stall torque for actuators without torque limitation has to be increased by factor $\gamma_F = 1.1$ or the impact of failing additional equipment like overload protection devices, safety couplings and thyristor controllers with factor $\gamma_F = 1.35$. Hereby, any impact listed in (DIN 19704-1/tab.6) has to be individually examined (DIN 19704-1/9.3) and assessed. Consequently, the stall torque for compact actuators in which limit and torque switches monitor the actuator travel does not have to be considered.

$T_{ALoad} = \gamma_F * T_{ASetting}$

"The set torque may be integrated within the proof of structural safety if secured by sealing, and if it cannot be exceeded during assembly, repair or maintenance." (DIN 19704-1/8.4)

This must be sufficient for the resistance variables of machine parts used for proof of structural safety (DIN 19704-1/9.4). The rated values of yield strength $f_{y,\,d}$ or of the 0.2 % elastic limit $f_{0.\,2,\,d}$ with partial safety factor $\gamma_M = 1.5$ are to be calculated as **load capacities**.

The load capacity is increased by multiplying the impacts with the safety factor γ_F (F = force) (DIN 19704-1 table 5/no.14 or DIN 19704-1 table 6). The permissible material load capacity is reduced when dividing by γ_M (M = material).

Hydraulic steel structure experts talk about extrapolation – determination of potential high forces in extreme situations – and about the subsequent breaking down – impact on the reduced load capacity of the additional equipment affected.

2.1.7 Examples for actuator gearbox combination set-up arrangements

According to our calculation example, two trapezoidal threaded spindles – one with left-handed thread, one with right-handed thread – are assigned to a combination between an actuator and two bevel gearboxes. If the bevel gearboxes are deployed with two

Figure 2.054: Actuator and 2 bevel gearboxes with adjustable output direction of rotation

exchangeable gear sets whereby the output direction of rotation is adjustable, two identical spindles can be used (**figure 2.054**).

Spindle nuts of output drive types A according to EN ISO 5210 or – in civil engineering constructions for water applications preferred – types AK allow pull and thrust movements (**figure 2.055**).

Figure 2.055: Bevel gearboxes with stem nut or spindle nut with lubrication point for bearings

Figure 2.056: Elastic safety coupling for torque transmission

The torque is often transmitted by **elastic safety couplings** (**figure 2.056**) allowing easy connection and separation.

For mere manual operation or actuator set-up, an input gearbox equipped with a handwheel can be used in lieu of an actuator (**figure 2.057**). The reduction ratio for the smallest gearbox is 1:1. When selecting larger gearboxes, the desired reduction ratio must be considered for actuator selection. According to EN ISO 3210/5210, **output drive type D/D** or **B3/D** may be selected for shaft connections.

A further option is inserting a shaft equipped with parallel keys at the end and into the provided actuator position through the actuator (positioned in the centre) specifically the

Figure 2.057: Bevel gearboxes with electric actuator, central mount

Figure 2.058: Bevel gearbox with through-shaft

Figure 2.059: Bevel gearboxes with electric actuator, side mount

gearbox. The protruding part connects with a second shaft with sleeve coupling or is inserted into the opposite gearbox (**figure 2.058**).

The actuator can be mounted directly to one of the two bevel gearboxes thus making the centre support post obsolete (**figure 2.059**). In this case, the actuator must be provided with output drive type B3/D.

The objective is to optimally select and combine the required system parts once the calculation has lead to results which can be implemented. The mounting position of all actuators and gearboxes can be selected as desired. Any questions relating to safety, accessibility and maintenance must be specifically answered now.

Any options regarding the tendered fieldbus control and further particularities are detailed in section 4.3.1 or in the manufacturer's documents.

2.1.8 Buckling safety

Spindles must be capable to support, lift and lower loads. Therefore, the material selection is particularly important for "cross section pressure load capacity" and permissible "thread surface pressure". The required core cross-section (2.1.5.2) for lifting is calculated on the basis of the stress proof (DIN 18800-1/747):

$$\frac{\sigma_d}{\sigma_{R,d}} \leq 1$$

wherein

σ_d = Normal stress
$\sigma_{R,d}$ = Limit normal stress

In familiar terms, it is often said that the giant gate is heavy enough to fall down by itself for closing. Besides the self-locking effect, this is quite true. And often, customers demand this feature. Instead of the 12 mm gate leaf, selection is then made of a more compact version for the increased weight. Besides this case, a sluice gate must often be pushed downward without self-locking against friction and buoyancy. Due to the weight of the closing element and pull, this downward movement is often much easier than pulling up. In the event of buckling risks, increased safety is necessary if the lifting and lowering force requirements for closing are identical. In this instance, a large spindle diameter is calculated for the closing operation.

2.1.8.1 Extreme loads

Experts just know it! They might talk about "straws" once they consider a spindle as too weak.

Demands on buckling safety: The length of a spindle may not exceed the specifications. With increasing free length and thus higher slenderness ratios, the risk of buckling during closing increases. In turn, this calls for larger spindle diameters.

The double-spindle installation provided to bear larger tensile forces – since each spindle bears 50 % of the total load – might work perfectly during many years. The difficulties start, when

- **Exceptional load cases** occur causing the failure of a gearbox or a connecting element. Once closing is required, the fully set actuator torque is transmitted to the second installation part which is still working. For torsion-resistant fish-belly flap gates equipped with lantern gears or electrical lifting cylinders, this may lead to a **provisional single-sided driving** (DIN 19704-1/5.4.1b). For sluice gates with narrow guide runners, this could lead to jamming or blocking. The customers must provide the evaluation of the true potential incident to be considered (DIN 19704-1/5.4.1).
- Eccentric gate loads occur caused by big branches or car wheels or if further movement is impeded on one side due to bed loads or freezing. This case (DIN 19704-1/5.4.2) must also be notified by the customer for evaluation.

In both incidents when sluice gates are blocked, the total driving torque initially calculated for two sides will then be applied to one side only. This can cause buckling of one of the connecting elements since it was only sized for 50 % of the driving torque (**figure 2.060**).

Figure 2.060: Buckled spindle

Consequently, according to DIN 19704-1/5.5, the highest possible impacts of driving forces/ torques of the actuator on the closing element – or the connecting elements – have to be considered when jamming or freezing occurs. The following buckling proof must be conducted at 100 % closing force plus safety factor.

2.1.8.2 Calculation of buckling safety

Specialist literature and calculation programs are available for appropriate sizing of spindles. The following is a brief comment on buckling safety:

> ***The verification of buckling safety for a connecting element must be performed using the rated tensile force (100 %) considering the tripping torque – according to DIN 19704-1/8.4, the tripping torque should exceed the required driving torque by minimum 25 % – and the partial safety factor for the rated values of loads (impacts)(DIN 19704-1/tab.6.1) γ_F = 1.35 according to DIN 18800-2 (issue Nov 1990) or using a different method. For this, the partial safety factor for load capacities (resistance variables) (DIN 19 704-1/9.4) γ_M = 1.5 must be considered.***

For one-piece bars, the verification of buckling resistance has to be performed according to DIN 18800-2/3.

Calculation procedure

According to DIN 18800-2 (issue Nov 1990), the following requirements must be fulfilled for proof of structural safety:

$$\frac{\text{load}}{\text{load capacity}} \leq 1$$

According to DIN 19704-1/8.4, the load amounts to:

$$\begin{aligned} N_K &= 1.25 * N \\ &= 1.25 * 50 \text{ kN} \\ N_K &= 64 \text{ kN} \end{aligned}$$

wherein

N_K = Characteristic value of N

N = Normal force = F_S (example calculation)

Load capacity is to be calculated as follows:

- Length of buckling S_K

  $S_K = \beta * l$

  $S_K = 1 * 300 \text{ cm} = 300 \text{ cm}$

wherein

Euler buckling case 2: $\beta = 1$

l = Spindle length

- Slenderness ratio λ_K

$$\lambda_K = \frac{S_K}{i}$$

Radius of inertia z: $i = \sqrt{\frac{I_Z}{A}} = \sqrt{\frac{r^4 * \pi}{r^2 * \pi * 4}} = 1.725 \text{ cm}$

$$\lambda_K = \frac{300 \text{ cm}}{1.725 \text{ cm}} = 173.9$$

wherein

I_Z = Area moment of second degree

A = Core cross-section

d_c = Core diameter = 6.9 cm

r = $d_c/2$

- Rated slenderness ratio $\overline{\lambda_K}$

With reference slenderness ratio $\lambda_s = \pi\sqrt{\frac{E}{f_{YK}}} = \pi\sqrt{\frac{21{,}000 \text{ kN / cm}^2}{65 \text{ kN / cm}^2}} = 56.47$

Material: 42CrMo4
E = Elasticity module = 21,000 kN/cm²
f_{yc} = Yield strength = 65.0 kN/cm²

$$\overline{\lambda_K} = \frac{\lambda_K}{\lambda_S}$$

$$\overline{\lambda_K} = \frac{173.9}{56.47}$$

$$\overline{\lambda_K} = 3.08$$

Area for calculation of κ: $\lambda_K > 3.0$ (simplifying) 3.08 > 3.0

- Reduction factor κ

Parameter α for buckling stress curve c for full cross-section of circle α = 0.49

$$\kappa = \frac{1}{\overline{\lambda_K}\,(\overline{\lambda_K} + \alpha)}$$

$$\kappa = \frac{1}{3.08\,(3.08 + 0.49)}$$

$$\kappa = 0.091$$

- Normal force in plastic condition $N_{pl,d}$

$$N_{pl,d} = A * \sigma_{R,d}$$

$$= A * \frac{f_{YK}}{\gamma_M}$$

$$= \frac{\pi}{4} * 6.9^2\ cm^2 * \frac{65\ kN / cm^2}{1.5}$$

$$N_{pl,d} = 1{,}620\ kN$$

wherein
$\sigma_{R,d}$ = Limit normal stress
γ_M = Partial safety factor for resistance variables (load capacities),
For mechanical engineering $\gamma_M = 1.5$,
For static steel construction applies $\gamma_M = 1.1$

- Proof of buckling resistance

$$\frac{N_d}{\kappa * N_{pl,d}} < 1$$

$$\frac{\gamma_F * N_K}{\kappa * N_{pl,d}} < 1$$

$$\frac{1.35 * 64\ kN}{0.091 * 1{,}620\ kN} = 0.58 < 1$$

wherein
N_d = Rated value of N_K;
N_d = $\gamma_F * N_K$
γ_F = Partial safety factor for impacts (loads), $\gamma_F = 1.35$

Consequently, the selected spindle offers double safety. The demand for buckling resistance is fulfilled!

- Proof of buckling resistance with minimum adjustable actuator torque

Section 2.1.5.4 defined that the torque switch of the actuator to be selected must be set to minimum $T_{AC+25\ \%}$ = 75 Nm for closing direction. SA 14.6 was selected with a torque range of 200 Nm to 360 Nm. This means that setting to 75 Nm is not possible. When referring to **table 2.13**, it becomes obvious that working with a reduced torque range between 100 Nm and 250 Nm is possible. The lowest adjustable value is 100 Nm. DIN 19704-1/8.4 defines that the adjusted torque should be at least 25 % above the required actuator torque according to 8.3. Consequently, 100 Nm must or should be adjusted. Can the spindle cope with this value?

Tab. 2.13: Technical data Multi-turn actuators – reduced torques (extract)

Type	Torque ranges [Nm] in duty type	
	S2 - 15 min	S2 - 30 min
SA 14.2 standard	100 – 250	100 – 180
SA 14.2 reduced	40 – 120	
SA 14.6 standard	200 – 500	200 – 360
SA 14.6 reduced	100 – 250	
SA 16.2 standard	400 – 1 000	400 – 710
SA 16.2 reduced	200 – 800	

Reverse calculation results in:

$$\begin{aligned} T_C &= T_{AC} * f1 \\ &= 100\ \text{Nm} * 5.0 \\ T_C &= 500\ \text{Nm} \end{aligned}$$

wherein
T_C = Closing torque
T_{AC} = Actuator closing torque

f1 = Conversion factor from output torque to input torque
F_C = Closing force

Result using our computer program:

$F_C = 85$ kN.

Proof of buckling resistance

$$\frac{\gamma_F * N_K}{\kappa * N_{pl,d}} < 1$$

$$\frac{1.35 * 85\ \text{kN}}{0.091 * 1{,}620\ \text{kN}} = 0.78 < 1$$

The result makes obvious, that the spindle is fit for withstanding increased loads, further to the fact that when introducing $\gamma_M = 1.5$, the load capacity was reduced to be on the safe side.

For submerged gates – with longer spindle lengths or spindle extensions – the permissible free buckling length is respected by lever arrangement guides in distances of for example 2 m or 3 m.

2.1.8.3 Supporting measures

What more is to be done?

- Torque seating must basically be considered (DIN 19704-2/9.2.3). Then another functional switch is available in addition to the limit switch, resulting in virtually 100 % reliability for end position seating.
- Refrain from excessive torque adjustments in closing direction? The opening and closing torque can be independently adjusted at the actuator. For two-sided driven closing elements, the total torque is applied on the second spindle if the first spindle fails. The additional adjustment tolerance of 25 % as requested by the DIN standard applies for both opening and closing torques. This requirement is justified since calculation of characteristic variables is connected to many uncertainties and the closing operation must imperatively be guaranteed. Higher value adjustments must be enabled. This higher value is extremely important for sufficient dimensioning. This also applies to the spindles. Once all these conditions are fulfilled, the closing procedure with low margin closing torque might be started!
- Making the position of the closing element evident. Probably due to tight dimensioning, spindle buckling occurred in an installation since the operator could not see the closing element position following automatic actuator cut-off caused by the brown water. As usual, he started again to push the remaining deposits to one side. But this time, the total torque acted on only one spindle since a branch of 10cm had jammed the second one. This was sufficient to cause spindle buckling. For this reason, it becomes increasingly important to improve **visualisation** by level rods (**figure 2.061**) or electronic equipment to reliably identify the closing element position and to allow the operator, like in

Figure 2.061: Visualisation by means of a level rod at a double gate weir

Figure 2.062: Visualisation of a lantern gear system behind a safety cover

the present case, to remove the obstacle prior to closing. In **figure 2.062**, improved visualisation and accident prevention measures were combined.

- Deploy actuators with reduced torques if necessary. In contrast to standard equipment, actuators may be provided with weaker spring packs to achieve torques other than standard torques. This way, it is possible to monitor an appropriate opening torque and a frequently lower closing torque without requiring to adapt the buckling safety of the connecting elements to the higher standard torque range.

For certain applications, weaker spindles can be used.

Reduced torques for opening and closing directions or for one direction only are available. They can be set irrespectively of each other. However, even for reduced torques, the total motor stall torque would be effective in the event of switch failure (refer to table 2.05 in section 2.1.4.5.1).

Mention is often made that the adjustable reduction ratio is not sufficient while hoping to further reduce the spindle size. But even sluice gates are worn by the years: slide strips and sealing strips. Bearings and spindle lubrication were possibly neglected and the whole installation is resinified. Let’s hope now that the actuator is strong enough. Otherwise, there is no way out – the system does not longer open or close as required. To remedy this development, torque reserve and spindle dimensioning must be sufficiently generous.

Larger spindle dimensioning improves smooth running and provides the advantage that the pertaining spindle nut is less subject to wear.

- Use safety couplings with torque monitoring. In recent years, the advocates in favour of slim spindles have won another ace. In addition to the actuator torque switches – which jointly monitor both spindle loads – lower adjustable safety couplings can now be deployed separately for each side. The calculation of the spindle buckling in end position seating can now be made at 50 % (+25 %) torque like for conventional applications and not at 100 % (+25 %) torque to account for potential accidents.

Figure 2.063 shows the working mechanism of safety couplings.

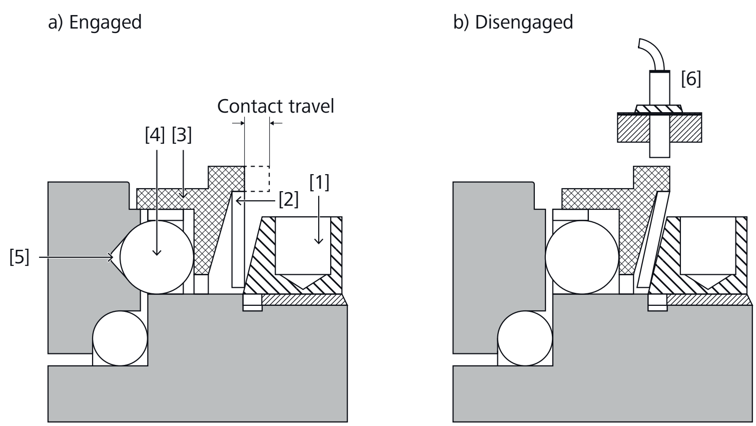


Figure 2.063: Safety coupling

Balls (4) are pressed into grooves (5) by means of spring force (2). If the torques exceed the force of the disc springs, the balls jump out of the groves and torque is no longer transmitted. The disengaging switching ring (3) causes the proximity switches (6) to electrically trip the actuator (**figure 2.064**).

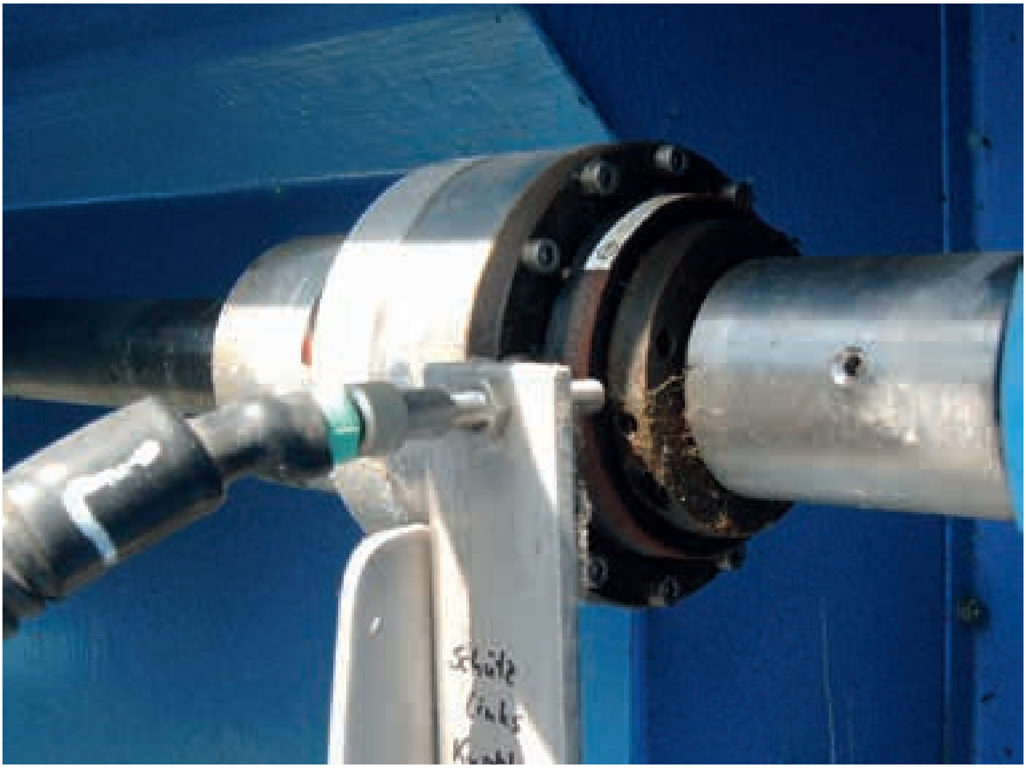


Figure 2.064: Slip clutch with proximity switches for electric tripping

At standstill, the safety coupling may not be used to support a load (DIN 19704-2/9.2.6). For this reason, the safety coupling is always placed between the actuator and a self-locking element like a spindle nut/spindle, worm gearing or anti-backdrive device (**figure 2.065**).

But safety couplings also have their issues. Although the tripping torques can be modified via adjusting nuts (1), the switching point is not constant in the long run. On the one hand, there is the influence of all known environmental impacts, in particular temperature and ageing. So it might occur that under certain conditions the safety coupling might either disconnect too late – then the acting torque is already excessive – or not disconnect at all prior to torque tripping. For safety reasons, customers charge specialists with disconnecting and refitting the couplings within the framework of conventional maintenance intervals. Of course, this is paired with wear. Therefore, it is advised to reduce the setting torque when performing these checks.

Even if a coupling has disconnected as expected due to disturbing obstacles, there is a need for action. The system is out of balance and must be equilibrated again which might be more or less demanding. This bears certain reservations.

Figure 2.065: Elastic safety couplings with torque monitoring at an actuator

Reservations against safety couplings for motor operation are made by users since they can only be set to one value – in general the lower closing torque is used. Uninterrupted opening is therefore no longer possible in case of sluggishness.

2.1.9 Manual operation

- Calculation of the required manual forces

According to DIN 19704-2/9.2.7, mechanical actuators must be equipped with manual drives. This is requires for stroke setting and safeguards emergency operation. For this, it is important that decoupling is performed between motor and drive, in the non self-locking range, so that change-over is still easy even when torque is applied. Priority is always awarded to motor operation. This means, once an electrical signal is set, motor operation must automatically overrule manual operation. This requirement occurred once in installations when motor operation could not be engaged after manual operation and the control centre could not start the actuator. Furthermore, the handwheel must not rotate during motor operation to prevent any injury hazard.

According to our example calculation in section 2.1.5.4, SA 14.6 can be deployed. n = 125 rpm was selected as output speed (**table 2.14**). At maximum torque T_{max} = 500 Nm in type of duty S2 - 15 min and reduction ratio 5.5:1 while using the standard handweel $d_{Standard}$ = 400 mm, results in the required manual force $F_{HaStandard}$ = 741 N at an handwheel input torque of T_{AHamax} = 148 Nm.

Tab. 2.14: Manual forces at AUMA actuators (extract)

Multi-turn actuator	Torque max. [Nm]	Output speed [rpm]	Hand-wheel reduction ratio	Standard handwheel dia. [mm]	Required manual force [N]	Option handwheel dia [mm]	Required manual force [N]	Torque at handwheel [Nm]
SA 14.6	500 500 500	90 125 180	8 : 1 5.5 : 1 4 : 1	400 400 400	758 741 808	500 500 500	606 593 646	152 148 162

Using the calculated actuator opening torque T_{AO} = 184 Nm, the actual input torque at the handwheel can be calculated as follows:

$$T_{AHaO} = \frac{T_{AO}}{i_{Ha} * \eta}$$

$$= \frac{184\ \mathrm{Nm}}{5.5 * 0.4}$$

$$T_{AHaO} = 84\ \mathrm{Nm}$$

wherein

T_{AHaO} = Opening torque at handwheel
T_{AO} = Actuator opening torque
i_{Ha} = Handwheel transmission ratio
η = Efficiency of worm gearing, single-stage

DIN 19704-1/8.3 demands: "The manual force should be between 80 N and 100 N per person and not exceed 250 N for a short time". The following calculation is to be made:

a) Standard handwheel:

$$F_{HaOStandard} = \frac{T_{AHaO}}{0.5 * d_{Standard}}$$

$$= \frac{84\ \mathrm{Nm}}{0.5 * 0.4\ \mathrm{m}}$$

$$F_{HaOStandard} = 420\ \mathrm{N}$$

b) Optional handwheel:

The standard handwheel can be replaced by a handwheel one size up. Using even larger handwheels is not permitted since this could damage the actuator and the surrounding fittings.

$$F_{HaOOption} = \frac{T_{AHaO}}{0.5 * d_{Option}} = \frac{84\ \text{Nm}}{0.5 * 0.5\ \text{m}}$$

$$F_{HaOOption} = 336\ \text{N}$$

The results do not yet comply to DIN. Therefore, the following reductions can be made:

If the planetary gearing preceding the handwheel is replaced by a transmission ratio 1:1, the handwheel transmission ratio multiplies by four.

For an output revolution of our example actuator, the handwheel must by rotated by 22 revolutions and not by 5.5. Whereby the torques applied are divided by four. The same applies for all other transmission ratios.

Calculation for 125 rpm and i = 5.5:

$$T_{HaO} = \frac{T_{AO}}{i_{Ha} * i_{PG} * \eta} = \frac{184\ \text{Nm}}{5.5 * 4 * 0.4}$$

$$T_{HaO} = 21\ \text{Nm}$$

wherein

i_{Ha} = Handwheel transmission ratio

i_{PG} = Planetary gear ratio

a) Standard handwheel:

$$F_{HaOStandard} = \frac{T_{AHaO}}{0.5 * d_{Standard}} = \frac{21\ \text{Nm}}{0.5 * 0.4\text{m}}$$

$$F_{HaOStandard} = 105\ \text{N}$$

b) Optional handwheel:

$$F_{HaOOption} = \frac{T_{AHaO}}{0.5 * d_{Option}}$$

$$= \frac{21 \text{ Nm}}{0.5 * 0.5 \text{ m}}$$

$$F_{HaOOption} = 84 \text{ N}$$

Consequently, the requirement of DIN 19704-1/8.3 is fulfilled.

The operating speed is calculated using the estimated manual speed n_{Ha} = 20 rpm:

wherein
P = Thread pitch
v_{Ha} = Operating speed using handwheel
n_{Ha} = Mean handwheel speed
i_{GK} = Spur gearbox transmission ratio

to:

$$v_{Ha} = \frac{P * n_{Ha}}{i_{Ha} * i_{GK}}$$

$$= \frac{10 \text{ mm} * 20 \ \frac{1}{\text{min}}}{5.5 * 5.6}$$

$$v_{Ha} = 6.5 \text{ mm / min}$$

and is consequently permissible (DIN 19704-2/9.2.2). The calculation cannot be put into practice due to the rest periods granted to the operator.

Calculation of the operating speed makes only sense for motor operation.

For closing operation, the calculation shall be repeated using the standard handwheel and the 1:1 transmission ratio for lifting, to satisfy DIN requirements. The actuator closing torque 60 Nm was determined in section 2.1.5.4. This includes the bed seal contact force F_{AC} = 20 kN for safe closing (DIN 19704-1/7.6.3).

Actually, the closing element is only lowered until the gasket is slightly bent. Field experience shows, that less than half the calculated maximum value is sufficient. In this position, the limit switch is set to “close”. In standard application, the torque switches are not used.

However, calculation shall be made with an actuator closing torque of 60 Nm.

$$T_{HaC} = \frac{T_{AC}}{i_{Ha} * i_{PG} * \eta}$$

$$= \frac{60\ Nm}{5.5 * 4 * 0.4}$$

$$T_{HaC} = 6.8\ Nm$$

$$F_{HaCStandard} = \frac{T_{AHaC}}{0.5 * d_{Standard}}$$

$$= \frac{6.8\ Nm}{0.5 * 0.4\ m}$$

$$F_{HaCStandard} = 34\ N$$

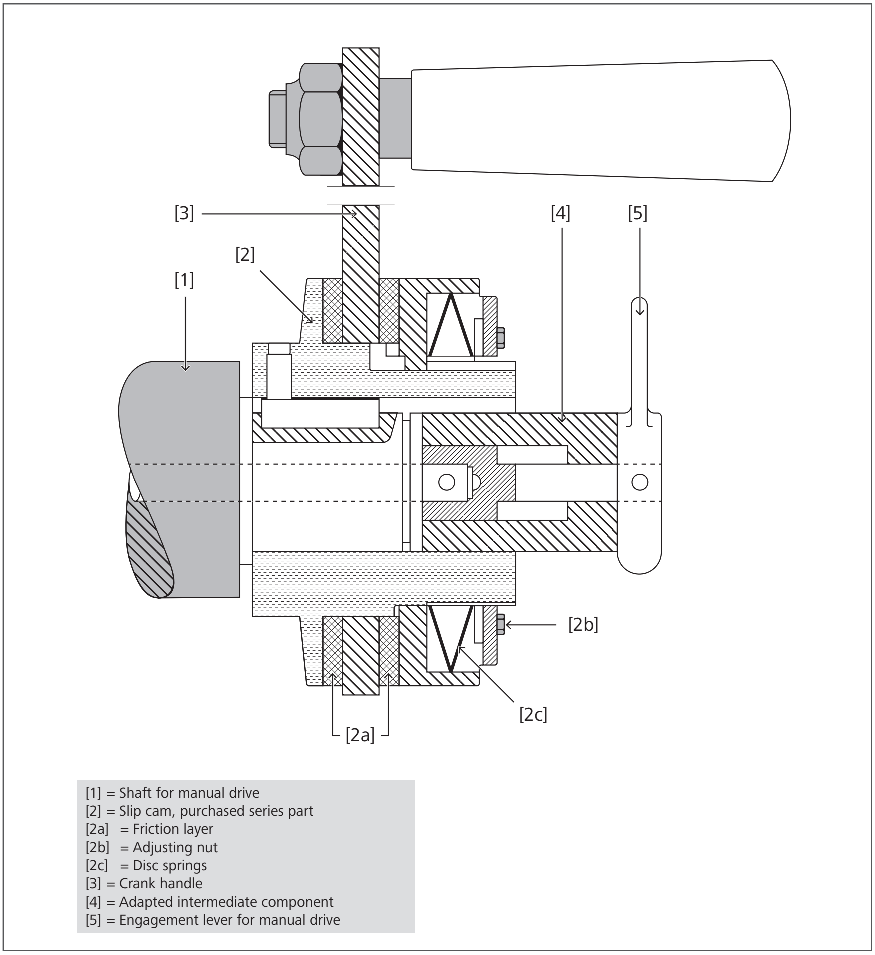


Figure 2.066: Slip clutch at actuator handwheel

For the low required force, torques can increase to dangerous levels.

When turning the handwheel, a torque rise might be unnoticed during travel or in closed position. Therefore, a protection is demanded to protect against the consequences of these "assumed" low torques. Three possibilities are available:

- Maintain planetary gear ratio iPG = 4:1.
- When reaching the preset torque value in motor operation, the actuator is switched off and the pertaining signal lamp is illuminated. Torque switches trip during manual operation. A lamp with UPS must be wired to clearly recognise the switching off.
- A slip clutch provided at the handwheel (**figure 2.066**) shall protect the spindle against excessive torque.

In turn, if the coupling is adjusted to the closing torque, it also reacts when opening at this low setting value (also refer to 2.1.8.3 – Safety couplings with torque monitoring). Therefore, the following must be observed:

Priority must always be awarded to emergency operation! The requirement for safe opening after longer standstill times must be complied with. There is also the risk that branches are located below the sluice gates and they must be removed with force. Alternating closing and opening can help to flush the installation.

Figure 2.067: Handwheel/motor change-over only possible using tools

Figure 2.068: A stairwell for handwheel operation and maintenance

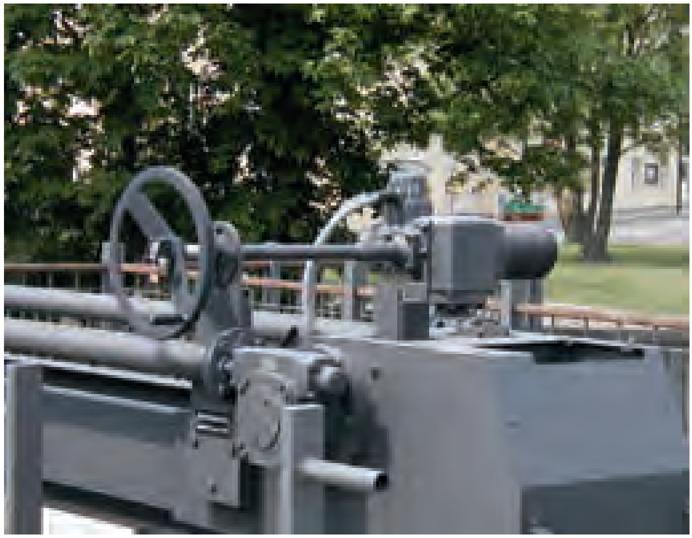


Figure 2.069: Actuator with handwheel spindle extension

Therefore, the implementation of a slip clutch deserves thorough consideration. Selecting a smaller handwheel could turn out as a viable solution – in this case d = 250 mm – equipped with a ball handle. This helps to increase the required manual force again. The ball handle facilitates turning and the operator is in command at any time!

Figure 2.070: Manual operation via Bowden cable and link chain

Figure 2.071: Manual operation with covered chain guide

- Accessibility of handwheel and motor change-over mechanism

For specific reasons, actuator mounting at heights might be required so that the operator can only access the actuator using a ladder or other equipment (**figure 2.067**).

Erecting a pedestal essentially facilitates manual operation and lubrication (**figure 2.068**).

Figure 2.072: Motor/manual handwheel activation

Handwheel spindle can be extended to ensure that the handwheel can be mounted at a distance of up to 2 m from the actuator (**figure 2.069**).

The use of a Bowden cable is very popular to ensure change-over between motor and manual operation. The length can be selected! And since the handwheel cannot be reached at that height, chainwheel and chain can be useful helpers (**figure 2.070**).

Concealed installation of a pinion and a link chain using a tube is also possible. Then, the handwheel is directly accessible (**figure 2.071**).

The motor-handwheel change-over lever is located below the tube (**figure 2.072**).

A third solution uses a shaft in lieu of a chain which is turned by means of a bevel gearbox. Change-over is made like in the example above (**figure 2.073**).

Figure 2.073: Manual operation via bevel gearbox and shaft

2.1.10 Spindle use

The result of the section dealing with sizing of spindle actuators is that an actuator mounted either in centre or side position can drive two bevel gearboxes. In standard operation, the actuator operates its output shaft toward the flange in clockwise closing direction. When mounting the actuator in a tilted position (**figure 2.074**), direction of rotation is transmitted to the subsequent gearbox (right position in the figure) and consequently to the gearbox output shaft. Simultaneously, a second – typically via output drive type B3/D or D/D – output drive turns toward the left within the actuator (in figure) in counterclockwise direction, seen from the actuator, and consequently also the subsequent gearbox. These different output directions of rotation are compensated since one spindle is equipped with a counterclockwise and the other spindle with a clockwise thread, so that they perform the same rising or lowering movement (e.g. Tr 80 * 10 CCW – counterclockwise, and Tr 80 * 10 CW – clockwise).

Figure 2.074: Actuator simultaneously generates clockwise and counterclockwise revolution at gearbox inputs and outputs

If spindles with identical threads are to be used, the direction of rotation for one spindle would have to be inverted. **Reversing gearboxes** are available for this purpose (**figure 2.075**).

Figure 2.075: Reversing gearbox ahead of a bevel gearbox with spindle mount

For larger two-stage bevel gearboxes, the output direction of rotation can be modified by refitting the gear sets.

Spindles and pertaining loads are moved in two ways.

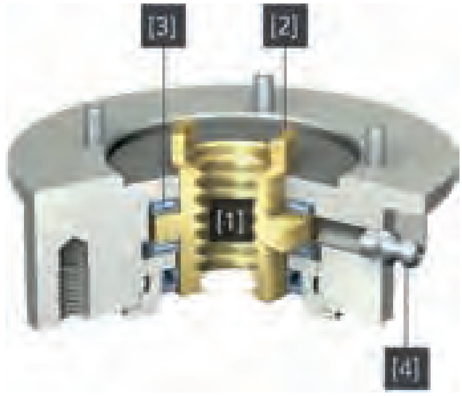
- **Rising spindles:**

Nut and screw belong together, this is the same for the spindle and the **spindle nut** – according to DIN 3210 the **stem nut A**. Generally, the stem nut is made of softer material than the spindle. It is placed below the gearbox and rotated by a journal connection. This causes the **non-rotating spindle** to rise or to descend (**figure 2.076**).

The axial bearings within the output mounting flange are capable of withstanding thrusts. Thus, the housing of the bevel gearbox if free of axial forces. After removing the sealing cover, the spindle can move through a gearbox or through an actuator.

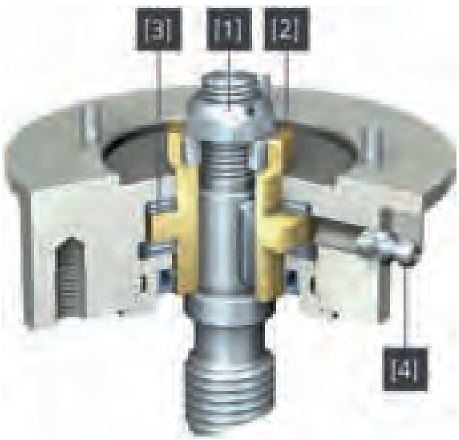
- **Non-rising spindles:**

In turn, the spindle nut can be fixed to the sluice gate, however according to DIN 19569-4:2000-11 – and if permitted by the construction – it should be placed above the mean water level. In this case, the **rotating spindle** is placed within the **spindle mount** (**figure 2.077**) at the gearbox as non-rising spindle. A pinned nut prevents axial displacement.



[1] Stem nut
[2] Dog
[3] Axial needle roller bearing
[4] Grease nipple for bearing lubrication; available with spindle access on request

Figure 2.076: Spindle nut, output drive type A for non-rising spindle



[1] Secured nut
[2] Dog
[3] Axial needle roller bearing
[4] Grease nipple for bearing lubrication

Figure 2.077: Spindle mount, output drive type A for non-rising spindle

Stem nut output drive types A and AK

The **basic output drive type A** of stem nuts is equipped with **axial needle roller bearings**. This is sufficient for the load of infrequently operated sluice gates. For frequently operated locks, roller bearings can be used.

With regard to withstanding loads, **output drive type AK** corresponds to the output drive type A with axial needle roller bearing. However, the AK type is equipped with a **spherical bearing**, allowing the spindle deflection by 1° in all directions. This way, it is capable to optimise the synchronous operation of two spindles in civil engineering constructions for water applications. The spindle mount at the sluice gate may also be gimbal mounted but this is quite complex.

Since the closing element (**figure 2.078**) should be located horizontally, which is however not always possible during gearbox setting, fitting a spindle connecting plate favours appropriate compensation.

Figure 2.078: Spindle connection

Figure 2.079: Strongly worn spindle within a narrow travel range

Spindle lubrication

The bearing of stem nuts A or AK can be supplied with grease by means of grease nipples (refer to figure 2.076). Lubricating the spindles is more complex. Spindle lubrication is extremely important because it helps to reduce the stick-slip effect, the frictional oscillation [10]. Often, they run dry which might cause damage to the spindle nut. When mainly operating within narrow stroke ranges, extreme slim spindles may be subject to material elongations. Within the limit range between normal and extended length, excess torque may occur resulting in switching off and in significant wear at the spindle nut. If it is made of high quality red brass, the weaker steel spindle flanks are being thinned and become razor-sharp (**figure 2.079**). The occurring spindle abrasion is quite evident.

In the past, spindles were always greased with a brush. The working stroke was known. The spindle was lubricated depending on accessibility and time. AUMA manufactures stem nuts prepared for spindle lubrication. A conventional grease cup can be used for easy local lubrication.

Many users apply grease manually to also take the opportunity to check on their installation. However, this might be pleasant but also alarming at times.

Since the current trend is automation, automatic **lubricators** are now quite common. They are mounted if possible at spindle nut level or at least close to the spindle nut (**figure 2.080**). For example, with quantities of 120 cm^3 for draining the lubricant onto the spindle nut in 3,

Figure 2.080: Bevel gearbox with A stem nut and lubricator

6 or 12 months, depending on the setting. A filling level indicator is provided. Unfortunately, once started, the grease flow is permanent. Sufficient drain is provided within the gearbox, even if the installation is only operated twice a year. For this reason, grease collection rings made of plastic or metal are often mounted below the stem nut.

- **Spindle protection**

Spindles are high quality mechanical products. Therefore, they are worth protecting. Rising spindles can protrude as much as 4 m out of the actuator or the gearbox. Then, they are subjected to dust, airborne sand and bird droppings. Since they are lubricated in normal conditions, they are prone to this type of pollution and deposits might ingress into the spindle nut during subsequent spindle descent. The lower spindle area is exposed to hazards by flotsam. Thick branches may get jammed and cause damage to the spindle. Furthermore, water in opencast pits can be particularly aggressive and dissolve the grease. In this case, monthly re-lubrication is required.

Upper spindle protection

For non-rising spindles (**figure 2.081**), the provided through holes are sealed with protective caps. They are made of plastic as standard. Since in particularly cold areas, deformations and freezing of infiltrated melt water occurred, particularly tight aluminium caps fitted with O-rings are available.

Figure 2.081: Non-rising spindle at sluice gate with 3-side seal

The typical spindle protection is implemented with an end-to-end stem protection tube, sealed at the upper end. It is screwed into the hollow shaft thread of the actuator or the bevel gearbox (**figure 2.082**).

Very long and protruding tubes might also be subject to rough mechanical impacts (vandalism) (**figure 2.083**). Supporting them could be quite complex and costly.

Since the problem is known, they are often divided in halves. The lower end is screwed into the gearbox, the upper end with the higher diameter is placed above and screwed to the spindle end. This type is known as **telescopic protection tube** (**figure 2.084**). The spiral protection solutions should also be mentioned (**figure 2.085**).

When lowered, the spiral protection is only visible as small cylinder or depending on the design as cube.

Figure 2.082: Compact stem protection tubes mounted on bevel gearboxes

Figure 2.083: Bent stem protection tube

Figure 2.084: Telescopic protection tubes in Open and Closed positions of sluice gate weirs

Figure 2.085: Risen spindle with extracted spiral protection and compressed bellows

Figure 2.086: Lowered spindle with retracted spiral protection and extended bellows

Downward spindle protection
Both previous figures also show the **bellows**. Once retracted and visible as cylinder as well as once completely descended (**figure 2.086**).

If a lubricated and unprotected spindle enters water with suspended solids, the lubricant combines with the finest sand and suspended solids into a grinding compound causing increased wear. Acid water will also rapidly remove the lubricant from the submerged spindle. These are good reasons to protect the spindle against water. Bellows are made of rubber, plastic or leather. They have hydrophobic properties. However, experience is needed to achieve watertight joints. Furthermore, they can protect the spindle from mechanical impacts.

In turn, bellows do not allow direct visibility of the spindles. For this reason, operators only detected a slight buckling occurred in closing direction because actuator torque seating took place every time this position was approached. If bellows are not considered resistant enough, they can be replaced by steel pipes. An example is shown in (**figure 2.087**).

Figure 2.088 shows a complete set-up. The bevel gearboxes turn the spindles at the spindle mount. The spindles screw into the spindle nuts pulling the four-side sealing sluice gate upward.

Figure 2.087: Spindle nut on a protection tube

Figure 2.088: Spindles protected by tube at a sluice gate in front of a baffle (Elster 2 1 weir)

The tube is welded to the closing element and prevents mechanical impacts. The grease cartridge ensures long-term lubrication. Possible bucklings would occur for non-rising spindles via the spindle nut – in the visible area.

For some applications, customers require non-rising spindles for optical reasons. "They do not interfere with the surrounding scenery".

> ***The many possibilities to house spindles make them safer against accidents then lantern gears with pertaining pinions.***

2.1.11 Sizing lantern gear drives

The sluice gate weir calculated in section 2.1.3 shall not be moved via spindle nuts but using gear wheels, so-called pinions. Two options are possible. The universal term for gear wheel pairs is rolling contact gears. The most known gears are spur and bevel gearboxes. To the limit, diameters of gear wheels in a pair are infinitely large and become a toothed rack or a lantern gear.

According to the requirements, toothed racks and pinions can be precisely manufactured by hobbing which is quite complex and consequently costly. They can be manufactured at lower cost when selecting simple flank profiles and lean structures. To keep the pitch circle diameter and consequently the required manual force as low as possible, pinions with less than 9 teeth can be used – contrary to DIN 19704-2/10.15 – for very small diameters, however reducing the root radius resistance. They are used in manually operated gate leaves for example (**figure 2.089**).

Figure 2.089: Toothed rack and pinion – enclosed – with crank handle

When rarely operated and if low torques are required, these simple versions can also be used for motor operation (**figure 2.090**). For this, the special requirements of DIN 19704 and their reference to DIN 18800 and other specific standards have to be heeded.

In civil engineering constructions for water applications, the pinion/toothed rack pair is nearly always replaced by the pinion/lantern gear pair which in turn is compared to the spindle nut/spindle combination.

Figure 2.090: Toothed racks are operated sideways by the centre mounted actuators via pinions

Lantern gear toothing (**figure 2.091**) can be manufactured using simpler tools. Furthermore, they do not require precision engineering as needed for spindles and they have good usage properties at low speeds. Furthermore, they have an increased breaking strength. Like all force transmission mechanisms, they must be run-in.

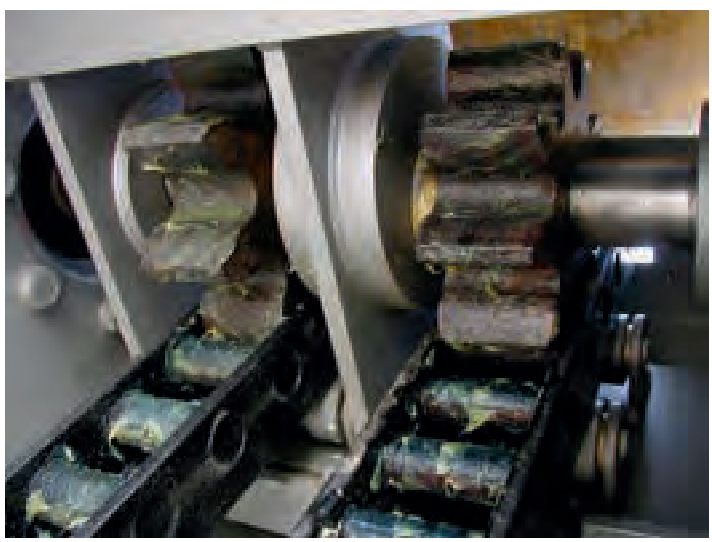


Figure 2.091: Well lubricated lantern gears and pinions with lantern gear toothing (pressure rollers bottom right)

Further advantages and drawbacks of the spindle nut/spindle and pinion/lantern gear combinations will be shown at the end of the chapter.

2.1.11.1 Self-locking and self-braking

Contrary to the spindle nut/spindle pairs, pinion/toothed rack and pinion/lantern gear combinations with their high efficiency (refer to table 2.18) have no static self-locking (usually just called self-locking) and even less dynamic self-locking (self-braking) properties. If these properties are demanded, manufacturers offer the following options:

- **Use of worm gearboxes**

Worm gearboxes (**figure 2.092**) allow for large reduction ratios, up to 150:1. Due to the bend-proof property of the shaft, the ideal value is 50:1. The desired efficiency of single-stage gearboxes is at 40 % when run-in. Under normal operation conditions, single-stage gearboxes are statically and dynamically self-locking.

Movements and vibration may cancel static or dynamic self-locking. A sluice gate may be closed even without driving force.

Figure 2.092: Worm gearbox with actuator

Cases have been known, in which an installation was self-locking in the morning but lost this property in the afternoon when subjected to strong sunlight. In these particular cases, the lower grease consistency favoured the smooth running of actuator and worm gearbox. On cold winter days, the opposite situation occurs.

It is also possible to stop a sluice gate to be lowered at normal water level by means of a stop command. However, if the installation were virtually dry, the hydrostatic force for generating buoyant force and above all friction has nearly disappeared; insufficient self-braking can then be noted.

In turn, smaller measures can restore self-locking or self-braking. If for example the gear ratio is increased within an actuator, which means that the actuator speed is reduced, the pitch angle α is reduced at the same time leading to an improvement of the desired properties.

However, these advices are only suited to remedy occurring problems. It is recommended to provide for safe solutions when selecting the appropriate automation.

- **Support by brake motors**

Typically, electric actuators are equipped with three-phase asynchronous motors. Their design is simple and robust. They achieve favourable torque curves thanks to their compact design.

Figure 2.093: Actuators with brake motors at lock by-passes

They provide a high torque already at standstill and have little overrun.

For reliable self-locking, brake motors are particularly useful. Three-phase AC motors equipped with electro-mechanical brakes are deployed. They are equipped with a brake disc which is electrically bled when starting and replaced with spring force when stopping. The motor winding is not subject to thermal load during the braking action. Thus reliable stop is ensured. The brakes can be bled even without electrical energy to allow manual operation – however without self-locking.

Certain issues have to be considered when using brake motors. Their degree of protection – in standard version – does not exceed IP65. Brake discs can freeze and block. When corroded, their braking reliability is no longer guaranteed.

Brake motors are often implemented at sluice gates for mitre gate locks and by-pass gates (**figure 2.093**).

Using anti-backdrive devices

The brake torque within the anti-backdrive device is generated by a wrap spring. If a torque is applied at the output side, the wrap spring is pressed to the housing wall and prevents a rotary movement due to the friction. If the input shaft is moved, the spring diameter reduces and allows rotary movement (**figure 2.094**).

[1] Input shaft
[2] Wrap spring
[3] Output shaft

Figure 2.094: Anti-backdrive device

By using anti-backdrive devices with an efficiency $\eta > 0.9$, seemingly contrary principles can be combined: self-locking and high efficiency for actuators and gearboxes. Even non self-locking actuators with output speeds of 125 rpm or 180 rpm and non self-locking multi-turn gearboxes can be combined. The financial comparison between a combination of anti-backdrive device with non self-locking elements and actuator/worm gearbox can be quite interesting. However, the first combination does not achieve the high working torques of worm gearboxes.

Anti-backdrive devices can be used for both motor and manual operation. It is mounted to the gearbox input so it does not have to be operated with the comparatively high output torques. The actuator is placed on the input shaft of the anti-backdrive device (**figure 2.095**).

Figure 2.095: Anti-backdrive device between actuator and bevel gearbox

2.1.11.2 Disassembly of self-locking elements

When disassembling power train elements, the installations must be blocked at a suitable position to avoid undesirable movement of the connecting elements and consequently the closing element. Basically, the closing elements must be mechanically locked at the required position to facilitate assembly and maintenance work and to avoid any potential hazard. Lugs or similar equipment must already be provided at the time of construction (DIN 19704-2/3.4).

2.1.11.3 Particularities of worm gearboxes in civil engineering constructions for water applications

In the valve industry, worm gearboxes are also called part-turn gearboxes since they are used for automating shutters and ball valves with 90° movements. Lifetime tables are provided.

When exceeding a complete gearbox revolution, the first tooth and consequently further teeth are subjected to load for a second and repeated time. Temperature rise and increased wear are the consequence. This must be compensated by torque reduction to maximum 50 %.

If the initial swing angle exceeds 360°, multi-turn gearboxes meaning gearboxes without 90° end stops must be used. The "D" is the AUMA designation for worm gearboxes in multi-turn version (GSD).

To increase the transmission performance, bronze worm wheels are used in civil engineering constructions for water applications.

In combination with the suitable material of the worm shaft, bronze worm wheels have superior sliding properties and efficiency. They have a better thermal conductivity and are less subject to wear. They have a longer lifetime.

Table 2.15 is an extract of the Technical Data sheet for worm gearboxes.

Tab. 2.15: Technical data for GSD worm gearboxes – Lifetime run torques; version with worm wheel made of bronze (extract)

Type	Output torque [Nm]	Reduction ratio	Input torque[1] [Nm]	Factor[2]	Lifetime[3] run torques [Nm]	Lifetime[4] no. of possible output revolutions[5]
GSD 200.3 with red.gear.[6] GZ 200.3	16,000 i = 4	i = 53 i_{tot} = 214	718 197	f = 22.3 f = 81.3	8,000	15,000
GSD 250.3 with red.gear. GZ 250.3	32,000 i = 4	i = 53 i_{tot} = 210	1,462 401	f = 21.9 f = 80.0	16,000	10,000
GSD 315 with red.gear. GZ 30.1	63,000 i = 8	i = 53 i_{tot} = 424	2,423 354	f = 26.0 f = 178.0	31,500	4,700

1) Input torque at maximum output torque
2) Factor f = ratio between output torque and input torque
The following applies: $f = i * \eta$
Whereby f = factor, i = reduction ratio, η = efficiency
3) Lifetime run torques for GSD worm gearboxes are reduced to 50 %
4) For higher lifetime requirements, the lifetime run torques are to be reduced by the lifetime formula
$L = (T_{GS} / T_{GSø})^3$ (2.1.5.6.2 Lifetime proof).
5) Maximum 10 output drive revolutions may be operated without pause. This prevents overheating of grease which could lead to leakage (special grease on request).
6) A GZ primary reduction gearing can be assigned to the GSD worm gearbox, with reduction ratios of 4:1, 8:1 or 16:1.

The breaking torque of these gearboxes is typically double the maximum output torque. Damage or plastic deformation by maintaining large forces can however occur for lower values.

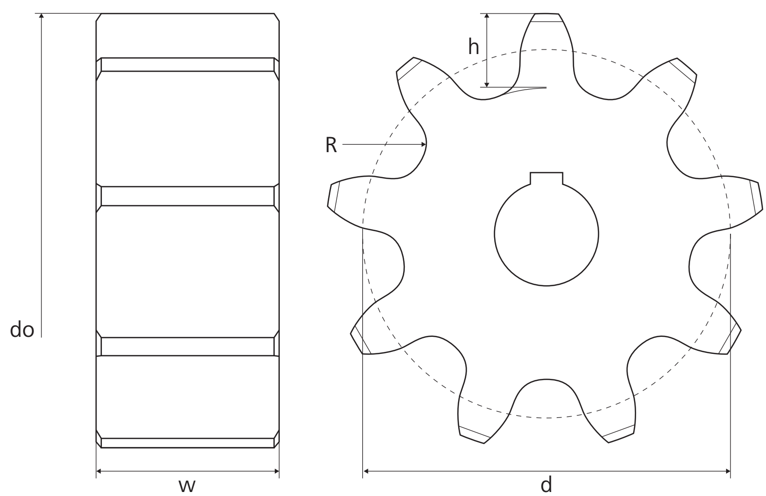
2.1.11.4 Calculating required torques in opening and closing directions

The required output torque is calculated as follows:

$$T = F * \frac{d}{2}$$

wherein
T = Torque
F = Force
d = Pinion diameter

A pinion is characterised by the variables as shown in **figure 2.096**.



do = Outside/nom. addendum
d = Pitch circle diameter
w = Width of teeth
h = Height of teeth
R = Root diameter

Figure 2.096: Parameters of a lantern gear pinion

According to DIN 19704-2/10.15, the number of teeth is at least 9 to ensure safe transmission but also to prevent too small and weak designs. The higher the number of pinion teeth, the smoother the operation.

The pitch circle diameter d can be calculated by the module.

$$m = \frac{d}{z_1}$$

2.1.12 Sizing lantern gear drives – example calculation

The procedure is the same as for sizing spindle actuators (2.1.5).

Given:
- Resulting forces (2.1.3.11 Closing element positions)
- Decisive water levels
- Number and length of sluice gate movement for assigned water levels
- Number of potential subsequent strokes: minimum 2 strokes
- Operating speed $v \approx (200 \ldots 400)$ mm/min

Wanted:
- For gearbox sizing: Forces and torques at gearbox outputs for unfavourable load distribution ($F_{OCE60\,\%}$ / $F_{CCE60\,\%}$, $T_{OCE60\,\%}$ / $T_{CCE60\,\%}$)
- For actuator sizing: Forces and torques at gearbox inputs or at actuator outputs (T_{AO} / T_{AC})

To be determined
- Running times per stroke [min]
- Number of possible subsequent strokes
- Operating speed

Proof to be shown
- Type of duty
- Lifetime
- Self-locking

To be considered
- One-sided stop
- Exceptional impacts of actuators in case of an accident
- Buckling resistance

To be defined
- Gearbox parameters
- Actuator parameters

2.1.12.1 Calculation of the opening and closing forces

In this example, lantern gear and spindle weight are equalised. Taken from the force table:

Opening F_O = 100 kN for EHW or F_{OCE} = 50 kN
F_{OCE} = Opening force per connecting element

Closing F_C = 50 kN for EHW or F_{CCE} = 25 kN
F_{CCE} = Closing force per connecting element

Opening and closing force per connecting element for unfavourable load distribution (2.1.5.1) shall be adopted for lantern gears. For 60 % load

$F_{OCE60\%}$ = 60 kN
$T_{OCE60\%}$ = Opening force for unbalanced load per connecting element

Both gearboxes must be selected.

$F_{CCE60\%}$ = 30 kN.
$T_{CCE60\%}$ = Opening force for unbalanced load per connecting element

The lower closing force is insignificant for the gearbox selection in our example.

2.1.12.2 Determination of the output torque in opening direction

- **Stress proof**

The statements made for spindle stress proof (2.1.5.2) are generally also applicable for lantern gears.

- **Pinion and lantern gear**

Compared to existing installations, the following pinion was selected.

Pinion:
Number of teeth $z_1 = 9$
Width of teeth $w = 90$ mm
Module $m = 30$ mm
The pitch circle diameter d is calculated using:

$$m = \frac{d}{z_1}$$

$$\begin{aligned} d &= z_1 * m \\ &= 9 * 30 \text{ mm} \\ d &= 270 \text{ mm} \end{aligned}$$

After having completed the prescribed proofs, the dimensions must be modified, if required.

Provisionally, the lantern gears shall have the following dimensions:

Lantern gear:
2 pieces flat steel: l = 3000 mm, h = 110 mm, w = 20 mm,
Distance of flat steel bars = 98 mm
Bolt diameter d_B = 50 mm

The later verification of buckling safety (DIN 18800-2) shall decide on details like material selection and possibly required reinforcements by a lantern gear rear to be welded. The calculated bolts are welded at identical distance in two flat steel bars or secured as appropriate.

Opening torque for unfavourable load distribution

If regular lantern gear lubrication can be guaranteed, good efficiency can be achieved η_{LG} = 0.9. Since the often mentioned insensitivity of pinion/lantern gear pairs to lack of lubrication might result in scarce lubrication, the mean efficiency η_{LG} = 0. 85 shall be used. This results in:

$$T_{OCE60\%} = \frac{F_{OCE60\%} * \frac{d}{2}}{\eta_{LG} * \eta_{PSB}}$$

$$= \frac{60\text{ kN} * \frac{0.27\text{ m}}{2}}{0.85 * 0.97}$$

$$T_{OCE60\%} = 9{,}824\text{ Nm}$$

wherein
η_{LG} = Efficiency lantern gear
η_{PSB} = Efficiency pinion shaft bearing

2.1.12.3 Gearbox selection

Using the opening torque $T_{OCEG60\%}$ = 9.824 Nm the maximum torque of 32,000 Nm for the GS 250.3 (refer to table 2.15) is used to the capacity of 30 %. Preliminary selection of GS 250.3 is made.

2.1.12.4 Proof for actuator opening and closing torques

Since with the driving torque determination a possible unfavourable load distribution is not considered, the required opening torque results from the opening force F_O = 100 kN:

$$T_O = \frac{F_O * \frac{d}{2}}{\eta_{LG} * \eta_{PSB}}$$

$$= \frac{100\ kN * \frac{0.27\ m}{2}}{0.85 * 0.97}$$

$$T_O = 16{,}374\ Nm$$

The actuator opening torque T_{AO} with f1 = 21. 9 (table 2.15) results in:

$$T_{AO} = \frac{T_O}{f1}$$

$$= \frac{16{,}374\ Nm}{21.9}$$

$$T_{AO} = 748\ Nm$$

At T_{AO} = 748 Nm, the actuator torque is relatively high. Based on experience, the operating speed resulting from the reduction ratio would be far too high. A compromise must be found. One option is to provide a preliminary reduction gearing to the worm gearbox – a spur gear stage. GS 250.3 will be equipped with GZ 250.3 at i = 4:1. So the total reduction ratio GS/GZ is: i_{tot} = 210:1 with factor f2 = 80.0. This results in:

$$T_{AO} = \frac{T_O}{f2}$$

$$= \frac{16{,}374\ Nm}{80}$$

$$T_{AO} = 205\ Nm$$

According to DIN 19704-1/8.4, the minimum setting shall be:

$$T_{AO+25\%} = 1.25 * T_{AO}$$

$$= 1.25 * 205\ Nm$$

$$T_{AO+25\%} = 256\ Nm$$

SA 14.6 is appropriate for the setting torque. In duty type S2 - 15 min, an adjustment reserve of up to 500 Nm and in duty S2 - 30 min up to 360 Nm are available. The required proofs still have to be provided.

The actuator closing torque is calculated with:

F_C = 50 kN

and

$$T_C = \frac{F_C * \frac{d}{2}}{\eta_{LG} * \eta_{PSB}}$$

$$= \frac{50\ kN * \frac{0.27\ m}{2}}{0.85 * 0.97}$$

$$T_C = 8{,}187\ Nm$$

to:

$$T_{AC} = \frac{T_C}{f2}$$

with f_2 = ratio between output torque and input torque

$$= \frac{8{,}187\ Nm}{80}$$

$$T_{AC} = 102\ Nm$$

or

$$T_{AC+25\%} = 1.25 \cdot\cdot T_{AC}$$
$$= 1.25 \cdot\cdot 102\ Nm$$
$$T_{AC+25\%} = 128\ Nm$$

The actuator torque switch in closing direction must be set at least to $T_{AC+25\%}$ = 128 Nm. Since the standard torque range of the selected SA 14.6 is relatively high with 200 to 500 Nm, there is the possibility to use the reduced torque range of 100 to 250 Nm. The verification of buckling safety of the selected lantern gear has to be made using the respectively increased closing force F_C.

2.1.12.5 Determination of the number of subsequently operable strokes and total actuator revolutions

■ **Determination of the number of subsequently operable strokes**

The following was defined:

GS 250.3/GZ 250.3, i_{GSRG} = 210 : 1

i_{GSRG} = Reduction ratio worm gearbox/primary reduction gearing

The actuator speed of 45 rpm results for this combination.

- **Running time:**

$$t = \frac{s * i_{GSRG}}{n * \pi * d}$$

$$= \frac{3{,}000\ \text{mm} * 210}{45\ 1/\text{min} * \pi * 270\ \text{mm}}$$

$$t = 16.4\ \text{min}$$

wherein

t = Running time/stroke
s = Travel of one stroke
i_{GSRG} = Reduction ratio worm gearbox/primary reduction gearing
n = Actuator output speed
d = Pinion diameter

- **Operating speed:**

$$v = \frac{n * \pi * d}{i_{GSRG}}$$

$$= \frac{45\ 1/\text{min} * \pi * 270\ \text{mm}}{210}$$

$$v = 182\ \text{mm} / \text{min}$$

Furthermore, the following is needed:

- **Worm gearbox revolutions per stroke:**

$$R_{GS}\ /\ \text{stroke} = \frac{s}{\pi * z_1 * m}$$

$$= \frac{3{,}000\ \text{mm}}{\pi * 9 * 30\ \text{mm}}$$

$$R_{GS}\ /\ \text{stroke} = 3.5$$

wherein

z_1 = Number of teeth
m = Module

- **Actuator revolutions per stroke:**

$$R_A/\text{stroke} = R_{GS}/\text{stroke} * i_{GSRG}$$

$$= 3.5 * 210$$

$$R_A/\text{stroke} = 735$$

Since with n = 45 rpm, the condition of minimum to subsequent stroke operations cannot be fulfilled for both standard types of duty S2 - 15 min and S2 - 30 min, calculation shall be made using the actuator speeds n = 90 rpm and n = 180 rpm (**table 2.16**).

Tab. 2.16: Number of possible strokes

n [rpm]	Run time/ stroke [min]	Strokes at S2 - 15 min	Strokes at S2 - 30 min	Operating speed [m/min]
45	16.4	0.9	1.8	0.181
90	8.2	1.8	3.6	0.363
180	4.15	3.6	7.2	0.726

In type of duty S2 - 30 min, the actuator can subsequently operate 3.6 strokes at 90 rpm. The calculated operating speed v = 363 mm/min is acceptable. If lower, the output speed 63 rpm can be used.

- **Determination of the total actuator revolutions**

As mentioned in section (2.1.5.5), the number of 60 strokes per year was defined. Consequently, the total figure of actuator hollow shaft revolutions results from:

$$R_{A\text{-}35y} = R_A/\text{stroke} * \text{strokes/year} * t_{tot}$$
$$= 735\ \text{R/stroke} \cdot 60\ \text{strokes/year} \cdot 35\ \text{years}$$
$$R_{A\text{-}35y} = 1.543\text{m}$$

According to table 2.04, for SA 14.6 with

$$R_{A\text{-}35y} < Y_{AHSS2}$$
$$1.543\text{m} < 4.0\text{m}$$

the condition for hydraulic steel structure applications HSS2 is fulfilled. The lifetime run torques are to be considered accordingly.

2.1.12.6 Duty type proof of actuator

Identical to the cycle considered with spindle nut/spindle (2.1.5.6), for the cycle using pinion/lantern gear by e.g. worm gearboxes, assumption has to be made of comparable self-locking or driving power for the lowering movement difficult to determine.

With the average force at EHW

$$F_{\varnothing EHW} = 57.5\ \text{kN}$$

the following torque results:

$$T_{\varnothing EHW} = \frac{F_{\varnothing EHW} * \frac{d}{2}}{\eta_{LG} * \eta_{PSB}}$$

$$= \frac{57.5\ \text{kN} * \frac{0.27\ \text{m}}{2}}{0.85 * 0.97}$$

$$T_{\varnothing EHW} = 9{,}414\ \text{Nm}$$

The actuator opening torque with f2 = 80.0 (table 2.15) results in:

$$T_{A\text{-}\varnothing EHW} = \frac{T_O}{f2}$$

$$= \frac{9{,}414\ \text{Nm}}{80.0}$$

$$T_{A\text{-}\varnothing EHW} = 118\ \text{Nm}$$

Using

$T_{A\text{-}\varnothing EHW} < T_{A\text{-}S2\text{-}30\ min}$
118 Nm < 125 Nm

the fitness for purpose has been proved.

2.1.12.7 Lifetime proof for actuator

Lifetime or proof of operational stability must be provided on the basis of the equivalent force for water levels to be considered (2.1.4.3.3). For the hydrostatic pressure triangle MHW and MW the equivalent forces were determined in section (5.1.5.6.2). The average torques result from:

$$T_{O\text{-}\varnothing MHW} = \frac{F_{\varnothing MHW} * \frac{d}{2}}{\eta_{LG} * \eta_{PSB}}$$

$$= \frac{63\ \text{kN} * \frac{0.27\ \text{m}}{2}}{0.85 * 0.97}$$

$$T_{O\text{-}\varnothing MHW} = 10{,}315\ \text{Nm}$$

or

$$T_{O\text{-}\varnothing MW} = \frac{F_{\varnothing MW} * \frac{d}{2}}{\eta_{LG} * \eta_{PSB}}$$

$$= \frac{54\ \text{kN} * \frac{0.27\ \text{m}}{2}}{0.85 * 0.97}$$

$$T_{O\text{-}\varnothing MW} = 8{,}842\ \text{Nm}$$

The actuator torques result in:

$$T_{A\text{-}\varnothing MHW} = \frac{T_{O\text{-}\varnothing MHW}}{f2}$$

$$= \frac{10{,}315\ \text{Nm}}{80.0}$$

$$T_{A\text{-}\varnothing MHW} = 129\ \text{Nm}$$

or

$$T_{A\text{ - }\varnothing MW} = \frac{T_{O\text{ - }\varnothing MW}}{f2}$$

$$= \frac{8{,}842\ \text{Nm}}{80.0}$$

$$T_{A\text{ - }\varnothing MW} = 110\ \text{Nm}$$

According to the Palmgren Miner method, the following mean value was calculated with the known cycles:

$$T_{A\text{ - }\varnothing} = \sum_{i=1}^{k} \frac{n_i * T_{Ai}}{N_i}$$

$$T_{A\text{ - }\varnothing MHW/MW} = \frac{n_1 * T_{A\text{ - }MHW} + n_2 * T_{A\text{ - }MW}}{N_1 + N_2}$$

$$= \frac{6 * 129\ \text{Nm} + 24 * 110\ \text{Nm}}{6 + 24}$$

$T_{A\text{-}ØMHW/MW}$ = 114 Nm

$T_{A\text{-}ØMHW/MW}$ = mean actuator torque at MHW/MW

With

$T_{A\text{-}ØMHW/MH} < T_{A\text{-}HSS2}$
114 Nm < 135 Nm.

the conditions for the lifetime proof (refer to table 2.06) are fulfilled. If

$T_{A\text{-}ØMHW/MH} > T_{A\text{-}HSS2}$,

and using

$$L = \left(\frac{C}{P}\right)^p$$

$$L = \left(\frac{T_{A\text{ - }HSS2}}{T_{A\text{ - }\varnothing MHW/MW}}\right)^p$$

wherein
L = lifetime factor
C corresponds to run torque according to table
P corresponds to the calculated run torque
Exponent p = 3 for ball bearings or gearings

the number of actuator hollow shaft revolutions can be reduced according to table. The nominal number of hollow shaft revolutions results from:

$U_{A\text{-nominal}} = L * R_{A\text{-HSS2}}$

- Lifetime test of worm gearboxes

Possible gearbox revolutions must comply with the actuator revolutions specified for the respective application.

The GS 250.3/GZ 250.3 worm gearbox must turn

$$R_{GS\text{ - }35y} = \frac{R_{A35y}}{i_{GS}}$$
$$= \frac{1.543m}{210}$$
$$R_{GS\text{ - }35y} = 7{,}350$$

7,350 times. 10,000 revolutions are permitted with a run torque reduced to 50 %. Consequently, no higher nominal gearbox lifetime must be calculated.

2.1.12.8 Proof of self-locking

For the present case and according to the tender, lifting and lowering of the closing elements is to be executed by means of lantern gears. A combination of an actuator and two worm gearboxes is the ideal solution. Pinion/lantern gear connections with an efficiency of η = 0.9 are not self-locking. However, worm gearboxes with an efficiency of η = 0.4 in combination with the selected SA 14.6 and a speed of n = 90 rpm offer sufficient self-locking properties (2.1.11.1).

2.1.12.9 Evaluation of example calculation

The following torques apply for an actuator SA 14.6 in version 90 rpm – S2 - 30 min – in application HSS2:

Permissible torque
$T_{Amax.\ S2\text{-}30min}$ 360 Nm

Opening torques
$T_{A\text{-}OEHW}$ 205 Nm
$T_{A\text{-}OEHW+25\ \%}$ 256 Nm

Mean torque for a double stroke at EHW for type of duty
$T_{A\text{-}ØEHW}$ 118 Nm

Mean torque for a double stroke at MHW/MW, for lifetime

$T_{A\text{-}\emptyset MHW/MW}$ 114 Nm

Closing torques

$T_{A\text{-}CEHW}$ 102 Nm

$T_{A\text{-}CEHW+25\%}$ 128 Nm

- **Starting torque**

With a speed of 90 rpm in type of duty S2 - 30 min, the SA 14.6 has a maximum torque of 360 Nm. Consequently, the opening torque required for a new installation of T_{AO} = 205 Nm and the supplement of 25 % as demanded by DIN 19704-1/8.4 as well as a possibly even higher torque increase by ageing and unforeseeable incidents are met. The lower closing torque is not considered here.

- **Duty type run torque**

SA 14.6 in type of duty S2-30 min for a mean torque of $T_{A\text{-}\emptyset EHW}$ = 118 Nm is suited for EHW extreme high water conditions.

- **Lifetime run torque**

The long-term running properties of the actuator allow for minimum 4.0m output drive revolutions during more than 35 years with an average of $T_{A\text{-}\emptyset MHW/MW}$ = 114 Nm in HSS2 applications. 1.543m revolutions are required. This corresponds to the lifetime of the selected worm gearbox.

- **Self-locking**

The calculated actuator combination with the selected actuator and the two worm gearboxes is self-locking.

- **Number of strokes**

The actuators can operate 3.6 subsequent strokes at 90 rpm. In case of excessive operating speed, operation can be made using a speed of n = 63 rpm.

- **Permissible operating speed**

According to DIN 19704-2/9.2.2, the speed of the closing element should not exceed 0.1 through 1.0 m/min when reaching the end position. The tender specified the operating speed at v ≈ (200 … 400) mm/min. The calculated operating speed v = 363 mm/min is permissible.

2.1.13 Proofs for exceptional loads

The basics were discussed in section (2.1.6).

2.1.13.1 One-sided stop for two-sided operated closing elements

The combination of pinion/lantern gear is not self-locking. Consequently, the following worm gearbox must be capable of holding the complete closing element in case of an accident.

$T_O = 16{,}374$ Nm.

The selected GS 250.3/GZ 250.3 has a breaking torque T_B at double the output torque of 32,000 Nm. With

$T_B > T_O$

64,000 Nm > 16,374 Nm

it is suitable.

2.1.13.2 Exceptional impacts of actuators in case of an accident

The same applies as mentioned in (2.1.6.2). Buckling safety will be discussed in (2.1.15).

2.1.14 Examples of actuator gearbox combination arrangements

When subjected to low loads, it might be sufficient to turn two pinions using a manually operated worm gearbox (**figure 2.097**). In this case, a hole is drilled into the gearbox cover and a seal is fitted.

Figure 2.097: Centre mounted worm gearboxes turn two pinions each

Wooden gates are mostly made of hard wood like oak trees (**figure 2.098**).

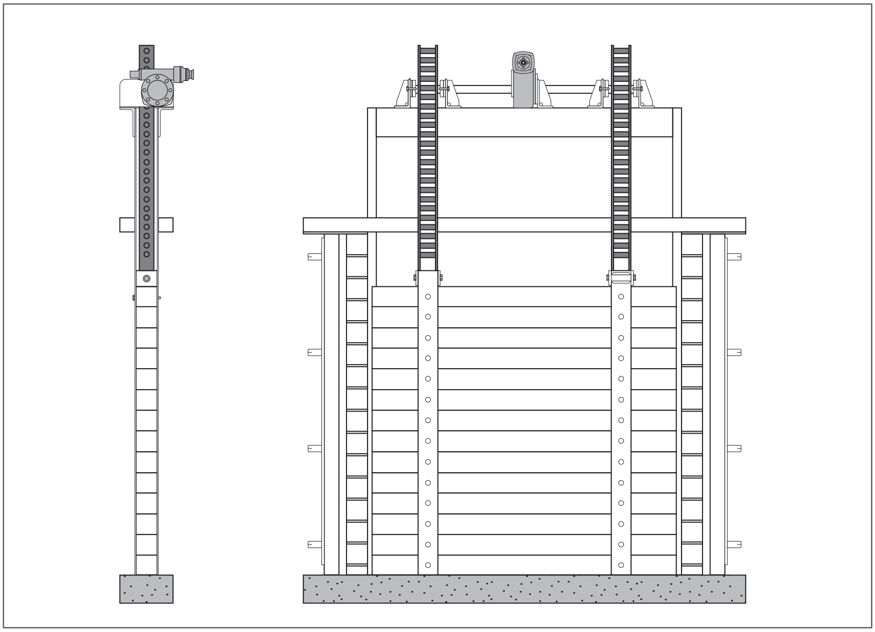


Figure 2.098: Pinions engage into lantern gears without additional gearboxes

Counter rollers or guiding equipment ensure safe pinion/lantern gear connection (**figure 2.099**).

Figure 2.099: Secured connection thanks to a guide plate

Figure 2.100: Manually operated sluice gate weir

Figure 2.101: Side mount of worm gearbox with handwheel. Pinion shaft diagonally to the flow direction

Gears and pinions are mounted into the weir and protected with suitable enclosures against unauthorised interference (**figure 2.100**).

The manual drive can be provided as middle mount and as lateral mount (**figure 2.101**). The set-up with the pinion shaft diagonally to the flow direction acts most gently on pinion and lantern gears when subjected to changing hydrostatic pressure levels.

Larger torques require provision of gearboxes at pinions (**figure 2.102**).

Figure 2.102: One actuator operates two worm gearboxes

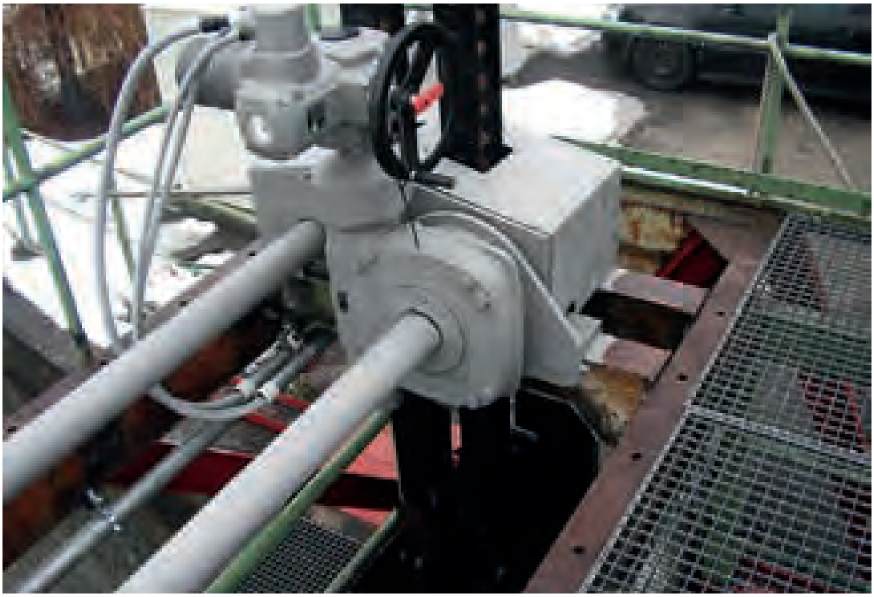


Figure 2.103: One worm gearbox operates two pinions

Both pinions can also be driven by an actuator/gearbox combination (**figure 2.103**). The pinion shaft is located in a favourable position diagonally to the flow direction.

When using two gearboxes, pinion shafts are often located in parallel to the flow direction which might lead to lantern gear displacement right through to lateral pinion abrasion due to water pressure variations (**figure 2.104**).

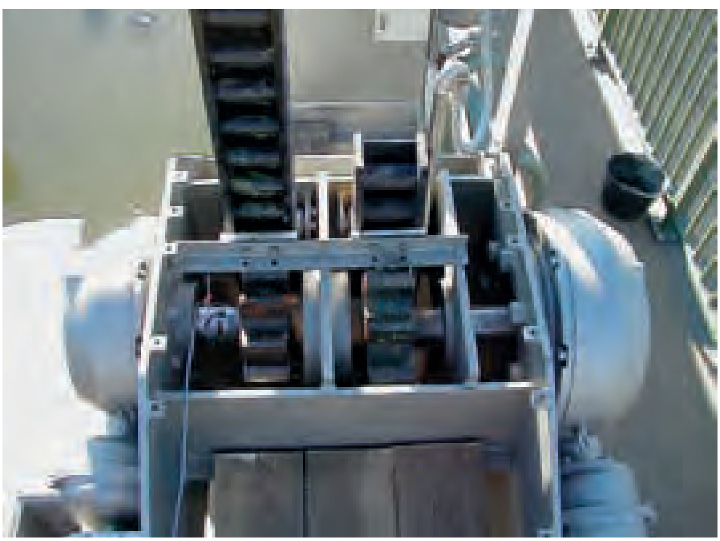


Figure 2.104: Pinion shafts in parallel to flow direction

A further variant would be the external mount of an actuator with worm gearbox (**figure 2.105**). The reason could be space constraints or the requirement of easy access. The set-up is favoured by the gentle pinion shaft arrangement diagonally to the flow direction. Asymmetry bears the risk that synchronous operation of outer and inner pinion is not ensured due to torsion. Furthermore, it must be considered that the simple load has to be transmitted to the outer pinion, but the double load to the inner pinion.

Figure 2.105: Side mount of actuator and worm gearbox for two pinions

2.1.15 Buckling safety

Like for spindles, the buckling safety has to be proved for lantern gears. In particular, this means dealing with the assembled compression members, in this instance with **double-span built up battened members** (DIN 18800-2). Undersized lantern gears risk to buckle when pressing against obstacles or when the gate leaf is seated onto the bed beam. In particular, in the lower area close to the gate suspension, they are particular sensitive, although according to Euler, this part is less at risk. However, stability is insufficient if no bolts are welded into the sides. Consequently, the previously used Euler proof is no longer state of the art.

According to DIN 18800-2 (issue Nov 1990), the requirement must be fulfilled for proof of structural safety:

$$\frac{\text{load}}{\text{load capacity}} \leq 1$$

- **Load**

According to DIN 19704-1/8.4, the load amounts to:

$$N_K = 1.25 * N$$
$$= 1.25 * 50 \text{ kN}$$
$$N_K = 64 \text{ kN}$$

wherein

N_K = Characteristic value
N = Normal force = F_C (2.1.12.1 example calculation)

- **Load capacity**

With
Length of buckling S_K = 300 cm
Used material e.g. S 355 (previously St 52)
E = Elasticity module = 21,000 kN/cm^2
f_{yc} = Yield strength = 36.0 kN/cm^2

the load capacity as mentioned in 2.1.8.2 with the respective particularities like the second degree area moment is to be calculated for a lantern gear:

$$I = I_{2Rectangles} = \frac{2 * w^3 * h}{12}$$

wherein
I = Area moment of second degree
w = Width of the lantern gear side
h = Height of lantern gear side

The following applies again:
γ_F = Partial safety factor for load = 1.35
γ_M = Partial safety factor for load capacity = 1.5

The result of the lantern gear proof shows whether the connecting elements are sufficiently sized:

$$\frac{N_d}{\kappa * N_{pl,d}} < 1$$

wherein

N_K = Characteristic value of N N_K = 64 kN

N_d = Rated value of N; $N_d = \gamma_F \cdot N_K$

κ = Attenuation factor

$N_{pl,d}$ = Normal force in plastic condition;

$$N_{pl,d} = A * \frac{f_{YK}}{\gamma_M}$$

$$\frac{\gamma_F * N_K}{\kappa * A * \frac{f_{YK}}{\gamma_M}} < 1$$

Should the result be insufficient, further detailed proofs depending on the design of the lantern gear are required in compliance with DIN 18800: For example, the fraction of the bolts can be considered. In marginal cases, a welded shoulder – to be included into the calculation – provides additional safety.

Beyond the buckling safety, the proof shall be conducted relating to the lantern gear transmission for at least one of the driving parts:

- Pinion
  - pinion shaft
  - parallel key connection pinion – pinion shaft
  - pinion shaft bearing

- Lantern gear
  - bolts
  - connecting bolts
  - eye plate
  - lantern gear counter guidance

Conducting all these additional proofs cannot and shall not be part of this browsing through actuator technology within civil engineering constructions for water applications.

2.1.16 Manual operation

The required manual force F_{Ha} can be calculated as follows:

$$F_{Ha} = \frac{T_{AO}}{i_{Ha} * \frac{d_{Ha}}{2}}$$

$$= \frac{205\ \text{Nm}}{8 * \frac{0.4\ \text{m}}{2}}$$

$$F_{Ha} = 128\ \text{N}$$

wherein
i_{Ha} = Handwheel reduction ratio (refer to table 2.14 "Manual forces")
d_{Ha} = Handwheel diameter (refer to table 2.14 "Manual forces")

It is slightly above the maximum value of 100 N specified in DIN 19704-1/8.3, however, can be further reduced – as shown for spindle actuators (2.1.9 "Manual operation").

2.1.17 Lantern gear application

All connecting elements must move into the same direction. If the hollow shaft of the actuator or the central gearbox (**figure 2.106**) turn clockwise to the flange side, it turns counterclockwise in opposite direction. This must be compensated by respective gearbox version.

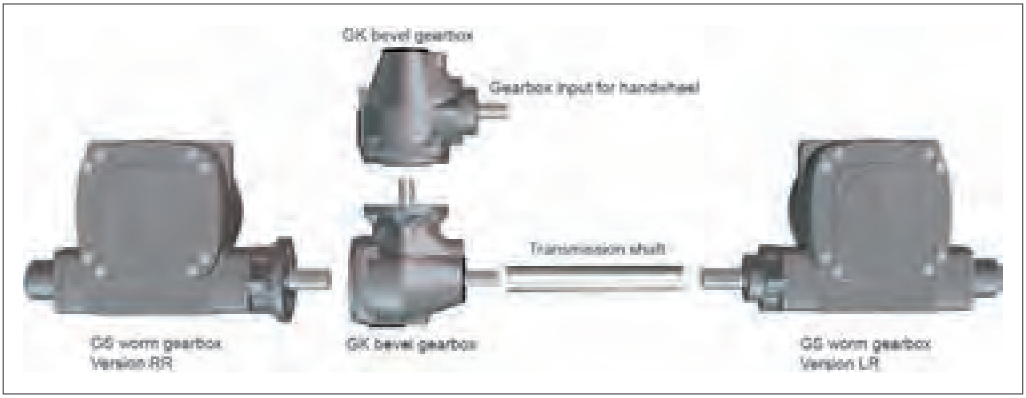


Figure 2.106: Gearbox pair for a weir with lantern gears

Worm gearbox versions are characterised by the letters R (right – clockwise) and L (left – counterclockwise). The first letter in the designation indicates the position of the input shaft – while input is upward – with regard to the worm wheel, the second letter the direction of rotation of the gearbox output for clockwise rotation at the gearbox input. Consequently, RR describes an input shaft located to the right of the worm wheel and clockwise output direction.

When implementing these combined pairs in practical applications, the following can be found. An actuator takes the place of the bevel gearbox. Thanks to the modified arrangement, the shaft is located upwards (**figure 2.107**).

Alternatively, the LR worm gearbox pair at actuator output drive can be used as RR on the opposite side (**figure 2.108**). By shifting, the shaft can be located downwards.

Illustrated LL – RL corresponds to counterclockwise gearbox outputs.

Figure 2.107: Worm gearbox pair: RR at actuator output and LR at the opposite side with shaft located above

Figure 2.108: Worm gearbox pair: LR at actuator output and RR at the opposite side

When comparing installations with lantern gears and those with spindles, the lantern gear actuators are virtually always located sideways and the spindle actuators are often centred. This can be explained as follows: The sideway actuator creates a torsion across the complete shaft to the opposite side. Lantern gears can generally compensate angle errors by their larger clearance with regard to the pinion. This is completely different for precisely manufactured spindles. For spindle applications, a centred arrangement is advised for compensation, unless the gate width is low or the shaft used sufficiently resistant to torsions. The sideways mounted actuators might also take a middle position between the worm gearboxes.

Should the worm gearboxes be replaced by smoothly running bevel gearboxes, self-locking anti-backdrive devices have to be integrated (**figure 2.109**).

Figure 2.109: Centre mount actuator/anti-backdrive device/spur gearbox for pinion-lantern gear operation

2.1.18 Spindle and lantern gear version

Which are the significant differences?

2.1.18.1 Results from spindle and lantern gear configuration

The qualities of spindle versus lantern gear version of our example are shown hereafter. The weir is loaded at average with the number of voltage cycles in compliance with the product requirement specifications. This corresponds to HSS2 application. For both applications, SA 14.6 actuators were selected. When respecting the corresponding run torques, this results in minimum 4.0m output drive shaft revolutions. Gearboxes shall comply with the required 3.528m or 1.543m. The results are shown in **table 2.17**.

Tab.2.17: Possible strokes with the selected actuator/gearbox combinations

	Spindle actuators	Lantern gear actuators
$U_{Gearb.}$/stroke	300	3.5
U_A/stroke	1,680	735
Strokes 35 y	2,100	2,100
$U_{A\text{-}35Y}$	3,528,000	1,543,500
$U_{A\text{-}HSS2}$	4,000,000	4,000,000
Required actuator revolutions in 35 years $U_{A\text{-}35y} \leq U_{A\text{-}HSS2}$		
$U_{Gearb.\text{-}HSS}$	600,000	10,000
$U_{Gearb.\text{-}35y}$	630,000	7,350
$U_{Gearb.\text{-}HSSnominal}$	1,500,000	10,000, since $U_{Gearb.\text{-}HSS} > U_{Gear.\text{-}35y}$

$T_{Gearb.}$/stroke = Gearbox revolutions per stroke
T_A/stroke = Actuator revolutions per stroke
Strokes/35y = Strokes in 35 years
$T_{A\text{-}35y}$ = Actuator revolutions in 35 years
$T_{A\text{-}HSS2}$ = Actuator revolutions for HSS2
$T_{Gearb.\text{-}HSS}$ = Gearbox revolutions for HSS
$T_{Gearb..\text{-}35y}$ = Gearbox revolutions in 35 years
$T_{Gearb.\text{-}HSSnominal}$ = Gearbox revolutions for HSS according to load

The nominal gearbox revolutions $R_{Gear\text{-}HSSnominal} > T_{Gear-35y}$ are:

Bevel gearboxes 1,500.000 > 630,000
Worm gearboxes 10,000 > 7,350

The lantern gear structure can operate the same number of strokes as the spindle structure while requiring half the number of actuator hollow shaft revolutions.

Striving for higher load capacities is quite evident. However, when using a smaller actuator, the calculated torques cannot be applied. The lower number of actuator hollow shaft revolutions by worm gearboxes can be an advantage for higher load requirements in HSS3.

The following results can be deducted: Worm gearboxes are better suited for large strokes, bevel gearboxes are more advantageous for partial strokes, considering that the running time per stroke for worm gearboxes amounts to 8.2 min., for bevel gearboxes to 13.4 min.

The question still needs to be answered whether it is more beneficial to work with lantern gear drives. The following point might be a useful contribution to the decision between spindles or lantern gears.

2.1.18.2 Comparison of essential differences with connecting elements

The **efficiency** represents the ratio between output and input power. The difference consisting mostly of heat is designed as power loss. The efficiency of a gearbox is mainly influenced by friction. Hereby, spur and bevel gearboxes have superior spline efficiencies. Due to the high sliding property, worm gearboxes have low spline efficiencies. During the first 10 to 20 operations, worm gearboxes must be run with increased input torque to achieve nominal efficiency. After many operations, the efficiency of all gearboxes is reduced.

The efficiency depends on:

- Surface quality e.g. by grinding, polishing
- Lubrication
- Temperature impact

The favourable self-locking property of single-pitch spindles turns however into a drawback when considering **table 2.18**, since good self-locking properties correspond to unfavourable efficiency.

Self-locking should be selected appropriately.

Tab. 2.18: Typical efficiencies in civil engineering constructions for water applications for conventional components

Transmission elements	Efficiency	Self-locking
Actuator, single-stage worm Actuator, double-stage worm	< 0.45 > 0.55	Yes, with reservations[1)] No
Bevel gearbox Spur gearbox Womr gearbox	0.9 0.9 0.4	No No Yes, with reservations[1)]
Anti-backdrive device	0.9	Yes
Spindle/spindle nut Tr 80x10, single-pitch, dry Tr 80x10, single-pitch, lubricated Lantern gear/pinion dry lubricated	 0.18 0.27 0.80 0.90	 Yes, until pitch approx. 10° Yes, until pitch approx. 10° No No
Lifting cylinder with ball thread Lifting cylinder with trapezoidal thread	0.85 0.30	No Yes, with reservations[1)]
Hyraulic cylinder Hydraulic coaxial piston engine	0.95 0.85	Requiring additional engineering Requiring additional engineering
Brake motor		Yes, not in manual operation

1) Single-pitch worm gearbox and trapezoidal thread are self-locking at standstill. The self-locking effect can be cancelled by movement and vibration.

This also means: Simple self-locking is favourable. Multiple self-locking must be closely monitored. Since it leads to increased heat dissipation and reduces efficiency leading to lower lifetimes due to increased wear.

Further criteria are to be considered when deciding on the "more favourable connecting element". Both basic types in (**table 2.19**) are considered here.

Tab. 2.19: Essential differences between spindle nut/spindle and pinion/lantern gear transmissions

Spindles	Lantern gears
Low efficience of single-pitch spindles	High efficiencies
Self-locking (conventional pitches)	Not self-locking
Cannot be corrected after buckling	Correction after buckling fairly easy
Easyly enclosed, safer against accidents than lantern gears	Enclosing is more difficult
Protective actions against airborne sand, water and flotsam required	Accident prevention usually sufficient
Sensitive to lack of lubrication	Less sensitive to lack of lubrication (provide protection against environmental pollution)
Gimbal-mounting to avoid squeezing is difficult	Guide plate or buckle plate, buckle and counter rollers are sufficient for toothing with backlash
Potential accuracy results in high modulating accuracy	Appropriate for settings requiring less accuracy
Cleanliness is a must	Unsensitive against any pollution like grass

Financial advantages of variants depend on: Size, material, manufacturer and pertaining procurement or processing facilities. Often identical pricing is assumed and decisions are taken in favour of clear advantages like the robustness of lantern gears. If extremely high torques are to be provided, the only viable option are worm gearboxes with pinions and lantern gears.

Experts often repeat: Elements for civil engineering constructions for water applications must be allowed free movement. Rattling is preferred to jamming! But this also means: Basically, the pinion/lantern gear combination is far more robust. The missing self-locking can be achieved by introducing further equipment.

2.1.19 Rope and chain hoist drives

The gate leaf calculated in section 2.1.can be also be operated either via ropes or via chains. The particularity is that the gate can now only be pulled and no longer be pushed. The dead load must be sufficient to allow for safe closing. Self-locking must still be provided by the actuator/gearbox combination (**figure 2.110**).

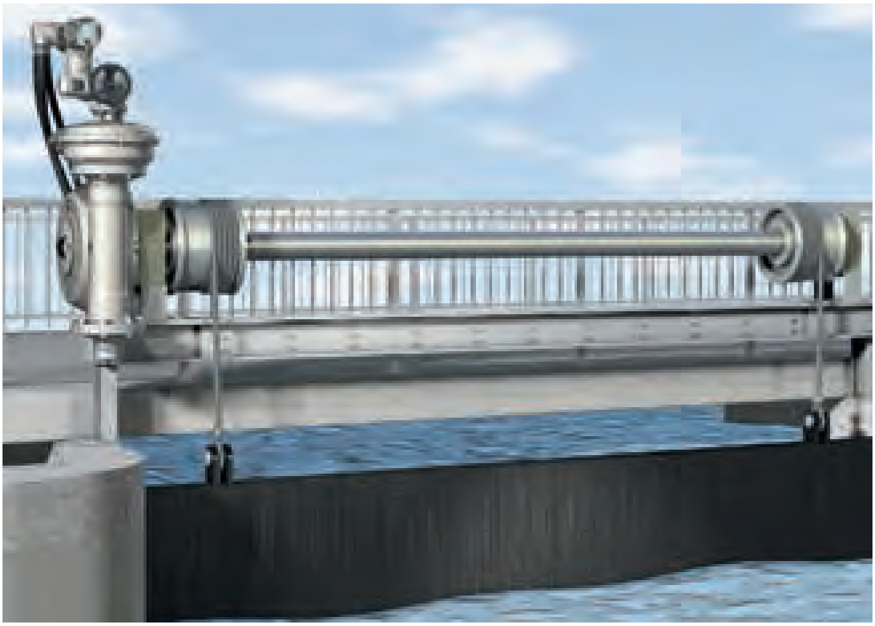


Figure 2.110: Sluice gate weir with double cable winch

Figure 2.111 shows an actuator/gearbox combination – actuator with AUMATIC actuator controls and worm gearbox with primary reduction gearing – mounted onto the shaft onto which two cable drums wind and unwind the ropes.

Figure 2.111: Actuator/gearbox combination with cable winch

Installations are known where a chain hoist was implemented as additional reduction between gearbox and shaft.

The disastrous consequences of the yearly monsoon rains in Southeast Asia are always dreaded. There is a high potential for introducing flood protection measures. The Muda River dam in Malaysia (**figure 2.112**) provides flood mitigation, safeguards drinking water supply and prevents the ingress of seawater into the Muda river during high tides. By means of laterally guided ropes, the actuator/gearbox combinations shown on top pull the gates visible in the lower windows by 5.55 m.

Figure 2.112: Flood protection structure at the Muda-River in Malaysia

SA 16.2 actuators and GS 400 worm gearboxes with GZ 35.1 primary reduction gearings rotate the laterally fixed rope drums via shafts (**figure 2.113**).

AUMA MATIC actuator controls allow for remote and local operation control (**figure 2.114**).

Large installations with link chain hoists are quite well known in the US. An example is the Tom Miller dam in Austin/Texas (**figure 2.115**). Nine "tainter gate" style flood gates can be opened to divert floods. On the left, a fixed weir was erected. A hydroelectric power station was installed to the right. The lake is an important drinking water reservoir.

Figure 2.113: Actuator/gearbox combination between respectively two cable drums

Figure 2.114: AUMA MATIC actuator controls mounted on an actuator/gearbox combination

Figure 2.115: Tom Miller dam in Austin/Texas

Two drums per weir field wind and unwind two chains respectively (**figure 2.116**). There is no possibility for a pushing operation. In fact, this is not required due to the high weight of the closing element.

Figure 2.116: Weir fields with hoist units

Figure 2.117: Hoist unit with link chains, in front GS 400 worm gearbox

Each flood gate is operated thanks to a combination of a centre mounted SA 25 actuator, two bevel gearboxes and two GS 400 worm gearboxes (**figure 2.117**). The output torque of each combination amounts to 135,500 Nm, the opening time required is 38 minutes for which the worm gearboxes require eleven revolutions. This leads to considerable heating up with the consequence that the standard grease had to be replaced by a special grease.

2.2 Double gate weirs

Deployment of single gate weirs is certainly justified when dealing with flood protection or water retention. A double gate weir or more specifically a double sluice gate allows for more varied applications. Basically, this is a horizontally divided gate leaf for which both halves run in parallel movement. They can be operated independently of each other. For correct sizing and daily operation, it is absolutely sufficient that the upper edge is located just above the target water level. However, the frame must allow lifting both sluice gates significantly above the high water mark. Different operation modes are possible:

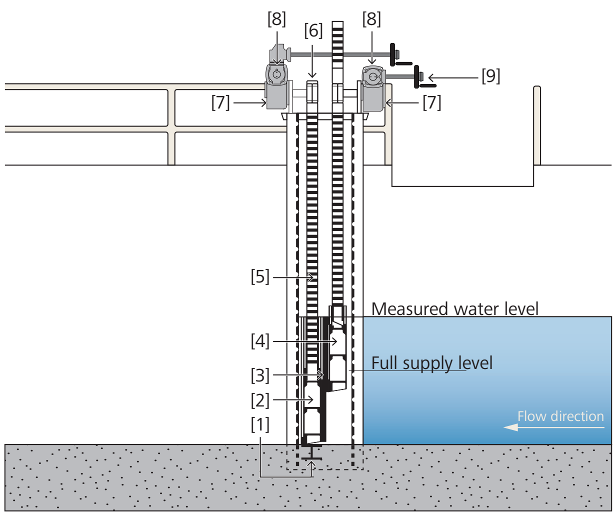
- Lower gate is closed, upper gate is completely lifted – flood protection, water retention
- Lower gate is opened on request, upper gate is completely lifted – headrace water regulation, flotsam and bed load discharge. If the lower gate cannot be opened, it will be discharged by completely lowering the upper gate. In practical applications, the lower gate is not operated sufficiently frequently. Therefore, the risk of sand and gravel deposits at the gate leaf is imminent. Sediments like suspended solids – residues from

rocks and stones – can be extremely dangerous. They mix up with water like concrete and there is the potential risk of complete blocking the gate which can no longer be opened. Regular flushing as specified in the operation instructions has to be performed to prevent aggradation.

- Lower gate closed, upper gate positioning on demand – fine regulation, floating debris and ice discharge (DIN 19704-2/3.1). Water regulation via the upper gate generates many operations.
- Lower gate opened on demand, upper gate lowered for floating debris discharge. Vibration at the closing element can occur in the event of simultaneous undercurrent and overflow.
- Lower and upper gates are lifted above the water level – discharging extreme floods. When secured appropriately, conducting visual inspection and repairs.

Double gate weirs can retain water and regulate water discharge. For head-race operation, the water bed is flushed, for tailrace operation, ice and floating debris are discharged.

Originally, each gate leaf was guided in separate recesses. Building this was very complex and often tightness problems occurred between both gate leaves. Nowadays, only the lower gate is guided in the frame of the retention installation. The upper gate runs in extended side members of the lower gate (**figure 2.118**).



[1] = Bed beam
[2] = Lower gate
[3] = Extended side member of lower gate
[4] = Upper gate
[5] = Langern gears
[6] = Pinion
[7] = Worm gearbox
[8] = Bevel gearbox
[9] = Handwheels

Figure 2.118: Double sluice gate with lantern gears in retention position (cross section)

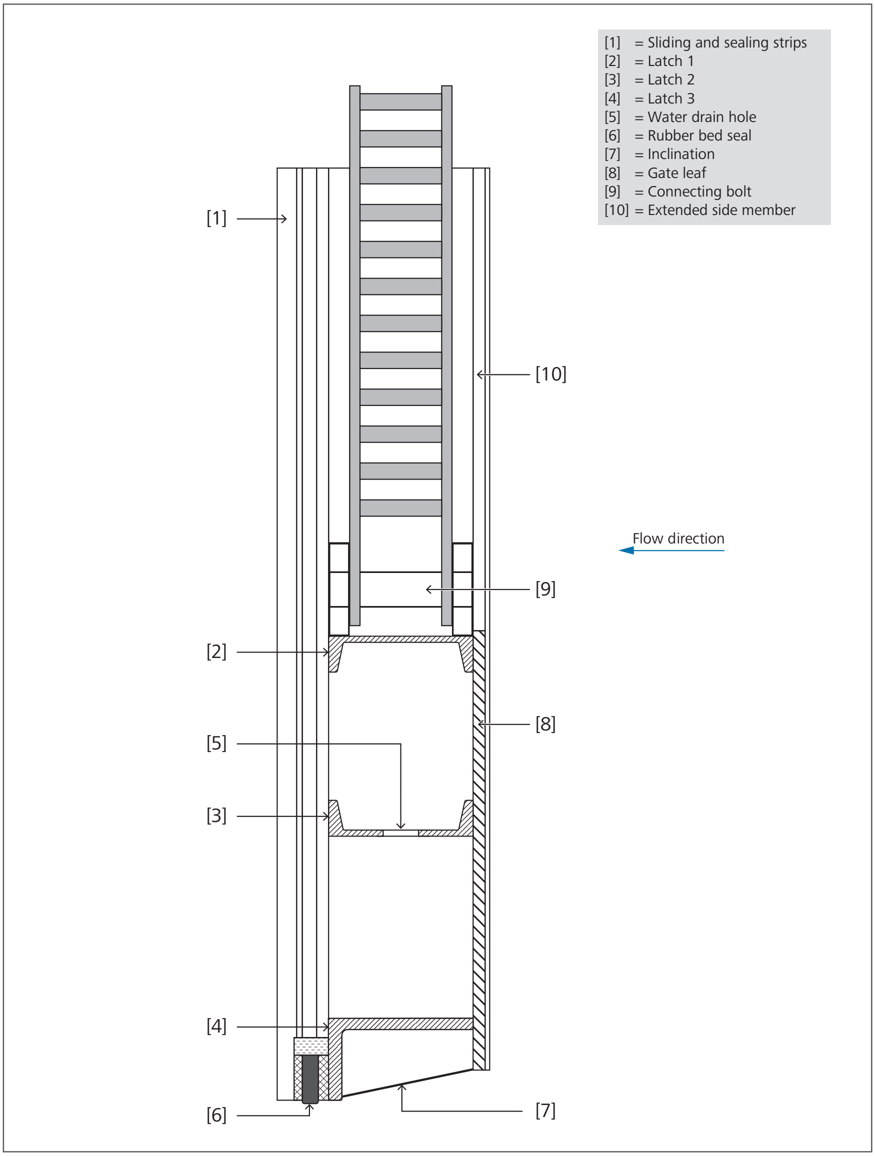


Figure 2.119: Lower weir with bar design

Possible installation of bars and bed seals is shown in (**figure 2.119**).

Like for the single sluice gate, sliding and sealing is made on slide and sealing strips with the known profiles (**figure 2.120**).

According to the hydrostatic pressure triangle, the highest pressure impact is on the water bed, the lowest impact on the water surface. As a consequence, the required actuator/

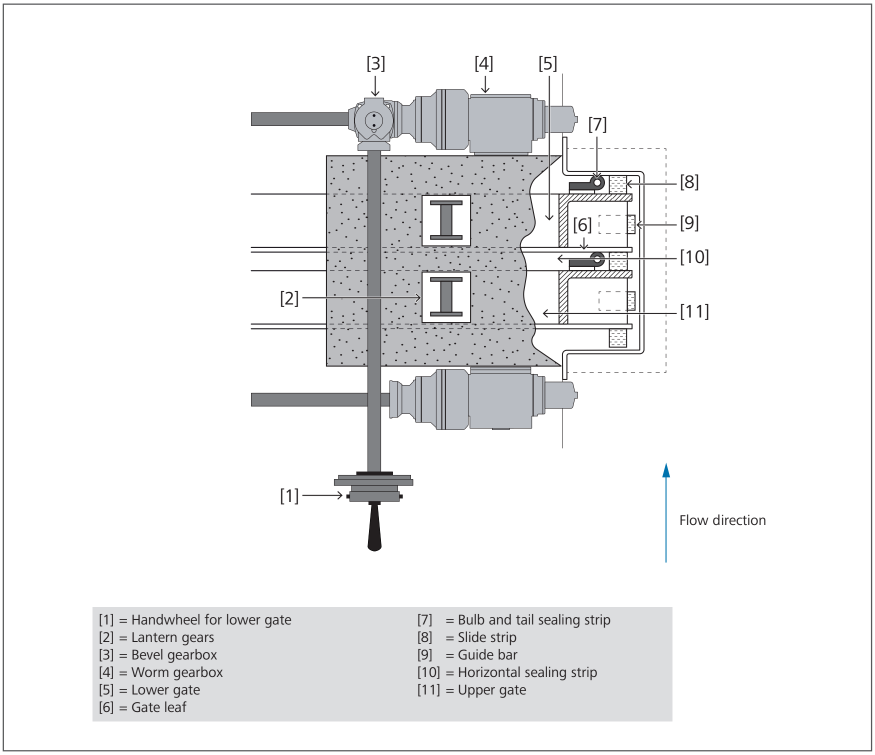


Figure 2.120: View on double sluice gate

gearbox combination for the lower gate located in tailwater with regard to flow direction must be larger and more powerful sizes must be selected for the the upper gate. The lifting force required by a partial sluice gate is significantly less than the force required by a single sluice gate.

At horizontal level, a sealing strip is provided across the total length between both leafs. Generally, one of the leafs is overlapping the second leaf by at least 20 cm. Electric end position monitoring and mechanical equipment like hooks or angles shall further prevent drifting apart of both gate leaves. In this instance or if the fixing structure breaks away, proper function would no longer be guaranteed. But even when everything runs smoothly, problems are likely to occur. For example for low waters if the horizontal sealing has run dry for quite a long time. A provisional remedy could be to increase the weight of the closing element – in particular for the upper gate – by attaching concrete bulbs, since in fact the selected actuator cannot cope with this situation. Furthermore, due to negligence, the seal might be damaged by the notorious stick-slip effect.

Like single sluice gates, double sluice gates are designed as sliding and roller gates. Spindles and lantern gears are used for force transmission. Due to confined spaces, arrangements of actuator/gearbox combinations either one behind the other or diagonally (**figure 2.121** and **2.122**) are feasible.

Figure 2.121: Double sluice gate with spindle actuators mounted one behind the other

Figure 2.122: Double sluice gate with lantern gear drive mounted diagonally to each other

Figure 2.123: Double sluice gate with lantern gears in a pier recess

Figure 2.123 shows an overflown double sluice gate with lantern gear drive. The water freely flows above both upper and lower gates. In the recess, the upper gate runs on the support arm of the lower gate. The picture does not clearly show that these are roller gates.

Figure 2.124: Double sluice gate with lantern gears fixed to the upper cross members

The frame allows lifting both sluice gates significantly above the high water mark. Recesses as inspection gates are located within the pier upstream and downstream of the sluice gate.

Figure 2.124 shows the suspension of the double sluice gate at the upper cross member within the flow area. This causes lantern gear abrasion and favours flotsam retention due to the offset arrangement. The sluice gate was operated in lifted position when the picture was taken.

A V-notch at the upper gate can prevent contamination since the water flow is directed through the sluice gate centre (**figure 2.125**). This measure also favours vibration reduction.

Figure 2.125: Upper gate with V-notch

To implement a low-cost version of double sluice gates, two half leaves of a sliding gate are located next to each other on the water bed at initial position (**figure 2.126**). Contrary to conventional double sluice gates, only one actuator/gearbox combination is required. Usually, the installation is tailrace operated. The same applies if the loose part remains on the ground and only the solid part is pulled to adjust a retention between minimum and maximum value.

If headrace operation is also required, the **latch hook** (**figure 2.127**) fixed to the upper gate shall engage into the lower part of the upper gate during lifting so that the lower gate can also be lifted. For this reason, this arrangement is also called **lifting gate**. After the required lifting operation, the water bed can be flushed free. The loose part is lowered by its own weight.

Figure 2.126: Water saving double gate. Bottom left in the picture: latch hook

Figure 2.127: Latch or pinion hooks of water saving double gate

Figure 2.128: The handwheels and their extensions were dismounted and stored

Usually, double sluice gates are only accessible from one side and can consequently only be operated from this side. To operate the remote sluice gates or their actuators, restriction to handwheel drive would not be sufficient as shown in **figure 2.128**.

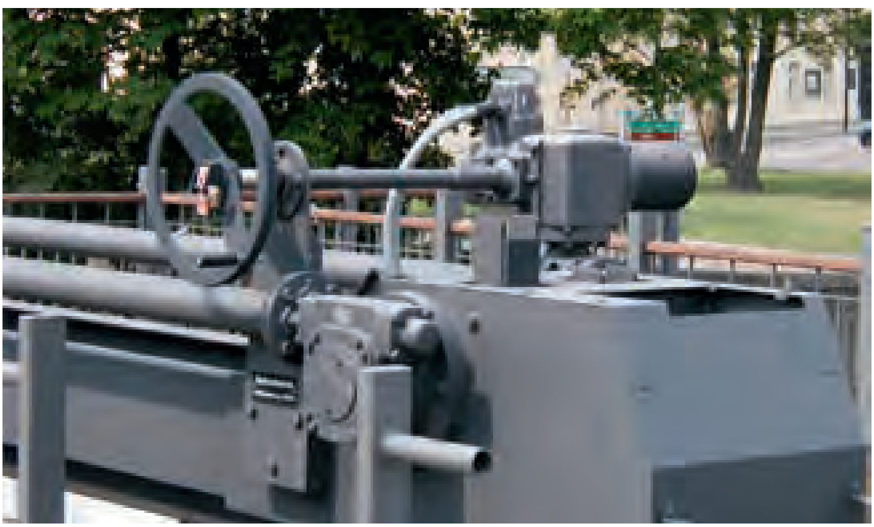


Figure 2.129: Upper and lower gate manual operation from the same side

Figure 2.130: Upper and lower gate operation from one side. Lever arrangements and handwheel connection are enclosed

In tenders, an acceptable design might also be called “collision-free crossover”. Many solutions are available. For the manually driven version, handwheel is equipped with a spindle extension (**figure 2.129**). An additional bevel gearbox allows for the difference in height of both handwheels.

Figure 2.131: Worm gearbox pair: RR at the actuator output, left in the picture, and LR right

The handwheel spindle extension can be installed as protected version (**figure 2.130**). Suspended mounting of right actuator allows for simple lower cross-over of the front connection shaft.

Furthermore, two different combinations are deployed at the same installation (**figure 2.131**). Front shaft bottom, actuator to RR gearbox version; rear shaft top, actuator at LR gearbox version. This also allows for one-sided manual operation.

It was quite difficult to find an appropriate solution for connecting an actuator with gear-boxes at the headwater side within a larger installation (**figure 2.132**).

Figure 2.132:
Actuator with connecting shaft for gearbox on head-water side (below)

At headwater side level, the input drive torque at the connection shaft is offset by 90° via a bevel gearbox to a second bevel gearbox for distribution to the two worm gearboxes with primary reduction gearings (**figure 2.133**). Both actuators are installed at tailwater side.

Figure 2.133: Tailwater side lower shaft cross-over of actuator to the headwater mounted gearboxes

Since it must always be guaranteed that the water within a weir can always be drained, the remaining waterway openings must take over the flood drain in case one weir field fails (**figure 2.134**).

The (n-1) rule applies.

Figure 2.134: (N-1) rule: If a waterway cannot be opened, the remaining waterways ensure water drainage

2.3 Shutter weirs

Up to now, we considered complete or total blocking of water courses by regulating the flow diameter using one or two gate leaves. The gate leaves can be operated in precise vertical direction via spindles, lantern gear, ropes or chains. Suspension is always located above the closing element.

Outlets in dykes – tidal outlets – are closed by a rotary movement when the water is fed during rising tide. During low tide or normal water level, they open due to the riverside water pressure. Sometimes, this process is supported by fixing a rod to feed any accumulated water from the back country to the receiving waters. In previous construction types, regulation was performed by a flap movement. Therefore, they are also named flap weirs. In later times, they operated by pivot movement like conventional house doors and are called tidal sluice gates (**figure 2.135**).

Figure 2.135: Water side sluice gate within a dam

Today, usually a second closing element – a penstock – is often added at inland water level for safety reasons (n-1 rule) (**figure 2.136**).

In the following considerations, the pivot-mounted closing elements – placed diagonally to the flow direction, horizontal to the water bed – shall be operated while using an appropriate lifting equipment from its horizontal position into the vertical position – if necessary – and vice versa. Consequently, shutter weirs are always **overflown** unless they are in closed

Figure 2.136: Penstock within a dyke

position. The are perfectly suited for water regulation and allow easy discharge of floating debris and ice. Contrary to the sluice gate weirs, the water bed cannot be flushed during operation. For this, the shutter gates must be lowered to 0° which means completely laid flat. The retention within the river would no longer be maintained.

Figure 2.137: Shutter with trapezoidal profile reinforcement, tailwater

The impact of ice force on the actuator configuration is stronger than for sluice gate weirs. The shutter is considered as particularly sensitive to vibration.

Shutter weirs can retain water and regulate water discharge. Exclusive tailrace operation allows discharge of ice and floating debris. Water bed flushing is not possible.

There are many variants. A smaller shutter can be operated single-sided by means of a connecting element. For narrower weirs, a steel panel reinforced with a bar is sufficient. Fitting a trapezoidal profile or similar is advisable for wider weirs to increase **torsional rigidity** (**figure 2.137**). The lateral bulb and tail seals – refer to the left in the figure – slides on baffle plates.

A single-side operated shutter can be as large as 20 m provided it is equipped with a fish-belly profile with torsional rigidity properties (**figure 2.138**). The name was assigned due to the fish-belly or lenticular shape in transverse section. If the rear of the fish-belly flap weir remains open, actions can be taken against rust but the disadvantages with regard to contamination prevail.

Figure 2.138: Fish-belly flap gate, closed tailwater side

For example, the actuator is driven by a worm gearbox in multi-turn version and pinion (**figure 2.139**). Buckle plate, buckle as well as eccentric and thus adjustable counter rollers fixed below allow for the variable angle of the lantern gear while lifting and lowering and ensure that the lantern gear is resting on the pinion.

Figure 2.139: Pinion and lantern gear connected by buckle and buckle plate

The drawing in **figure 2.140** shows the interaction of the elements. Here, one particular issue which has already been discussed in chapter 2.1.3.7 Ice force shall be highlighted: The impact of ice force on sluice gate weirs is mostly negligible for sizing the actuator if no lifting and lowering movement has to be made under frosty conditions. However, this is not the case for shutter weirs supported by connecting elements. If the shutter is closed

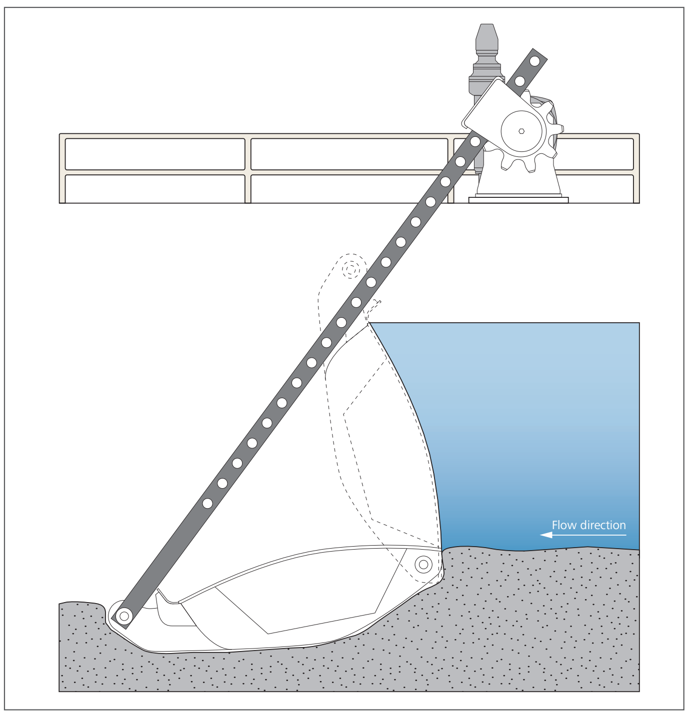


Figure 2.140: Single-sided shutter actuator with rising lantern gear

during winter operation which means it is in vertical position, the ice force acts almost perpendicularly against it (DIN 19704-1/525). Whereby in running waters, water almost always rises between the shutter and the ice thus preventing freezing. However, it must be taken into consideration.

Connecting element and worm gearbox must be suited for holding. Ice force and the resulting torque can easily amount to ten times the traction torque. However, the impact is only static and is therefore only referring to the breaking torque of the gearbox. Depending on the gearbox size, designers specify the breaking torque at a multiple – with AUMA at double – the nominal torque, whereby plastic deformations may already occur at factor 1.5. This applies provided the process is static and that self-locking is not cancelled by handwheel movement, for example. For this reason, the handwheel should be secured with a padlock.

A safe alternative is of course to equip the shutter with a locking mechanism to prevent winter operation if this agrees with the provided operation scheme. The necessary equipment like mounting and inspection eyes are already provided. Due to the assumed ice force, which can be considerable in calm waters, large radial gates or shutter weirs had to be ruled out when selecting the closing element type.

Besides lantern gear drives, spindle actuators and their further development, the lifting cylinders, are deployed. Electrical lifting cylinders – another designation for lifting cylinders – are equipped with gimbal mounted bearing journals (**figure 2.141**). For winter operation, similar considerations have to be made for the lifting cylinder solution as with the previous example.

Figure 2.141: Shutter with actuator and gimbal mounted lifting cylinder

Figure 2.142: Fish-belly flap gatewith flow deflector and V-notch

Flow deflectors and a V-notch (creating an irregular breakaway edge) are provided to reduce vibration (**figure 2.142**). If required, vent pipes are provided for air compensation.

While for a sluice gate weir the initial operation from the closed position is far more important, the torque curve of a fish-belly flap gate reaches its maximum at 45° or 60° depending on construction type and hydrostatic pressure (**figure 2.143**).

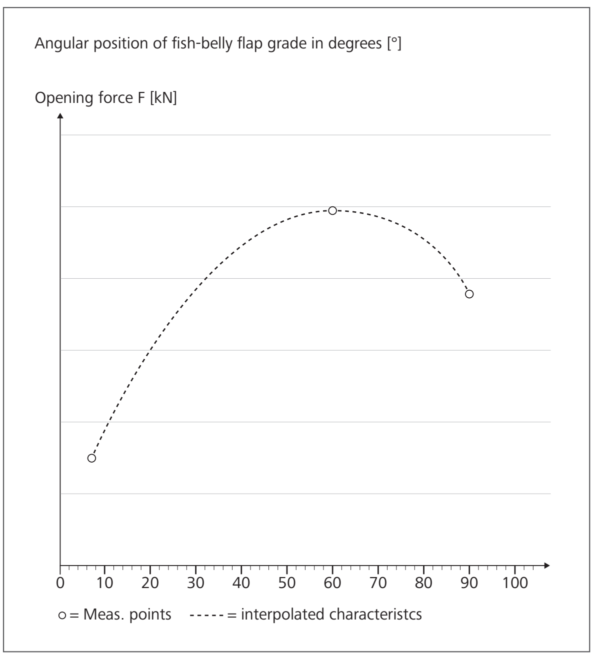


Figure 2.143: Force evolution of a fish-belly flap gate

Figure 2.144: Compression gate weir in Berlin Spandau with electrical lifting cylinder

The citadel weir in Berlin/Spandau looks like a shutter weir. With the pivot point located below the bridge, it belongs to the segment gate weirs (**figure 2.144**) since the segment gate pushes against the arriving water thus forming a compression gate weir. It is headrace operated.

Radial gates with tension gate arms are true alternatives to compression gate weirs. In our example, the water should be on the opposite side.

Since broad shutters are less stable in design, they have to be suspended at two points. In this instance, a handwheel or an electric actuator can be mounted onto the side mounted gearbox (**figure 2.145**).

The actuator/gearbox combination located at the right outside the weir and the pinions is connected by mechanical shaft coupling (**figure 2.146**).

The following shutters are suspended via two ropes. The ropes are shackled into holes provided within the shutter and led through telescopic guide runners into the rope pulley (**figure 2.147**). Consequently, they are protected against mechanical impacts and, by introducing respective measures, also against water impact. DIN 19704-1/10.21 indicates a safety coefficient D/d for the ratio between the rope pulley diameter D to the nominal rope diameter d. From the point of view of certain experts, the drawback of this special equipment could be that operation is limited to a pulling action and pushing is not possible in the event of ice and pollution. This could lead to disturbance for automatic level preservation. Constructors know about this issue but is it not considered as a major problem.

Figure 2.145:
Shutter operated manually via primary reduction gearing; pinions engage into lantern gears without additional gearboxes

Figure 2.146: Shutter with two lantern gears and outside actuator with gearbox

Figure 2.147: Shutter with two-sided cable winch and one worm gearbox

Figure 2.148: Shutter with two-sided cable winch and two worm gearboxes

Figure 2.149: Flap actuating gear with pulley

Depending on the size, one actuator and one worm gearbox could suffice.

Fitting two worm gearboxes would be rather more sophisticated (**figure 2.148**).

A further interesting solution is pulling the shutter with an electrical lifting cylinder and a pulley (**figure 2.149**). This divides the required tensile force in half, also called a simple shear. However, no pushing operation is possible. This structure is particularly appreciated since it is virtually invisible from the street level. In turn, service technicians criticise the difficult accessibility.

2.4 Combined weirs

Here, we are mainly dealing with shutter weirs equipped with flap gates. Two of the most common variants shall briefly be introduced.

- Shutter weirs with flap gates equipped with two adjustment units

The shutter weir with flap gate excels by the same advantages as the double sluice gate while finer weir regulation is possible (**figure 2.150**). Both components are simultaneously lifted and lowered via sprocket chains. Two-sided suspension is required. The sluice gate offers a drain at water bed level thus allowing an undercurrent. The drain is flushed and if necessary the level is modified either on headwater or tailwater side.

Figure 2.150: Sluice gate weir lifted with chain hoist. Fish-belly flap gate was straightened up thanks to electrical lifting cylinders

Figure 2.151: Sluice gate weir with laid down flap gate

The shutter is operated at both sides for fine regulation of the headwater level and also to discharge ice and floating debris (**figure 2.151**). Sealing is made via laterally mounted baffle plates. The shutter is overflown.

Figure 2.152 explains the interaction between chain and roller gate as well as actuator / lifting cylinder and shutter. The fish-belly flap prevents torsion at a maximum. The integrated reinforcing trapezoidal profile limits bending.

Synchronous operation of sluice gate actuators is based on an electric synchronous link, for shutter actuators on an **electronic synchronous link** (4.3.3.6).

- Shutter weirs and flap gates with joint adjustment unit

Another combination for the same task uses one chain with actuator at both sides (**figure 2.153**).

If the pinion – below the cover (**figure 2.154**) – is running counterclockwise, the closing element unit is lowered by means of the left chain part. If rotation is continued once completely lowered, the right part of the chain pulls the suspended shutter and straightens it up. This can be made until maximum dam height. In turn, if the pinion is rotated clockwise, the shutter is tilted first and the sluice gate with the shutter is pulled. Then, it is possible to operate simultaneously at underflow and overflow. Consequently, the weir can ensure both bed drain and fine regulation.

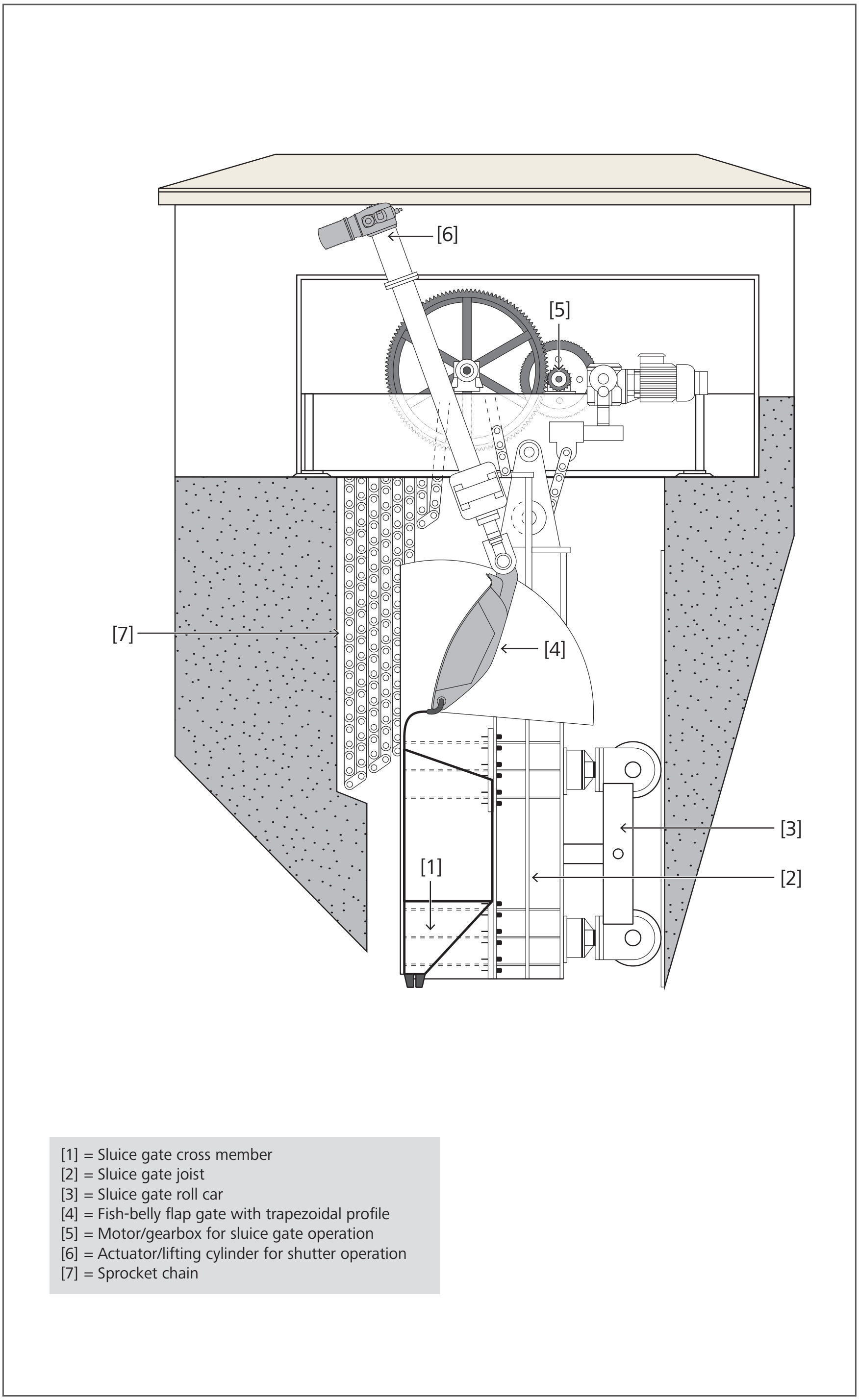


Figure 2.152: Schematic diagram of roller gate with mounted shutter

Figure 2.153: Sluice gate weir with fine regulation via flap gate. Both sides are driven by one actuator respectively

Figure 2.154: Sluice gate and shutter movement are made by a joint chain

Figure 2.155: Combined weir subject to undercurrent

The described chain movement is made simultaneously on the left and right side of the sluice gate. Mostly, the actuators are started and stopped simultaneously. If neither electric nor electronic synchronous links are used, tolerances in operating behaviour can be noted even for equally sized actuators. From time to time, synchronous operation must then be restored via mechanical adjustment.

Figure 2.155 shows the tailwater side of a lowered sluice gate with a shutter slightly pulled by a chain. The sluice gate was stabilised with bars.

3 LOCKS

Ship lifts or locks are required to ensure navigability along all water sections of waterways. Locks allow vessels to negotiate differences in water levels. Various considerations can be made on the advantages and drawbacks of locks. We shall discuss only one of them. "Comparative analysis for the Danube weir with lock in Jochenstein/Germany resulted in time savings for the vessel traffic of 8 % and fuel savings of 25 % compared to the time prior to the construction of the weir with lock." [2]

Locks are usually built in combination with mobile or fixed weirs. A **weir with lock** is a construction allowing to regulate and preserve the optimum water level for ship traffic, for energy generation or other subsections (**figure 3.01**). In smaller sites, the locks can maintain the water level without requiring a weir. In the first chapter, some lock types were mentioned. DIN 19703 is dealing with the equipment for locks. Frequently used **mitre gate locks** shall be further considered as an example.

Due to their construction type, the chamber lock with the lower and upper gates pushing against the water are the basic principle. Based on their function, they are classified as mitre gate locks (**figure 3.02**).

Figure 3.01: Weirs and locks in one site

Figure 3.02: Chamber lock; classified according to locking element: mitre gate lock

The basic design of mitre gate locks is shown in **figure 3.03**.

The upper gates are usually not designed as tall as the lower gates (**figure 3.04**).

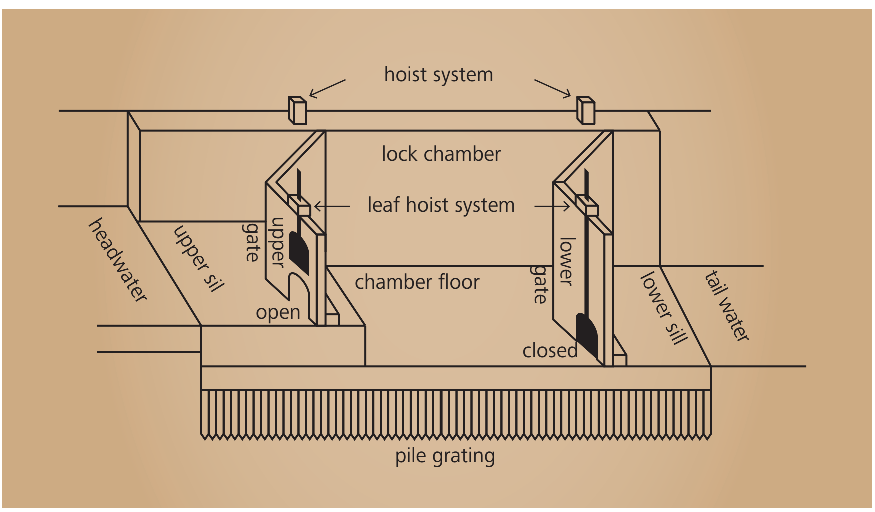


Figure 3.03: Historic representation of the basic function of a mitre gate lock

Figure 3.04: Drained lock with upper gate above upper sill

When closed, the lock gates are not straight but form an obtuse angle of about 140° against the water current, or two angle stops of respectively 18° to 22° (**figure 3.05**).

Complete locking process of a mitre gate lock with sluice gates Downstream locking is performed as follows:

Figure 3.05: Mitre gate in closed position

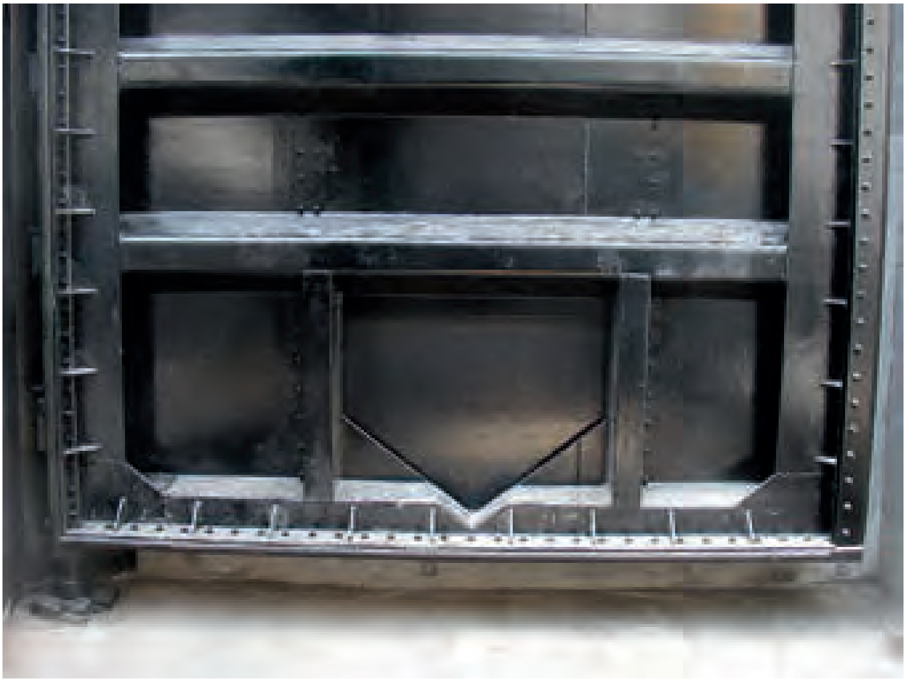


Figure 3.06: Mitre gate with four-side sealed sluice gate in the lower field, seen from the tailwater side

- The **filling gates** integrated in the double leaf lock gate of the upper gate (**figure 3.06**) – of course also one of the inventions of Leonardo da Vinci [7] – are opened, until the water level within the **lock chamber** is identical to the headwater level.
- When inspecting or repairing locks requiring draining the water, all the details are clearly visible (**figure 3.07**). Due to the large difference in water level, the water fed from headwater through the sluice gates is not led into the lock chamber via the **upper sill**, the concrete edge corresponding to the headwater bed, – since this would result in significantly turbulent water – but in front of the **bed drop** against the concrete **impact wall** through the lower openings of the **pre-chamber**.
- Once water level is equalised, the headwater mitre gate can be opened while the sluice gates remain open. The energy required to operate the gate is relatively low for this application. Due to time constraints or special water conditions, gates are more frequently opened against the still present, high water levels – specifications by the contractor can be for example 5 cm (2 inches) to 10 cm (4 inches) or even 30 cm (1 foot) for sea locks. This must be considered when sizing the closing elements and actuators. It is quite frequent, that the incoming water flood spills over the tailwater gate (**figure 3.08**).
- Once the vessel has entered the lock chamber, the upper gate is closed again and then the sluice gates are closed. Now, the water level must be lowered to tailwater level through the drains at the lower lock gate.
- Once equalised to tailwater level, the lower gate with opened sluice gates are opened – this saves both time and effort. The ship can leave the lock (**figure 3.09**).

Figure 3.07: Headwater gate: Smooth water inflow via the inlets located in the chamber floor

Figure 3.08: Water overflows the tailwater side due to premature opening of the headwater side

The load cycle is complete with consecutive **upstream locking** including

- Vessel entry
- Closing the lower gate
- Closing the sluice gates in the lower gate
- Opening the sluice gates in the upper gate
- Filling the lock chamber
- Opening the upper gate
- Vessel exit

A complete locking process is achieved.

Details of this abbreviated description shall be further explained:

Figure 3.09: "Have a good journey!"

Draining and filling the lock chamber

Lock chambers are drained and filled by:

- Sluice gates or slide gates within lock gates
- By-passes with wedge roller gates or cylindrical gates
- Filling shells in radial gates with compression gate arms
- Hotopp siphons

Sluice gates within lock gates

In our example, either sliding gates (**figure 3.10**) – the gate is properly positioned by means of a clamping fixture – or wedge roller gates (**figure 3.11**) fitted within the lock gate must be opened. In the past, wedge roller gate profile design was predominantly wedge-shaped. A modern wedge roller gate is slightly larger at the top than at the bottom. In closed position, the gate is tightly seated at both sides as well as at the lower and upper horizontal sealing strips. When opening, it is pulled off the four sealing strips at the same time.

Figure 3.10: Adjustable sliding gate within a mitre gate, headwater side

Figure 3.11: Wedge roller gate within a mitre gate, headwater side

As mentioned in section 2.1.1.2, the tailwater force can be deducted from the headwater force for calculating the gate actuator (**figure 3.12**). The hydrostatic differential force remains. In this case, the wave pressure – swell and suction = adding upstream side while deducting downstream side – must be considered (DIN 19704-1/5.2.2b).

$$F_{HD} = F_{HHW} - F_{HTW}$$

Roller gates are typically used as intake and drainage gates followed by wedge roller gates and sliding gates [6]. All gate types are mainly operated via spindle or electrical lifting cylinders but at times also via lantern gears (**figure 3.13**).

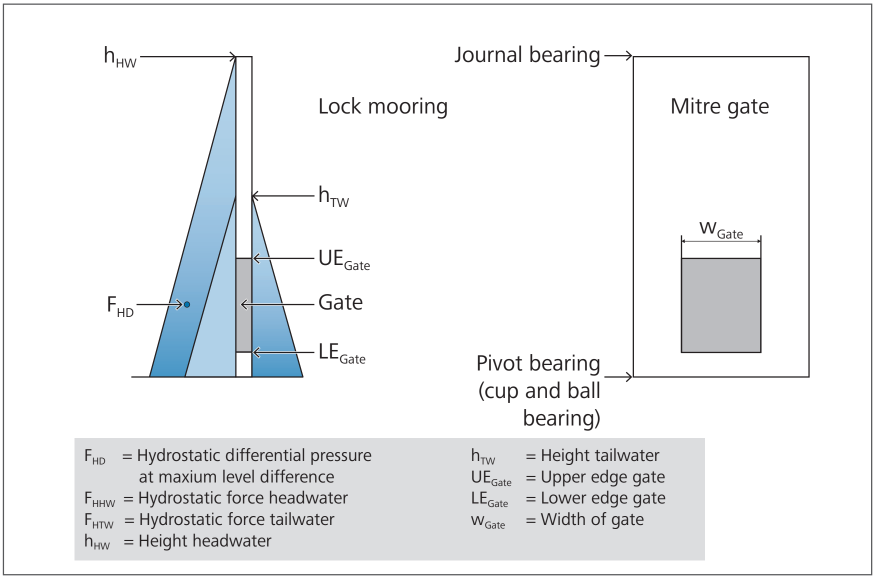


Figure 3.12: Conditions at the mitre gate with locked gate inlet

Since lock gates are weakened in their stability due to the integrated sluice gates and are more complex to manufacture, this solution is only used for about 45 % of locks and for smaller sites.

Figure 3.13: Lantern gear gate mechanism at a mitre gate

By-passes

By-passes [6] are implemented in about 50 % of locks. The water level equalisation is achieved using wedge roller gates or cylindrical gates (**figure 3.14**).

The water flows via short by-passes around the lock gate into the chamber. However, the water might also be led via **side by-passes** along the lock chamber or via **base by-passes** at the lock bed, which means at several inlets at the chamber floor (**figure 3.15**).

For safety reasons, underwater inlets – here yellow triangles – are often signalled at the lock walls (**figure 3.16**). Gate actuators are often equipped with brake motors to ensure safe stopping.

Radial gates with compression gate arms and filling shells

Segment gates are only integrated within upper gates. When lowering the gate, the horizontally opening gap allows the water to stream into the lock chamber. This principle is called **pre-head filling**.

Hotopp siphons

The siphons work as follows: Via a suction tube placed at the highest point of the reservoir bend, (**figure 3.17**), a vacuum is generated within the siphon. A large vacuum tank is connected. If necessary, the respective valve is opened to allow the siphon to transport

Figure 3.14: Cylindrical gate with actuator and linear thrust unit at the by-pass of a lock gate

Figure 3.15: Drains in side by-pass channels

Figure 3.16: Lock chamber with by-passes.
In front, the cylindrical gate drive of a by-pass.
Opposite at the chamber wall: water inlet points marked in yellow

Figure 3.17: Hotopp siphon

water into or out of the lock chamber. The suction process can be interrupted either by an emergency opening at the siphon or a second opening device which can be electrically operated from the control station.

Siphons were used in locks and weirs. They save propulsion force, operating and maintenance cost, but are no longer built. Existing older installations (5 % of all locks) are gradually replaced with different technologies. Also due to the fact that the high transport capacity is challenging. To improve the outflow behaviour, mixed systems have also been implemented like the simultaneous use of siphons and sluice gates.

In general, other filling and draining combinations are used with locks, like for example sliding gates and by-passes.

Lock gates and their operation

■ Mitre gates

Typically, mitre gates in locks must surpass an increased starting torque. After these starting peaks which have to be considered when sizing the actuator, the applied torque is slightly reduced which has to be considered for lifetime determination. **Figure 3.18** shows the opening graph of the gate position from 0° considering the stop angle of 20° while increasing to 70°. This results to 90° including the stop angle. The same is valid for the closing direction.

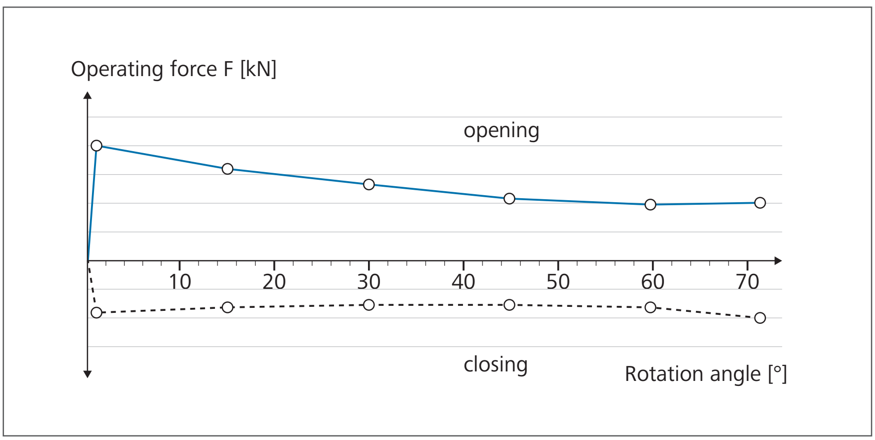


Figure 3.18: Force graph of a mitre gate

The low operating time of approximately 3 min. and subsequent pause time of minimum 20 min. is quite favourable for the actuator combination. If the permissible lifetime run torques were considered for actuator selection, the standard type of duty S2 - 15 min is sufficient for mitre gate actuators.

Typically, lock gates are operated faster than weir gates.

Figure 3.19: Mitre gate operation via lantern gear and pinion

The determination of the number of complete strokes or the actuator hollow shaft revolutions is quite straightforward for mitre gates. Except the possible different hydrostatic pressures, each locking follows the same principle. The travel is always one complete stroke. The number is recorded into the lock log.

The main operating devices for mitre gates are horizontally mounted lantern gears (**figure 3.19**) and electrical lifting cylinders (**figure 3.20**). Operators appreciate or decide in favour of the following characteristics:

- Compact and environmentally friendly design,
- Low installation efforts and expenses,
- Efficiency or self-locking are determined by the available thread – trapezoidal or roller ball thread. Planetary roller ball threads are capable of transmitting higher loads,
- Enclosure protection depends on the motor used: e.g. IP54 for brake motors, up to IP68 for pot-type motors

The following must be considered: The extended, slightly greased tube is not protected and therefore exposed to airborne sand and bird droppings. There is a potential risk of leakage when retracting. The following versions are available for deciding on the robustness of the piston rod: NIRO, hard chrome plated, ceramics.

Figure 3.20: Mitre gate operation via electrical lifting cylinders

To improve accessibility, the laterally mounted actuator handwheel can be realigned by 90° when using a bevel gearbox (**figure 3.21**).

Figure 3.21: Horizontal handwheel offset by 90° in front of an actuator

Figure 3.22: Toggle lever at a mitre gate led via pinions

For installations in confined spaces, gear segments with buckling push rods are favoured (**figure 3.22**).

The drive torque of slider crank gears (**figure 3.23**) is virtually constant across the total stroke. At constant operating speed, the lock gate speed is slowed down when approaching the closed position. This corresponds to the soft stop electronically generated by frequency converters.

Figure 3.23: Slider crank gear with rounded lantern gear

Figure 3.24: Lantern gear segments at a mitre gate

Sometimes, rounded lantern gears, so-called lantern gear segments (**figure 3.24**) or rounded toothed racks are used (**figure 3.25**).

Mitre gate drives shall not significantly change the actual position, but may slightly withstand wind and hydrostatic pressure caused by an entering vessel – also called swell – or against the pull on the gate caused by the vessel leaving the lock – also called suction.

Figure 3.25: Mitre gates operated via rounded toothed racks

Figure 3.26: Improvement by self-locking actuators: slowed-down gate operation of a mitre gate lock

Lantern gears and lifting cylinders with highly efficient ball screws operate quite well with self-locking actuators. If the actuators are not self-locking and external forces are high, electric counter steering is required in particular in the end positions. If, however, the gate position can only be maintained by extremely frequent adjustments, brake motors are absolutely necessary (refer to 4.3.3.5 Opening and closing mitre gate locks). Apart from this special case, self-locking provided by the actuator is typically sufficient (**figure 3.26**).

Lantern gears and electrical lifting cylinders can be equipped with spring packs in cups for approaching the end positions (**figure 3.27**). The dampening effect of the springs upon starting and stopping can be electronically replaced by soft start and soft stop by a frequency converter (refer to 4.3.3.7 "Electric special solutions").

In lock designs, the use of flap gates instead of mitre gates are considered as exceptions due to their form and function. They are virtually never used as lower gates; in turn, mitre gates represent 73 %, slide gates 8 % and lifting gates 15 % [6]. Typical applications for upper gates design are 8 % for flap gates, 55 % for mitre gates, 30 % for lifting gates and 7 % for segments gates. Flap gates are usually combined with mitre gates.

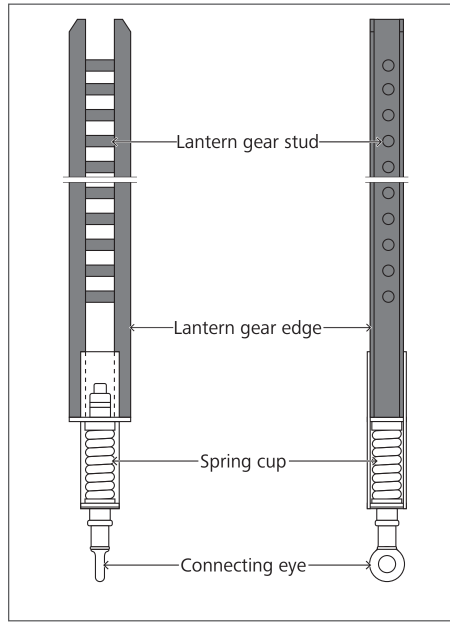


Figure 3.27: Lantern gear with spring cup

Flap gates

A flap gate can be operated by an actuator rotating a round lantern gear (**figure 3.28**) or with an electrical lifting cylinder (**figure 3.29**).

Figure 3.28: Buckled lantern gear with pinion, planetary gearing and actuator for a flap gate

Figure 3.29: Flap gate as upper gate

For downstream locking, the by-passes are opened (**figure 3.30**).

Once the water level is equal on both sides, the flap gate can be lowered to the ground (**figure 3.31**). Vessels moving upstream exit the lock above the flat-lying flap gate, ves-

Figure 3.30: By-passes within the sill base are open

Figure 3.31: The flap gate is laid flat

sels moving downstream enter the lock. The flap gate is closed behind a vessel moving downstream. The lock can be left once the sluice gates and the mitre gate in the lower gate are opened.

Manually operated locks

For smaller and less frequently used locks like in the Spreewald, manpower must be used for the locking process (**figure 3.32**).

Figure 3.32: Lower lock gate. Mitre gates and sluice gates are manually operated

Automatically operated locks

Automatically and electrically controlled locking processes completely replace locking aids and are independently operated by the skippers themselves (**figures 3.33 and 3.34**).

Figure 3.33: Automatically operated lock: "Turn the green lever for locking!"

Figure 3.34: Self-service at an automatically operated lock

Figure 3.35: Control centre

Differentiations can be made between **locally** and **remotely** or **centrally operated locks**. Locks operated by guardsmen waiting in the sluice gate keeper's house become more and more scarce (**figure 3.35**).

Depending on the importance of the locks, older models have already been converted in self-service locks. They are supervised from a central control centre. However, it happened and still happens that self-service locks are converted into remote-control locks. Control conventionally happens via cable. In recent times, the trend of retrofitting with fieldbus technology via fibre optic cables or via satellite can be observed. All sites are monitored via high resolution cameras to allow immediate identification of abnormalities. High power microphones are installed to detect abnormal noises. Skippers might also be instructed via loudspeakers.

4 OPERATION TYPES FOR CLOSING ELEMENTS

Closing elements as well as filling and draining equipment of hydraulic steel structures are subject to mechanical (20 %), electro-pneumatic (only marginal), electro-mechanical (55 %) and electro-hydraulic (15 %) operation [6].

- **Mechanical operation** using a handweel or crank handle are considered as mere emergency solutions in the event of auxiliary power failure or as standard when commissioning. In smaller installations, mechanical operation is quite common and often the only type of operation.

- **Electro-pneumatic operation:** Applications where sluice gates or lock gates are subject to electro-pneumatic operation are not known. Usage is not feasible since a high pressure level must be continuously maintained. However, pneumatic operation is used for other important tasks.

- **Electro-mechanical operation** and pertaining properties have already been described with examples. Electric actuators are often deployed in combination with gearboxes. Basically, all connecting elements can be operated either by electrical or mechanical principle:

  - Pinions/lantern gears
  - Spindle nut/spindle
  - Cable drum/cable
  - Chainwheel/chain
  - Lifting cylinder: The lifting cylinder was created by combining spindle nut/spindle. When mounting an electric actuator, the combination results in an electrical lifting cylinder (**figure 4.01)**.

Figure 4.01: Electrical lifting cylinder at fish-belly flap gates

- **Electro-hydraulic operation** has always played and still plays a crucial role in civil engineering constructions for water applications. We will have a closer look at their advantages and drawbacks.

4.1 Mechanical operation

Smaller closing elements which are rarely operated are designed for exclusive manual operation (**figure 4.02**). The inside view of a sluice gate activation system (**figure 4.03**) shows the interaction of handwheel – worm gearbox – pinion.

Figure 4.02: Manually operated double gate weirs (left) and manually operated pleasure boat lock

4.2 Electro-pneumatic operation

Electro-pneumatic operation is mainly deployed for lifting and hoisting technology in locks (**figure 4.04**). **Negative pressure** is applied when a vacuum pump or the impact of a water jet of flowing water (i.e. no electricity is required to operate the pump) is used to almost evacuate a hermetically closed room. The vacusum is a useful method to fill or drain lock chambers or reservoirs via weirs at a moderate cost. Today, such installations are no longer used.

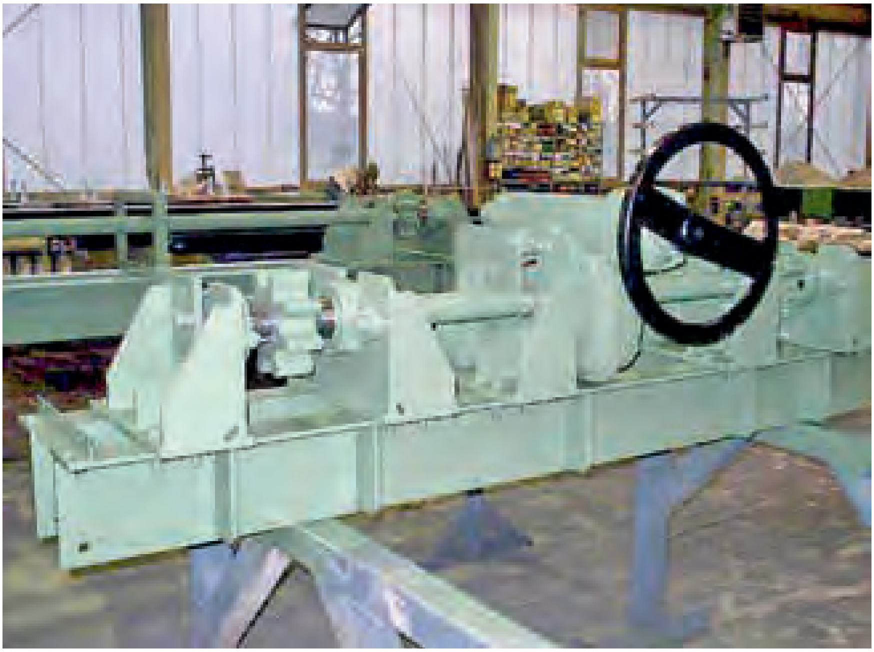


Figure 4.03: Manually operated pinions

Figure 4.04: Hotopp siphon on a side basin of a lock for draining and filling the locks on both sides

Inflatable weirs work with **excessive pressure**. A compressor is used to set the desired headwater **sill height**. Flow deflectors – also called breakers – promote stability (**figure 4.05**).

With the "Obermeyer system", the total length of the tube is preceded on the headwater side with a pivoting steel shutter fixed to the water bed. Consequently, the tube is protected against damage, floating debris and bed load.

Figure 4.05: Inflatable weir with flow deflectors

4.3 Electro-mechanical operation

4.3.1 Electric drives

Electric drives are electric open-close actuators and electric modulating actuators. They are available with or without integral controls and must comply with DIN 19704-2/9.2.

4.3.1.1 NORM actuators

Standard or NORM actuators (**figure 4.06**) require external controls. Besides the motor, they house mechanical components such as worm gearings, bearings and springs. Furthermore, they comprise limit and torque switches for open and close directions as well as blinker transmitters and thermoswitches. They are equipped with switch compartment heaters to

Figure 4.06: NORM actuators with common local controls at lower lock gate

protect against condensation. On request, they can be equipped with a mechanical position indicator and a remote position transmitter.

In civil engineering constructions for water applications, AUMA deploys the SA version in duty type S2 for open-close duty. For modulating duty demanding frequent adjustment of the MOV at short time intervals, the SAR version in duty type S4 is used.

SAR is a German abbreviation meaning SA = [StellAntrieb] actuator and R [Regelbetrieb] modulating duty. A multi-wire control cable and a four-wire motor cable are used to liaise the actuator to the control cabinet housing the necessary switchgear like contactors or thyristors. Separate local controls in proximity of the actuator require a further control cable to the control cabinet. Further information for NORM actuators as well as the actuators equipped with AM or AC actuator controls is available in the manufacturer’s documents.

4.3.1.2 Actuators with AM actuator controls

Actuators equipped with AUMA MATIC (AM) actuator controls (**figure 4.07**) and AUMATIC (AC) actuator controls are supplied with complete motor controls, switching elements and local controls. All electrical components such as limit switches, torque switches, blinker transmitters and thermoswitches including heater and position transmitter are housed within the actuator controls and readily wired for operation.

Figure 4.07: Actuators with AUMA MATIC (AM) actuator controls on single-stem gate valves

Figure 4.08: Actuators with AUMATIC (AC) actuator controls on a double-stem gate valve

Locking of several actuators is not possible with actuators equipped with AM actuator controls because external switching elements are required. Contrary to NORM actuators, combinations with AM actuator controls save the torque signal. This means that the notorious pumping effect of continuously switching on and off will not occur. If required, actuator controls can be mounted separately from the actuator on a wall bracket.

Equipped with comprehensive intelligence, actuators with integral AC controls (**figure 4.08**) can be parameterised via the local controls' push buttons without opening the housing when using a magnetic limit and torque transmitter (MWG). They are equipped with a display visualising all important operational parameters as well as the number of strokes. Basically, the MWG allows for highly precise limit and torque measurements. AC actuator controls are capable of working with all conventional fieldbus systems. It can be operated by either parallel or serial wiring or both, if desired. When equipped with both types, the required version can be easily switched over.

- Parallel wiring (conventional): In the past, devices were exclusively wired in parallel. This meant that each device was liaised to the control cabinet by means of a multicore cable. Today, more sophisticated connection types are taking over.
- Serial wiring (fieldbus): A two-wire cable interconnects all devices either individually or jointly to the control cabinet or master. Further selection can be made between star, line or loop wiring topologies, either using copper cables or fibre optic cables (FOC) forming an optical loop (**figure 4.09**).

The advantages of fieldbus applications compared to parallel wiring:

- Savings in wiring
- Distribution boxes and measuring cables are no longer required since measured values like levels and pressure can be directly transmitted to the AUMATIC.
- Later modifications and expansions are easy to implement.

Figure 4.09: Optical ring

4.3.2 Differences between measuring encoders

The reasons for maintaining water at a constant and predefined level is of crucial importance. To accomplish this task, actuators are equipped with position transmitters. For position measurement systems to assigned weirs, the precision tolerances should be appropriate. The following features are possible:

- Potentiometer. Version as film potentiometer. Linearity ±1 %. Suitable for display and control in frequent operation schemes. Maximum distance of 5 m from the low voltage supply.
- Electronic position transmitter (RWG) consisting of film potentiometer and measuring transducer for standardised current signal 0/4 – 20 mA. Linearity ±1 %. Suitable for closed-loop control. Applicable even for larger distances.
- Magnetic limit and torque transmitter (MWG, absolute encoder). Linearity depends on the stroke. Therefore, indication of absolute accuracy makes sense. The MWG signal reduced by the actuator controls has an absolute accuracy of ±8° at the output. Due to the improved linearity, the MWG is also suitable for more demanding control and synchronisation tasks. Distances up to 100 m are possible.

Since these position transmitters are housed within the actuator, they are closely interconnected via geared wheels and other transmission equipment but backlash might occur due to gear wheel tolerance. To exclude this backlash at an early stage, the transmitter should be as close as possible to the actuator or gearbox output. One possibility is to mount a limit switching device for gearboxes in multi-turn version including the transmitter directly onto the worm gearbox (**figure 4.10**).

Figure 4.10: Position indicator at output drive to pinion of lantern gear

Figure 4.11: Absolute encoder on actuator of a lifting cylinder

To obtain direct and consequently more precise measurements, the inclusion of an absolute encoder – in this instance an optical transmitter – is placed on the actuator output shaft and thus the input shaft of the electrical lifting cylinder (**figure 4.11**).

4.3.3 Special control functions for hydraulic steel structures

Automating sites in civil engineering constructions for water applications is a broad field. Previously, this was often performed outside the actuators using separate controls. Today, integral actuator controls are state-of-the-art. Reliable diagnostics is the prerequisite. First of all, this affects the motor temperature monitoring. Thermoswitches and PTC thermistors within the motor winding prevent motor overheating by tripping on time. Today, the AUMATIC actuator controls provide the following features:

- Operating time warning when exceeding the operating time set for the stroke
- On time warning when exceeding the maximum running time relating to duty type, for example after 15 min at S2 - 15 min
- Fault signals for PTC thermistor motor protection tripping

Self-monitoring is available for
- Switchgear control
- Magnetic limit and torque transmitter
- Interaction of internal AUMATIC sub-assemblies

In the past, customers in civil engineering constructions for water applications required electrical companies to fulfil the following control tasks:

4.3.3.1 Locking of double sluice gates

Double sluice gates must be operated up and down either individually or as a combination and sometimes have to be completely pulled out of the flood area. Sluice gates which run next to each other – the upper gate runs on the extended support arm of the lower gate (**figure 4.12**) – must not run apart neither in the upward nor the downward movement since the resulting gap would be very difficult to close. This might particularly occur if both sluice gates operate at different velocities.

Mechanically, the interlocking of the sluice gates is implemented while respecting the safety overlapping – for example by 20 % – using angles and latches. They are mounted to engage in case that the sluice gates are driven apart and this blocking will cause the actuator to shut down. The drawbacks are that the fixing structure might break. In **figure 4.13**, two T-steel pieces welded onto the upper gate prevent the lower gate from continuing the rising movement. The angle bolted horizontally onto the extended side member limits the upper gate's rising movement.

Figure 4.12: Right position: Lantern gear at lower gate. Mid position: Extended support arm of lower gate with bulb and tail profile. Left position: Lantern gear of upper gate

Figure 4.13: Cross-over protection at double sluice gate

A more elegant solution would be to set the actuator's intermediate position switches in a way that they trip once the rising upper gate reaches the end of the safety overlap. In the same way, external controls can activate the lower gate to run simultaneously with the upper gate until the combination has reached the preselected value. The so-called **gap lock** can be programmed via a PLC, which is subject to tender.

When only operating the lower gate, dual friction occurs: firstly at the frame and secondly at the immobilised upper gate. For this reason, the locking request favours the **simultaneous operation**.

For safety reasons, mechanical locking must always be provided!

Locking mechanisms are also necessary for manually operated double sluice gates, since both sluice gates are virtually invisible under water and there is a potential risk of a gap between the two gates.

For water level control, upper and lower sluice gates are both equipped with a duct comprising measurement probes (**figure 4.14**). Three measurement probes are available to ensure that in the event of failure of one measurement, two identical and thus realistic results are obtained. Once a selected critical condition is reached, an SMS or another type of message is transmitted to the control station via modem. The problem can be identified via computer and corrective actions are taken.

Figure 4.14: Measurement point installation in a pit

4.3.3.2 Automatic deblocking

It often occurs, that weirs cannot be closed due to water contamination through floating debris. Flotsam can comprise thick branches or bed load. This also includes gravel and rubble. All these elements sink down to the river bed and consequently onto the weir bed beam or the lock sill.

Once the sluice gate is pushing down on the flotsam, torque increases until actuator tripping. Later, staff will open the weir again to eliminate the contamination and allow free sluice gate movement. Of course, this happens at regular intervals.

Upon tripping, the automatic deblocking feature automatically opens the gate by a predefined stroke length and closes it again. This can be repeated as often as desired. Hereby, a cross section reduction results into a higher flow velocity allowing the debris to be flushed away. This method will certainly not work for large debris, but these are rather rare.

Of course, particular attention has to be paid to this feature. If we are dealing with a one-sided deposit – for example a car tyre –, the complete driving torque which has been calculated for two sides is acting on the connecting element with the deposited obstacle. The consequence can be the buckling of the connecting element. If a lock mitre gate with spherical bearing is operated against a wedge-shaped deposit during closing, the gate risks to be levered out.

4.3.3.3 Emergency start-ups

- **Multiple start-ups**

For example, in case of floods, a sluice gate is closed and opened again once the water level returns to normal. This might function perfectly for a long time. The site was calculated for this purpose and the actuators were sized accordingly. But suddenly, the sluice gate can no longer be pulled. The torque switch in opening direction has tripped. Long standstill times of the closing element can lead to a type of microscopic positive locking. In particular older slide and sealing strips literally glue with the slide rails. This makes the transition from static friction to sliding friction more difficult at starting operation. This is one of the reasons why the actuator to be selected should have a reserve to provide an increased unseating torque for a short time, if required. If this is not the case, larger actuators are the demanded solution.

But the question is what happens during start-up when integrating a new “force pack” into an old site? The glued part of the sealing and sliding material which might additionally be deformed by incorrect mounting or temperature impact is pressed in the operational direction. Rubber is compressed and eventually starts to crack. Weaker slide strips are pushed and will finally corrugate and break. The closing element performs a jerky operation. The consequence is the notorious stick-slip effect.

In practical applications, this problem will be carefully tackled, provided it is not too late. After a short jerk of the actuator in opening direction, it is closed again. This opening and closing movement is repeated several times until smooth running. Furthermore, it is even possible to automate this scheme. Selection can be made if triple or multiple starts are necessary. This way, the material is protected and overload avoided.

- **By-passing safety tripping**

The valve industry knows cases in which, for example, a shutter was closed at virtually 100 % T_{Amax}. When opening and applying the identical torque, the closing element did not even move, but the torque switch in direction open had already tripped. Here, actuators equipped with AUMATIC actuator controls offer the perfect solution. The torque switches in open and close direction can be by-passed for the critical start-up phase which is often subject to torque peaks. Times between 1 s and 5 s can be selected. This ensures that the actuator can start while using a stall torque fraction. After the pre-set time, the standard torque monitoring is reactivated. This measure must represent an exception since it always translates into an overload impact on the actuator components and might impair the lifetime. During normal operation, a sufficient opening torque exceeding the closing torque must be within the setting range of the selected valve actuator.

In civil engineering constructions for water applications, a torque by-pass is only acceptable in open direction since the connecting elements risks to buckle when closing. However, this by-pass may only be a solution for emergencies!

There is only one suitable solution: The actuator must be sized for the maximum required torque adding a safety factor of at least 25 %. Regular functional tests of the complete weir must be performed for site maintenance, in particular the condition of the slide and sealing strips must be checked.

There is no danger should actuator spring packs be subject to wear due to overload. It can be proved by performing measurements outside the actuator using appropriate measurement flanges or measurement gauges. When obtaining the measurement result, the torque setting can be increased at the unchanged actuator or reduced at the actuator or the spring packs can be replaced. Measurement equipment inside the actuator's AUMATIC actuator controls work with springs to be controlled and cannot signal any material fatigue.

4.3.3.4 Elimination of resonance vibration

Depending on the water level and flow velocity, closing elements like sluice gates or shutters might be subject to vibration at undercurrent or overflow in certain areas. The resulting resonance vibration acts on all connecting elements, gearboxes and actuators. This might cause considerable noise and strongly disturb neighbours and even lead to further damage to the site while increasing wear. AUMA's AUMATIC actuator controls are equipped with an integral vibration sensor which switches off the actuators in case of exceeding predefined gravitational forces and frequency limits and prevent actuator damage. However, to protect the civil engineering constructions for water applications, using an external sensor on site directly at the closing element makes sense. In critical situations, the sensor can emit a signal to an external control unit. This will activate the actuator and the closing element will leave its position and deviate upward or downward within its pre-defined level range. Following a pre-set time or once the water parameters have changed, the actuator can be returned into the initial state by the controls.

4.3.3.5 Supporting locks to enter the desired operating status

- **Filling and draining lock chambers**

Typically, lock chambers are filled by pulling sluice gates integrated within the mitre gates. The consequence of the opening is a gush of water into the chamber which might present a particular hazard for pleasure crafts (**figure 4.15**).

A sluice gate roof might have a positive effect (**figure 4.16**). The opening pointing towards the bottom causes a reasonable flow capacity increase when pulling the sluice gate.

Figure 4.15: Strong roaring within the lock chamber

Figure 4.16: Roof of a sluice gate for settling the water surface

In turn, a hazardous pull effect can occur during draining. Consequently, the sluice gates have to be moderately operated into both directions which also preserves the constructions. The following options are feasible:

– Stepping mode: The actuator operation can be interrupted by pauses. Operation is made in steps. This is commonly called stepping mode. The active and pause times can be separately adjusted at the timer of the actuator controls. The timing of 10 s operation and 20 s pause is quite feasible to allow water level settling.

– Variable speed operation: Frequency converters allow modification of the standard frequency of 50 Hz into frequencies of the often selected ranges between 30 Hz and 60 Hz. For example, if an actuator runs at 50 Hz with 45 rpm, it will run at 30 Hz with 25 rpm. When running at low velocity and consequently at low frequencies, the total running time must be considered which might not exceed the duration of 15 min in duty type S2 - 15 min.

– Combined operation: The possibilities to fulfil certain requirements are quite versatile. Stepping mode is often used. Sometimes three times, five times... including pauses across the stroke. The sluice gates can be softly started and stopped.

The following aspects are crucial for consideration:

– Dimension of the usable lock length. If the lock spaces have to be used up right until the gates, the inlet must be throttled.

- Initially, at low water level, the water inlet must be slower than when the water level increases. In particular, for large effective depth, the inlet quantity must be quickly increased.
- For small boats, the flooding must be thoroughly controlled.
- If further vessels are located at tailwater, the water drain must be checked attentively.

Baffle plates and bed drops are useful measures to adopt the inlet water jet and deviate the water downward. Respective equipment settings must be optimised by trials and documented within the operating regulations. Often brake motors are used for operating sluice gates in mitre gates or operating sluice gates in by-passes as previously described. Requirements are based on field experience and feedback. The necessary self-locking is decisive for selecting the required actuator-gearbox combination.

Opening and closing mitre gate locks

Mitre gates are opened in opposite direction of the water flow and increasingly against retained water. The necessary torque is particularly high. Start-up occurs softly at reduced speed. After opening, the required torque will reduce to 50 % and slighly increase prior to reaching the end position. After the soft start, the speed increases to the 50 Hz nominal value and drops immediately prior to reaching the end position to the adjusted soft stop value. The reduced speed allows draining the water retention between the gate and the wall to achieve the seating position without significant counter pressure. For closing, both gate leaves are softly approached and then operated against each other at constant nominal speed. To achieve a defined closing, one of the leaves could be operated first into the last 10 % of the opening at reduced speed until mid water. The second leaf remains at 90 % (**figure 4.17**) until the first leaf has reached its end position and is then operated either

Figure 4.17: Uneven closing of mitre gates

Figure 4.18: Bulb and tail seal at mitre gate

at 10 % reduction until the end position or only at 5 %. The remaining 5 % are closed by the water pressure against the vertical seal (**figure 4.18**). These settings are specific to the gate. Many solutions are feasible.

If none of the elements within the drive chain is self-locking – for example for actuators with speeds of 125 rpm and 180 rpm, lifting cylinders with ball thread or corresponding gearboxes and lantern gears – a sluice gate might be opened or closed by impact of wind. In such cases, brake motors are provided at the mitre gates.

Decisions in favour of brake motors are also taken to retain the second gate leaf (refer to figure 4.18) closely to the bulb and tail seal.

4.3.3.6 Notes regarding water level regulation

The frequency of required sluice gate operations is quite variable and hardly predictable. At one site, the level has to be changed twice a year, at other sites level changes can be more than one thousand times. Increasingly, complete sections of watercourses are centrally operated and complete control links are jointly managed, movement is made in very short steps. This can lead to increased operating temperature, adding up with the direct impact of the sun. For this reason, sun shades might be deployed (**figure 4.19**).

Figure 4.19: Sun shades mounted on actuators

If the operating scheme requires frequent readjustment, it is advisable to replace actuators in duty type S2 by modulating actuators in duty type S4. Depending on the size, they can start up to 1,200 times per hour. In the end, frequent starts are supposed to have a negative impact on the automation equipment.

The objective is to achieve the desired water level while operating as little as possible, favouring reasonable lifetimes for the actuators. The quality of the used control algorithm plays a decisive role.

The engineer responsible for calculating the modulating scheme must design the system so that the response does not result in an endless upsurge and downsurge around the set-point (**figure 4.20**) but approaches the horizontal level by immediate asymptotic solution.

Basically, two ways can be implemented:

1. To achieve a certain level, the upper gate of a double sluice gate is lowered in steps of respectively 3 cm, for example. First step, second step, maybe third step .... until achieving the expected results. The gate might even be required to be lifted up again. The steps are shortened.
2. The inlet flow is calculated on the basis of the level change speed (gradient) whereby the subsequent outlet flow must be included into the calculation. This is the basis for calculating the operation command.

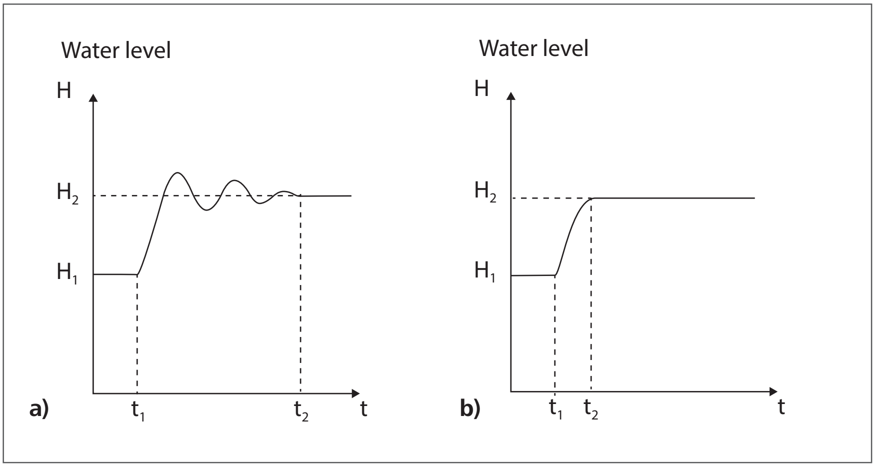


Figure 4.20: a) Step response with long settling b) Step response with optimum settling

The second way sounds good but is rather costly and often only implemented once the first solution is not sufficient. For the quality of both solutions, the following factors are decisive:

- High quality level measurement,
- High quality position transmitters,
- Optimum agreement between stroke and transmitter,
- Deploying actuators with infinitely adjustable speeds is not of prime importance.

Since all waterways react differently, the modulating behaviour of any site must be observed over a longer time period and parameters are to be adjusted.

4.3.3.7 Reducing the number of variants

To limit costs generated by

- Working effort,
- Training,
- Storage of spare parts and accessories,
- Provision of reserve actuators and gearboxes,

efforts are made to harmonise the set-up of weirs and locks. The following solution was successfully implemented:

- Lifting cylinders (**figure 4.21**) of one size used with just one actuator size can be used for a larger torque range if **retrofitting schemes** are provided.
- Actuators of a certain size can be easily modified for smaller or larger torque ranges by means of **retrofitting accessories**.

Figure 4.21: Actuator with lifting cylinder in basic sizes at mitre gate

At the end, only one size of lifting cylinders and actuators including various equipment accessories is required. However, standardisation is not always a prime focus. Cost comparisons are also important factors. However, if the actuator and lifting cylinder range cannot be limited to one basic size but two, three or four basic types, this is already a considerable advantage.

In this respect, the possibility to use an available reserve actuator or lifting cylinder to replace actuators or cylinders during regular maintenance and overhaul is quite appreciated. Further harmonisation of material deployed and spare parts available is possible and widely utilized.

To ensure that selected and proven equipment is also supplied for subsequent fittings, specifying the order number(s) of the already deployed products is always helpful when placing new orders. In turn, specifications can be drawn up with the manufacturers including all desired options, from the bronze worm wheel to the seawater blue paint. Maintenance schemes can be agreed and collateral technical documents supplied.

4.3.3.8 Electrical special solutions

- Sort starters compared to frequency converters

Soft starters are typically used to limit the starting current. We will not discuss further the possibilities of soft start and soft stop.

Soft starters work at constant frequency and speed. Voltage is reduced in time by potentiometer setting. This also decreases the starting current, however with an undesirable side effect: While the voltage is decreased, the torque reduction is square, for example $(0.8U)^2 \triangleq 64T$.

Frequency converters work with variable frequency and speed. They control motor voltage and frequency at a constant ratio. U/f = const. corresponds to Ψ (Ψ = magnetic flow, number of field lines occurring at surface A)

Apart from starting current limitation and variable speed, frequency converters offer a range of further assets like synchronisation control.

For soft starters, the minimised starting current reduces the actuator torque. They are reasonable in price but can only be deployed in case of sufficient actuator torque reserve. They have been largely replaced by frequency converters.

- Criteria for the use of frequency converters

The speed calculation of an asynchronous motor is based on

$$n = f * 60/p$$

wherein
n = Speed of rotary field
f = Frequency
p = Pole pair number
U = Voltage

The speed changes with the frequency.
In U/f operation, the converter controls both motor voltage and frequency at a constant ratio. This results in a largely constant torque without motor overload. Only when reaching 15 Hz, the torque is reduced to approximately 65 % of the stall torque which results in increased heat formation. This can be solved by using an actatuator with two lead worm gearboxes – the output speed then amounts to 125 rpm or 180 rpm – in combination with a brake motor retaining the self-locking effect.

The frequency converter operation of a pulling sluice gate bears the risk that the brake motor might accelerate excessively, Thus acting as a generator. To remedy this situation, frequency converters are equipped with so-called brake choppers deviating the excessive energy to a brake resistor where it is converted into heat.

In turn, if asynchronous motors are supplied by the mains, they possess a fixed speed based on the pole pair number and mains frequency.

For lower frequencies, noise developments are observed at motor level. The reason is the modified relative movement between cursor and rotary field leading to a rattling, separation and slaving of motor coupling.

For frequency converter operation, actuator controls must be externally supplied with 24 V DC for the reversing contactors.

Manufacturers of frequency converters list a maximum output current for 60 s in their technical data tables. Due to the low power factor cos φ, the current of the deployed pot-type motors must be 1.5 times higher than I_{max} higher of the selected actuator.

According to DIN 19704-1/8.3, the driving torque for electric motors controlled by frequency converters must be selected at least 5 % higher than the required driving torque. Impairing drive efficiency is caused by frequency converter operation.

- **Synchronous operation of connecting elements**

According to DIN 19704-2/9.2.4, synchronous operation of connecting elements with the risk of tilting must be monitored. For small and medium-sized weirs, commonly one actuator is used, which operates two gearboxes and consequently operating two connecting elements. For example, it is fixed to one gearbox. The connecting shaft is rigid, the low torsion to the second gearbox is neglected. In case of doubt, the shaft diameter selection is increased or more robust lantern gears are deployed in lieu of spindles. For large weirs, the actuator is placed in the middle of two gearboxes.

However, every set-up has its limits and imponderabilia as mentioned in chapter (2.1.5.1) often occur. With time or the impact of temperature, one side is subject to stronger braking than the other. As of a width of 25 m or 30 m, the loss of synchronisation can be excessive, the tilting of a shutter for example can be seen with the naked eye. To remedy, the **synchronous link feature** without mechanical connection can be used. The **mechanical shaft** is simulated electrically by two slip ring rotors. The stator windings of the motors are supplied with mains voltage. If the rotor of the transmitter motor is turned, the electrically linked receiver rotor strictly follows the transmitter rotor like if they were mechanically coupled. This is not easy to install and therefore rarely used for new sites.

In practical applications, the **electronic link feature** is preferred. Two actuators are controlled with identical commands via frequency converters, whereby the electric tolerance of both transmitters should be as low as possible. The mechanical tolerance which increases due to change in direction of rotation is decisive for the accuracy. One actuator can be programmed and act as **controlled master**. If it is ahead in the seating position, the **slave** detects the discrepancy and follows the master with increased speed. With other schemes, the faster actuator waits for the slower actuator in case of asynchronous operation. All known connecting elements, even the electrical lifting cylinder can be used for this scheme (**figure 4.22**).

Figure 4.22: Synchronous operation of two actuators on lifting cylinders with synchronous link

- **Using solar technology**

A large number of plants for civil engineering constructions for water applications are remotely located and have no power supply. Laying long cables would be very complex and the voltage drop compensation extremely costly. Furthermore, providers might limit the connection power in certain regions. For specific applications, manual operation might suffice or the use of a power generator be feasible. But safe power supply is still the preferred option. Why not use solar technology? Basically, all electric actuators are suited to be operated with solar power. Irrespective of the type of current: 3-phase AC, 1-phase AC or DC current at any voltage level. For voltages up to 48 V, no specific protective measures must be provided. From the user point of view, longer running times based on the need of gearboxes can be neglected. Eligible motors and gearboxes can be supplied. The customer's requirements are decisive. A consultant can select the ideal actuator and determine size and quantity of the accessories for providing sufficient storage capacity, even for several subsequent rainy days.

The following component groups are required for solar power generation:

- Solar panel (**figure 4.23**)
- Solar controller (**figure 4.24**)
- Battery charger
- Battery
- Chopper

Figure 4.23: Solar module with supervision camera and anti-climb guard

Figure 4.24: Site with solar controllers and modem

Together with the customer, the consultant can specify that:

- The actuators are operated via Profibus,
- The site is connected with the control station via modem,
- The control station issues operation commands and enables local operation on site if required, identifies and processes events like failures, burglary, vandalism providing photos of potential offenders.
- The actuators are interconnected via fibre optic cables to resist lightning.

However, the major drawback is the availability requirement of electricity during the winter months when floods might occur while it is cold and the sky is grey. A solar plant providing 200 W in summer may only be capable of providing merely 20 W in winter.

4.4 Electro-hydraulic operation

Feeding liquids under pressure into a **hydraulic cylinder** causes the piston rod to perform a linear movement. Sluice gates and lock gates are thus operated. Self-locking is achieved by solenoid valves. Locking elements which are electrically operated, can also work on hydraulic operation. Starting with a simple sluice gate (**figures 4.25** and **4.26**), to a

Figure 4.25: Hydraulic cylinder with local controls

Figure 4.26: Hydraulically operated sluice gate

Figure 4.27: Flotsam screen

flotsam screen upstream the inlet of a hydropower plant (**figure 4.27**) to a mitre gate lock (**figure 4.28**). In our example, the hydraulic medium under pressure is fed into **hydraulic motors** for generating a rotary movement operating the pinion for the lantern gear, the wedge roller gate deployed as by-pass gate and even the agitator as ice prevention system. Hydraulic motors and generators are located in the cabinet shown on the left.

Figure 4.28: Hydraulic site with lantern gears and by-passes

Figure 4.29: Hydraulic unit

The force required is generated by a **hydraulic unit** (**figure 4.29**). Biodegradable oil is used as hydraulic medium.

Figure 4.30: Hydraulic cylinder at a compression gate weir

Figure 4.31: Highly frequented hydraulic lock

Since closing element control can also be handled by electrical options and possibly more field experience is available, associations claim that they will only deploy hydraulic solutions after thorough investigation and on receipt of special permits. The prime reason is danger of leakage and consequently complex maintenance involvement.

However, hydraulic applications are still at the forefront when requiring synchronous operation of two actuators installed in very large weirs (**figure 4.30**), or if commercial navigation requires frequent locking (**figures 4.31** and **4.32**). When frequent locking is required, electric actuators reach their thermal limits faster than hydraulic actuators.

Figure 4.32: Mitre gate with hydraulic cylinder for gate operation

Figure 4.33: Large lock with hydraulic cylinders

Hydraulic solutions are virtually always favoured when high forces must be supplied on large waterways (**figure 4.33**).

A gate might weigh up to 50 t (**figure 4.34**). The own inertia and the considerable water retention must be overcome at short notice.

Figure 4.34: Large lock gates ready for overhaul

Figure 4.35: Radial gate with tension gate arms as upper lock gate

The use of pivoting segments (**figure 4.35**) are typically unpopular, due to their size, their enormous weight and the resulting difficulties in terms of transport, mounting and the complex vertical and horizontal sealing requirements. In hydraulic applications, they still play a crucial role. However, the necessity to install underground service rooms must be pointed out.

During normal operation, the closing segment is placed in the sil recess and is in vertical position when closed. The pivot mounting (**figure 4.36**) comprises the hydraulic cylinder – stopped at the left – offering a hydraulic blocking facility for two positions to the right and at the right a hole to relocate the cylinder in revision position. Then, the closing segment lies horizontally above the water level.

Due to these and other considerations, the following typical advantages and drawbacks for hydraulic actuators compared to electro-mechanical actuators can be made. Further reading is available in brochures and specific literature.

- Advantages of hydraulic actuators
  - Infinite speed adjustment of output drive within a large range
  - Easy reversal of operation direction
  - Simple generation of large forces and torques with high efficiency
  - Excellent synchronous operation properties
  - Long lifetime, since the fluid is self-lubricating and can also be used as cooling medium
  - High positioning accuracy
  - Start from standstill at full load
  - Underwater use at submerged gates does not require long connecting elements. Biodegradable oil is a must.

Figure 4.36: Pivot mounted radial gate with hydraulic cylinder in an underground service room below the lock

- Drawbacks of hydraulic actuators
  - Risk of leakage and environmental pollution
  - Complex maintenance actions
  - No mechanical self-locking; achieved by solenoid valves
  - Protected cabinet for hydraulic unit is required
  - High control efforts

For the before mentioned advantages, consideration must be made that electro-mechanical actuators have already caught up. They can now work with electronics, frequency converters and thermal stable motors with forced cooling. This allows:

- Start from standstill at full load
- Extremely precise measurements, high positioning accuracy
- Good synchronous operation properties
- Infinite speed adjustment, for example soft start, soft stop
- Depending on the make, good continuous duty properties
- Mechanical self-locking

Electro-mechanical actuators are compact in design, virtually maintenance-free and installation times are short. When combining with gearboxes, nearly any force and torque requirement can be fulfilled. Frequency converters allow to overcome inertia of large radial segments.

Previously, there was a large number of advantages in favour of hydraulic actuators. Today, the capability of providing suddenly large forces across long durations is the main argument. Decision on implementing hydraulic or electric actuators must be taken depending on the application and conditions.

4.5 Auxiliary operations

- **Manual operation**

When electric or hydraulic auxiliary power supplies fail, the site should always provide an operation facility via handwheel or crank handle.

- **Power tool for emergency operation**

Due to the often large strokes in civil engineering constructions for water applications and large reduction ratios at handwheel or crank handle for whatever reason, many revolutions are required for operation. This requires a lot of force and is very time consuming. For this reason, electric power tools are the perfect substitute. You may use a drill with distortion lock (**figure 4.37**).

In areas with unstable grids where power failures often occur, manual actuator operation is not very efficient. The search for alternatives suggest the use of power tools. For this reason, actuator manufacturers offer a square adapter for power tool operation (**figure 4.38**).

Figure 4.37: Drilling machine with distortion lock

Figure 4.38 Actuators with red motor-manual operation change-over lever and square for power tool operation

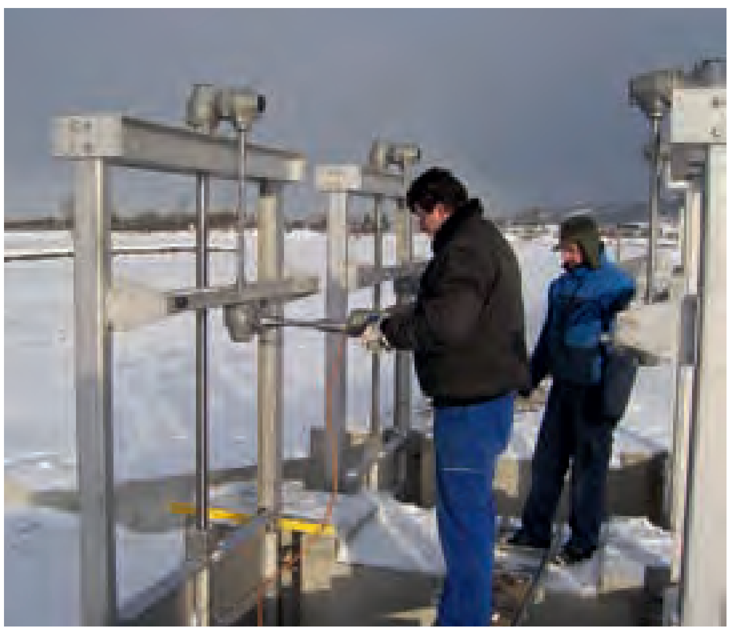


Figure 4.39: Power tool operation at bevel gear-boxes

The power tool is placed onto the input of the actuator or the gearbox (**figure 4.39**). The gearbox input shaft might be round or possess a straight or tilted square. The direction of rotation must be selected. The counter must be set to zero. Torque and speed must be set to a minimum and increased slowly. To prevent accidents, the maximum speed of 300 rpm must not be exceeded and even reduced prior to reaching the end position.

- **Operation for relocation and alleviation**

For many years, low cost actuator technology was deployed to automate large sites. An actuator can be fitted from one sluice gate to another sluice gate of same design, without requiring any changes or adjustments. This was also rationalised by installing a transport rail to save hoisting technology and human force (**figure 4.40**). However, this has never achieved acceptance and is no longer provided.

Figure 4.40: A rail actuator operates 16 sluice gates

- **Emergency power supply**

Emergency power supply by batteries is scarcely used. Besides 400 V master actuators, 12 V or 24 V slave actuators can only be used for secondary functions (**figures 4.41 and 4.42**).

- **Self-sufficient power supply**

In remote locations with little energy provision, a transportable diesel or gasoline generator can supply voltage of 3 x 400 V for example (**figure 4.43**). When selecting the generator, the maximum output current for 60 s should be considered (refer to chapter 4.3.3.7 Frequency converters).

Figure 4.41: Emergency power supply 24 V DC

Figure 4.42: Actuator with emergency power supply 24 V DC

Figure 4.43: Gasoline generator

The voltage can be applied to the plug/socket connector of an actuator or to the control cabinet to simultaneously supply several actuators (**figure 4.44**). The control cabinet installations ensure the specified fuse protection.

Figure 4.44: Control cabinet with voltage supply via generator

- **Safety power supply**

For safety reasons, some waterways or sections are supplied independently of the conventional power grid by a separate voltage supply. This deals with an **emergency power supply via standby power supply system**.

5 ACCIDENT PREVENTION

Within the framework of the **EC Machinery Directive** 2006/42/EC, sluices and weirs are considered as machines. For this reason, the plant operator must provide a risk assessment for all sites, documenting potential areas of danger and actions for hazard prevention. Turning and shifting parts of a machine or a system must be secured against accidental or intentional intervention (**figure 5.01**).

Figure 5.01: A precursor in sluice gate actuation technology

This is particularly important if the object is easily accessible via a public road, for example (**figure 5.02**). Vertically shifting lantern gears are furthermore covered by tubes against intervention. Simultaneously, inserting sticks or similar objects into the pinion area is more difficult.

In the following example, one cover could be used to house two lantern gears of a double gate weir (**figure 5.03**).

Figure 5:02: Weir with lantern gear protection tubes

Figure 5.03: One cover protects respectively two lantern gears

Figure 5.04: Actuator technology of a double sluice gate with open and accessible shafts

Turning shafts are particularly dangerous. Especially, since the site is freely accessible from all sides (**figure 5.04**).

Figure 5.05: Weir with covered connecting shafts

Figure 5.06: Cable winches installed in telescopic ducts

The shafts must be securely and reliably protected using tubes or cover plates (**figure 5.05**). Upward and downward spindle protection was discussed in detail in section 2.1.10 “Spindle use”.

Ropes are tightened, meaning they are wound and unwound and must be installed inside ducts (**figure 5.06**).

6 PROTECTION AGAINST UNAUTHORISED USE

Hydraulic steel structures are located at rivers, channels and other waterways. Sometimes at sites with public bridges offering direct access, sometimes in forests and meadows in the middle of nowhere. It might be possible to fence some parts of the sites, but not all of it. For this reason, clearly visible warning signs are erected and video cameras are installed. This only frightens away peaceful and interested people.

To prevent vandalism, actions were taken. Unauthorised users like to turn the handwheel for example. After a few tries, they learn how to activate manual operation. And the intruder can operate the weir. Water features can be heard and seen. Lever operation can be inhibited by using a special lock and the handwheel operation is deactivated and only runs idle (**figure 6.01**).

Figure 6.01: Lockable handwheel

To prevent incidents with more tenacious “testers”, it is advised to enclose the actuator combination (**figure 6.02**). This does, of course, not deter graffiti – see installation in the background.

Lockable protection covers are available to mask local controls of AM/AC actuator controls. For reasons of precaution, the actuator handwheel is removed (**figure 6.03**).

Figure 6.02: Completely housed actuator – bevel gearbox set-up

Figure 6.03: Lockable protection cover at AUMATIC local controls

Figure 6.04: AUMATIC local controls without selector switch

At times, the selector switch for local or remote control is not required (**figure 6.04**). As a matter of fact, the on-site engineer must contact the control station by phone to enable OPEN-STOP-CLOSE commands via Profibus, for example. AUMATIC parameters may only be changed after password entry.

Figure 6.05: AUMA MATIC actuator controls on wall brackets in a lockable control cabinet

If this still does not suffice, the whole set-up can be locked away. Actuator controls can be mounted on special wall brackets and hosed into a lockable steel cabinet or a nearby building from which an operation can still be controlled (**figure 6.05**).

Since the colour of the LEDs signal end positions, like open and closed or active operation or faults, it is better to have visual access to the actuator controls display (**figure 6.06**). The AUMATIC display provides even further information.

Figure 6.06: AUMA MATIC local controls behind lockable glass doors

A further option to protect the site from unauthorised use is the assignment of lockable service entry into each weir segment (**figure 6.07**). A socket can also be installed within the cabinet.

Figure 6.07: Service entries for two weir segments

Figure 6.08: Repositioning local controls

A portable local control unit can be assigned to the weir to be operated as required (**figure 6.08**).

Figure 6.09: Lubricator safeguard

Figure 6.10: Artistic water bed decoration of a site

The shown examples deal with electrical issues. However, protection against vandalism must also be considered on a mechanical level. Experience has proven that if intruders cannot manipulate, they might destroy or remove items. This is particularly the case in times of high scrap metal prices. It is very annoying when grease cups are stolen. Are the perpetrator aware of the consequences? Constructors of hydraulic steel structures or operators sometimes react as shown in **figure 6.09**.

All these additional safety measures are cost and time intensive. And above all, their efficiency cannot be proved. What is the chance for art in this instance? Let's just make our site look beautiful! (**figure 6.10**).

7 LANDMARK PROTECTION

The Landmarks and Historic Buildings Protection Act stipulates: “Country’s historical heritage must be protected and preserved. Potential dangers are to be avoided.” In civil engineering constructions for water applications this refers mostly to weirs and locks. Our beautiful shutter weir already provides the lever technology (**figure 7.01**). A hole-punched rail allows reselection of previous settings. Even if the original site was slightly modified, it is still worth preserving its original condition as much as possible by performing appropriate repair work.

Figure 7.01: Ancient sluice gate weir with lever actuation

The walls of a mitre gate lock were erected in a straight line of sandstone. At the time, the gates were made of wood. And on top of the hill, the castle is overlooking the site! Considering this scenery, it becomes clear that for reconstruction, same materials and identical processes should be used (**figure 7.02**).

Of course, a weir does not require a roof (**figure 7.03**). Maybe, the builder provided the roof to protect the bridge and to allow guiding the lantern gears. In any case, best endeavours are made not to change the original appearance.

Figure 7.02 Wooden mitre gate in sandstone chamber, original condition was restored

Figure 7.03: Sluice gate weir with roof

Often, the original condition is made at least visible to demonstrate previous function principles (**figure 7.04**). This type of slatted drums were initially used to wind up chains

Figure 7.04: Sluice gate weir with ancient chain hoist and modern rack and pinion hoist

of sluice gate weirs. Curtain weirs have a similar appearance. Today, the sluice gates are operated with pinions and the laterally arranged toothed racks.

If the site was operated manually, actuators with back-up energy should not be visible after retrofitting. For this reason, they were hidden inside buttresses (**figure 7.05**). We have to pity the service technicians!

Figure 7.05: Actuator hiding place inside the hollow buttress

Figure 7.06: Compromise between automation and landmark protection

Ancient sites should be preserved and rebuilt to their original condition. However, automation cannot be completely banned since the technology is to be presented to an interested public in a museum (**figure 7.06**). The manual gear, being very cumbersome to operate, was replaced by a faster electric actuator. Aware of potential dangers, no efforts were spared to secure the site against unauthorised intervention.

Such an ancient site might also be of particular interest to apprentices (**figure 7.07**).

Figure 7.07: Lantern gear system in a training room

Figure 7.08: Manual drive for a double sluice gate with pinions and toothed racks

Today, purely mechanically driven double sluice gates are no longer in use today (**figure 7.08**).

Interest on technical landmarks is widely spread. It is not possible to preserve all ancient steam trains, paddle steamers, and shaft towers. But some are worth saving. This is the way of thought of constructors for hydraulic steel structures. An important element of a site is reliable operation over many years and this is presented and proudly showcased.

Figure 7.09: Roller gate weir with gate shield on top and gate shield with sealing strip at the bottom, cutout

Where to identify better the stiffening of a roller gate weir (**figure 7.09**)?

Figure 7.10: Bell-type gate

Bell-type and cylindrical gates are still widely used. They are easy to operate since the counterweights – perfectly visible – support the movement (**figure 7.10**).

8 CONSTRUCTION, OPERATION AND MAINTENANCE

Submission of quotations or bids

All stakeholders with experience in civil engineering constructions for water applications know the standards and the literature for finding advice and information. The basic requirements are well known. One prerequisite is the submission of a **Declaration of incorporation** or the **"Proof of suitability certificate** according to DIN 18 800-7 (DIN 19704-2/5.1) with the supplement for manufacturing welded steel structures for loads which are not predominantly static". Furthermore, according to the Machinery Directive, proof must be provided for the **CE mark**. When fulfilling these prerequisites, proof of **bidder suitability** is given. The person in question is **pre-qualified** according to **VOF**.

The applicant will satisfy the demands of the tender in words, calculation and drawings. Very often, clear specifications of the task are not available. He/she must issue drawings, calculate and evaluate pricing knowing that the customer must verify the submitted bid. If necessary, **the applicant raises concerns**. There is no need to agree with all elements of the bid. Besides the **main bid**, a **secondary bid** can be submitted if approved by the customer. However, the suggestions for implementations must be explained and detailed within the **product specification**. With the support of the consultant, the customer will evaluate all bids and select the most favourable variant and award the order for implementing the steel structure building work to the most appropriate applicant. Constructive definitions are made in **works planning**, based on an existing **execution planning** while performing measures on site.

Technical processing

The applicant selected for the steel construction must submit all "documents for technical processing" to the customer four weeks prior to placing the order. This includes:

- Constructional details
- System and static calculations
- Strength calculation
- Welding test plans and welding sequence plans
- Workshop drawings, parts lists
- Corrosion protection schemes

The electrical part is separately processed.

Manufacture of components

Depending on requirements and possibilities, the following elements are pre-manufactured in the workshop:

For a weir:
- Frame, closing element, inspection gates
- Machine bridge (**figure 8.01**)
- Connecting elements such as spindles and lantern gears
- Spindle nuts or pinions

Figure 8.01: Workshop intervention: Machine bridge

For a lock:
– Gates
– Sluice gates such as wedge roller gates or sliding gates
– Inspection gates

The following must be procured:
– Actuators, gearboxes or lifting cylinders
– Slide and sealing material

The further sequence from the workshop until final work inspection by the customer is as follows:

1. Workshop tests
The necessary tests are performed in the workshop. For example, steel sheets are examined and weld seams x-rayed. Films, x-rays and evaluations are part of the as-built documents to be supplied to the customer. **Inspection certificates** must comply with DIN EN 10204.

2. Transport
Depending on the size, hydraulic steel structure parts transport might require special vehicles and special authorisations as well as respecting certain off-times. It is quite common having to apply for traffic directions to perform assembly on site.

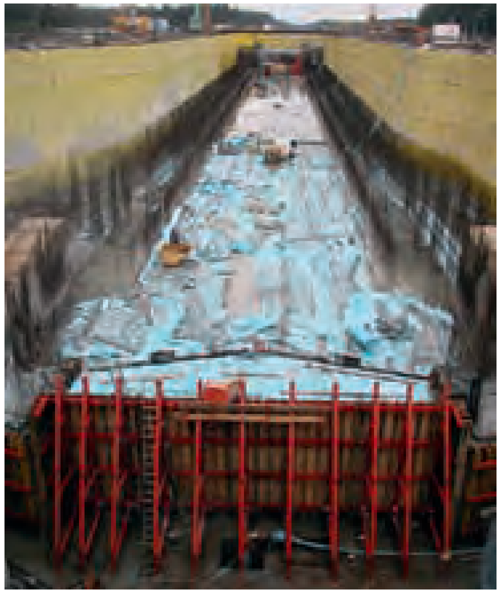


Figure 8.02: Lock building site.
The concrete work is being prepared

3. Assembly
The earthmoving and excavation work is finished. Now concrete work commences (**figure 8.02**).

Figure 8.03: Mounting the closing elements avoiding constraint forces and leakage

Figure 8.04: Prepared actuators and lifting cylinders

Recesses for sluice gates and inspection gates have been provided on site. Now the assembly work is starting (**figure 8.03**). "The closing elements of the weir or lock must be operable and free of constraint forces across the whole travel or swing angle." [5].

Actuators and connecting elements are prepared (**figure 8.04**) and mounted (**figure 8.05**).

Figure 8.05: Last actions for actuator installation

According to DIN 19704-2/9.2.1, actuators must be easily accessible. Intervention access to constructions parts requiring setting and/or regular maintenance, e.g. switches, transmitters, lubrication points must be straightforward.

4. Commissioning

After mounting the actuator-gearbox combination and respective wiring, the closing element can now be operated using the actuator handwheel. The closing element must be operated out of the end positions. In this position, the electric actuator can be started. The selected running direction must be checked – for example: "OPEN" – to make sure that the actuator actually travels into the selected direction. Otherwise, the pertaining stop button cannot be used to switch off the actuator. In this instance, the rotary field would have to be inverted. Actuators equipped with AUMA MATIC actuator controls automatically correct the direction of rotation. After pressing the stop push button, the actuator stops (**figure 8.06**).

Figure 8.06: Connecting AUMATIC actuator controls

For implementation, the correct match of open and close must be observed. The terms open and close are specifically defined in the valve industry. If a globe valve is closed, flow is stopped. To perform this in the same way as the globe valve but using a double sluice gate, the lower gate must be operated downward and the upper gate must be operated upward. To avoid mistakes, constructors for hydraulic steel structures speak of lifting and lowering with regard to weirs and locks. For actuator manufacturers, the most specific terms are clockwise and counterclockwise operation.

The required change of direction for spindle actuators can be achieved by selecting the trapezoidal thread LH (left hand) or RH (right hand) and for lantern gear drives by selecting the worm gearbox in LR or RR, respectively RL or LL (2.1.17 “Lantern gear application”).

Now, end positions have to be set. A sluice gate can be lowered electro-mechanically until the seal contacts the bed beams. The fine tuning to ensure slight pressure on the gasket can be made using the handwheel. This is the position for setting the end position switch to “closed”. The torque switch “close” which is active across the whole stroke will trip before reaching the setpoint in case of an obstacle, or later (if the end position switch is not wired) when reaching the preset closing torque.

Then, operate the actuator in direction Open and set the end position switch “Open” at the point of maximum stroke. The torque switch “Open” is ready to operate across the total stroke.

Mechanical and electrical position indicators are to be aligned with the “Open” and “Closed” positions of the closing elements.

Speed and type of duty must already be observed when performing the settings. For low speed and S2 - 15 min, only a partial stroke might be managed. For covering the total stroke, possibly type of duty S2 - 30 min and a higher speed would be required.

Differing specific features relating to locks are discussed in chapter 4 “Opening and closing mitre gate locks”.

5. Dry run

During dry run, slide and sealing strips must be protected using environmentally friendly lubricants. For setting end position switches, perform one complete operation from “Closed” to “Open”. This stroke can be used for dry testing. Pay attention to any potential running noises or possible current peaks. The test run includes running from any position into any direction comprising intermediate stops. Tripping of electrical protective equipment must also be checked.

Tight contact of slide rails and the pretension of sealing strips must be verified. Leakages can be detected using a feeler gauge. The efficiency of the mechanical locking mechanism must be proved. This means, that when removing a gearbox, lugs or hooks must allow securing the closing element.

Grease cartridges must be mounted and their running time be set. Efficiency of lubrication must be verified.

Customers increasingly require the proof of buckling resistance of connecting elements. This means closing until reaching the end position and torque seating. When placing a wooden block below the connecting element as part of the test, the consultant for civil engineering constructions for water applications might hesitate. The spindle must now withstand $F_{C+25\,\%} = 2 * F_{CCE} \cdot * 1.25$ (whereby F_{CCE} = closing force per connecting element)!

Once this was successful and any basic problems have been remedied including the documentation of small defects for later elimination , functional tests in water can be performed.

6. Tests in water

Leak tightness of seals have to be proved for the impact of one-sided hydrostatic pressure **(figure 8.07**). Possible subsequent tests have to be considered. The performance of closing elements needs to be verified. For weirs, particular attention is paid when opening at maximum water pressure and when closing at maximum undercurrent. Attention must also be paid to potential vibration across the complete stroke and remedied or limited by introducing appropriate measures (2.1.2.8). Just prior to closing – in case of high flow velocity – **cavitation damage** [5] might additionally occur. This becomes visible by strong blistering which will lead to **cavitation damage**, and finally to material abrasion. During later operation, positions incurring the previously mentioned impacts are to be avoided.

7. Test run

The whole site must be operated via local and remote control during a defined time span. Also heed running-in periods of new parts like worm gearboxes (refer to manufacturer's specifications).

Figure 8.07: Tests in water of a lock

8. Final inspection and commissioning
The final inspection is based on successful test operation. The customer inspects the completeness of supplies and the fulfilment of specifications and tenders of the construction. Inspection reports for material, dimensions, welds, corrosion protection and successful functional tests will be submitted to the customer. Manufacturer's declarations for actuators and gearboxes will also be part of the collateral documents.

After this, the site will be commissioned officially (**figure 8.08**).

Figure 8.08: Water on!

Depending on the agreement, the constructor for hydraulic steel structures will have to intervene in case of failures, maintenance and warranty claims. If everything works to plan and the customer is content, follow-up orders for new sites might be considered.

After site handover, the following points are of particular interest:

- **Operating regulations**

The operator of the steel construction site will issue operating regulations. They will include the descriptions and documents for use of the site supplied by the manufacturer (who will also supply the Declaration of Incorporation according to EU directive) such as:

– System documentation
– Operation and maintenance instructions
– Instructions for inspection and corrective maintenance

and site-specific instructions like:

- Behaviour guideline in the event of danger
- Discharge control specifications
- Operating log.

Descriptions and documents of use must be at least stored at the builder's yard. The second part, the operating instructions, must be stored on site or with the operator.

Maintenance

Maintenance starts with diligent and responsible operation. The site including all components must be handled with care. When inspecting the site, pay attention to clean, smooth, noiseless and vibration-free operation of mechanical parts. During visual inspection, potential grease leakage and corrosion are to be assessed and recorded. If sites are not used very often, functional tests should be performed at least once a quarter at appropriate partial stroke. In particular slip clutches are subject to failure when operated infrequently. They should be tripped in periodically, defined intervals and adjusted again, to keep the value of the set tripping torque as constant as possible. "Since this type of inspections induces wear, it is recommended to reduce the setting torque for the inspections" [5]. Parts subject to weir like connecting elements, spindles, lantern gears, ropes, chains, spindle nuts and pinions are to be greased at least during maintenance and in compliance with the manufacturer's specifications and local experience.

Since many tasks are versatile and complex at the same time, service contracts are often concluded with the manufacturers. This generally includes enhanced warranty – in civil engineering constructions for water applications typically 4 to 5 years.

Figure 8.09: Stop log weir below the lift gate of a lock

Inspections are to be performed at safety-relevant intervals. Generally, locks are to be put out of service every five years, weirs however every ten years.

Depending on the severity, the sites might have to be run dry. A possible option for blocking the water could be an upstream stop log weir. For inspection gates, steel or aluminium elements are used today instead of their predecessors made of wood.

In one lock, the confined space below the lift gate must be used (**figure 8.09**).

A needle weir is typically used if sufficient space is available (**figure 8.10**). The steel pipes are individually placed on site.

Figure 8.10: Blocking a shutter weir by a needle weir consisting of metal pipes

Many installations are already equipped with guides for inspection gates for dry running of the site. The locking equipment is often available and ready for use at the site (**figure 8.11**). The mission of such equipment is definitely required when closing elements as well as slide and sealing strips are to be inspected.

In the following case, a sector weir has to be lowered and run dry using the laterally placed stanchions with inspection gate panels (**figure 8.12**).

Figure 8.11: Inspection gates

Figure 8.12: Laterally placed inspection gates protect a sector weir

Figure 8.13: Two mounting lugs – lower centre of the picture and counterpart – allow mechanical locking of the segment gate

Actuator inspection covers the complete power train. The power drive from the actuator to a possibly available multi-turn gearbox, couplings and bevel or worm gearbox across connection equipment and elements right through to the closing element. Particular attention must be paid to running noises, heat development and grease leakage. When working on

Figure 8.14: Lockable hook (lower centre of the picture) to maintain a lock lift gate

Figure 8.15: Locking bolt at a roller gate

these components, just like working on the hydraulic steel structures, the closing element must be positively locked above the potential high water mark (DIN 19704-2/3.3.4) (**figure 8.13**). Likewise, a sluice gate (here a lift gate) must be kept outside the extreme high water mark (**figure 8.14**).

Locking bolts at the closing element are typically used as counterparts (**figure 8.15**).

Actuator revaluation by inspection

If the number of possible output drive shaft revolutions as shown in table 2.04 in chapter 2.1.4.4 are expected to be exceeded in 35 years, inspections can be agreed by the customer already during the planning stage. According to DIN 19704-1/9.5.3, the duration of use of wear parts can be defined by the customer. Inspection must be scheduled at suitable intervals, for example, every five years and included within the tender. This should anticipate any later financial issues. Inspection typically includes actuator overhaul including lubricant change and seal replacement and even changing the actuator gearing and actuator bearings if this is required. To make sure that the inspection intervals are respected, this particular scheme must imperatively be included in the operating regulations.

Maintenance

An actuator should be subject to maintenance every ten years [5]. Depending on load, components must be replaced or at least the lubricant changed. Replacement of seals and O-rings are imperative. Screw connections, in particular cable glands, must be fastened again. If necessary, corrosion protection has to be restored.

Figure 8.16: Removal of reed and debris

Lifting cylinders must be overhauled in the factory after ten years of mission.

As a matter of fact, operators must not forget ambient conditions having to allow smooth operation of the components. This way, the closing section at the water bed must always be free of bed load. Depending on the situation, cleaning is necessary once a quarter.

Floating debris including grass and flotsam are unfavourable for spindle nuts. Even when considering lantern gears as robust, the weir catchment area should be kept as clean as possible (**figure 8.16**).

9 RETROFITTING OLD SITES

The previous paragraph discussed the production and maintenance of steel constructions for water applications. In spite of comprehensive maintenance and corrective maintenance actions, modernisation and retrofitting might be required. Even when perfectly operating, installations might no longer meet the operator's requirements. They are to be replaced – redesigned – or at least adapted – retrofitted.

Retrofitting means to adapt to new conditions. It is rather seldom that existing sites are completely redesigned since most waterways preserve their typical behaviour, unless there is human intervention and channels are built or waterways redirected.

Why is retrofitting necessary? Meanwhile, most of the original hydraulic steel structure constructions have disappeared (**figure 9.01**).

Figure 9.01:
Former: Open worm gearbox with pinions and toothed rack for manual operation

Pinions, worm shafts and worm wheels are open and can easily be lubricated, but this is where the risk lies. Accidents cannot be excluded. Oil or grease might drip and pollute the water. Operation requires a handwheel or a hand crank.

Figure 9.02: New: Enclosed worm gearboxes and actuators for motor operation

In modern sites, turning and lubricated parts should be enclosed to prevent accidents and preserve the environment. Worm gearboxes (**figure 9.02**) as well as bevel and spur gearboxes function on this principle. To improve the functional impact of the weir, the single sluice gate was replaced by a double sluice gate. Instead of the combination between toothed rack and pinion, lantern gears and pinions with robust lantern gear toothing are state-of-the-art.

Manual operation is another issue. Of course, modernisation of manually operated sites is made, maybe to give more stability to the site or to change the reduction ratio. However, manual operation can be very consuming in both, force and time. Therefore, auxiliary energy is preferred. Mostly, electric energy thanks to its reliability. The actuators deployed are enclosed just like the gearboxes. For operating the adapted weir, a lockable control cabinet with reversing contactors and buttons is erected next to the site.

Another significantly larger weir which was initially manually operated used non-rising spindles. The sensitive spindle nuts – attached to the sluice gate – were often submerged in dirty water and shortly after commissioning, they were relocated below the bevel gearboxes to allow the spindles to pass through (**figure 9.03**). Thus, the problem was remedied but this was not enough. The weir shown had to be completely modernised at a later stage.

For the new bevel gearboxes, the option was used to lead the rising spindles within stem protection tubes (**figure 9.04**). The actuators were placed at the centre to keep torsion on both sides at the same level. Slip clutches for protection against excessive torques were

Figure 9.03: Former: Weir was not working as desired after first retrofitting

integrated into the horizontal synchronous operation shafts at both sides of the actuator. The turning shafts were enclosed to avoid any potential accidents.

Figure 9.04: New: Weir with electro-mechanical operation via Profibus

Figure 9.05: New: The small control room

Electrical operation was a major requirement for the operators. The AUMA MATIC actuator controls were installed within the small and lockable control room on site, thus protected against vandalism. For better accessibility, they were mounted on wall brackets (**figure 9.05**). The installation is connected to the control station via Profibus, a modem and GSM.

The next relatively new weir is quite unusual (**figure 9.06**). A person would have to stand in the middle with two long arms to simultaneously turn the handwheels. Or perform one revolution on the left handwheel and one revolution on the right handwheel. Even better, two people could turn the handwheels at the same time.

Figure 9.06: Former: Sluice gate weir with hand-operated spindles

But the operator did not really enjoy the set-up of the site. The installation was considered as difficult to operate. In comparable sites, the situation is quite similar. Often, less consideration is given to manual operation. In fact, it is rather a nuisance to pay much attention to such a small weir. Consequently, a pendulum stem nut used to compensate the lack in synchronism is not an option.

The successive plant operator had decided to replace the sophisticated and delicate spindles and to deploy robust lantern gears. The pinions are completely enclosed. The electric actuator adapted to the worm gearboxes will satisfy the customer (**figure 9.07**).

Figure 9.07: New: Sluice weir with electro-mechanically operated lantern gears

The small mitre gate lock was adapted in the 1960s and has provided perfect service ever since. The actuators worked according to the electro-mechanical principle, the worm gearboxes were hidden but not enclosed in their housings (**figure 9.08**), so they lacked tightness against oil and grease loss. The worn pinions, shafts and bearings had to be individually manufactured which was costly. Although repainting gave the installation a better look, finally this generation had to be retrofitted again.

From the hydraulic steel structures point of view, there was a lot to do. Gates and sluice gates had to be disassembled and overhauled, seals replaced and fender beams were provided (**figure 9.09**). Compact electro-mechanical actuators were mounted together with lifting cylinders. For operating the sluice gates, brake motors were provided. Gates

Figure 9.08: Former: Mitre gate lock with enclosed actuators and gearboxes

Figure 9.09: New: Mitre gate lock with automated gates

and sluice gates are controlled via frequency converters ensuring soft start and soft stop. An external timer allows setting operation and pause times of sluice gates. The automated self-service lock is part of a larger lock group.

Reasons for operators to retrofit weirs and locks:
1. Better solution for water construction than previously possible
2. Environmental protection
3. Improved operability
4. Increased safety against failure, accidents, and vandalism

The procedure for retrofitting does not significantly vary from redesigning the site. The chain of customer – consultant – constructors for civil engineering constructions for water applications is still the same. Product requirements and tenders need to be issued. Available calculations and assessments can be used as basis. Typically, experienced planners and constructors for civil engineering constructions for water applications are awarded the order. Each customer feels better when reference customers and schemes of other building sites are available and if the feedback from other customers is positive.

EPILOGUE

The goal of this book is to provide comprehensive technological and practical information and expertise to contractors for hydraulic steel structures, consulting engineers and consultants. This was paired with the desire to ensure that all participating planners achieve similar actuator configurations for one specific automation requirement, including the specification of all relevant parameters to the number of subsequent strokes and the necessary operating times per stroke. The book discusses these issues in detail. The major expert information has been covered.

Furthermore, all relevant impacts were considered and evaluated. It occurred that due to their complexity, force and torque calculations are quite versatile and sometimes can only be solved by further investigations and with the support of computer-aided calculations. Often, DIN 19704 only provides recommendations and refers to model tests. Experienced consulting engineers select different ways.

Although constructors for hydraulic steel structures achieved similar results for calculating the resulting impact on the machine construction, based on their experience and consideration for potential difficulties, it occurred that actuators and gearboxes were sized too weakly, considering that they have been scheduled for a total deployment period of 35 years.

This book now offers valuable support and expertise to achieve optimum actuator and gearbox selection when consequently used.

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MAJOR FORMULA SYMBOLS

Closing element dimensions [m]

l = Length of sluice gate
h = Hight of sluice gate
w = Width of sluice gate
w_{GL} = Width of gate leaf
w_{Bar} = Width of bar
w_{A} = Width of ark

Force [N]

F_{H} = Hydrostatic force
F_{HR} = Residual hydrostatic force
F_{h} = Horizontal force
F_{v} = Vertical force
F_{SP} = Seal preload force
$F_{SP/l}$ = Seal preload
F_{FAH} = Hydrostatic frictional application force
F_{GRH} = Hydrostatic guide runner friction force
F_{BH} = Hydrostatic buoyant force
F_{BWD} = Buoyancy by water displacement
F_{WL} = Water load
F_{IL} = Applied ice load
F_{Ice} = Ice force
F_{Pull} = Pull force
F_{W} = Wind force

F_{O} = Opening force general
F_{O1} = Opening force when new
F_{O2} = Unseating force after longer standstill times and wear
F_{OCE} = Opening force per connecting element
$F_{OCE60\%}$ = Opening force for unbalanced load per connecting element

F_{C} = Closing force
F_{CCE} = Closing force per connecting element
$F_{CCE60\%}$ = Closing force for unbalanced load per connecting element

$F_{ØEHW}$ = Mean force for EHW (extreme high water)
$F_{ØMHW/MW}$ = Mean force for MHW/MW (mean high water/mean water)

F_{Ha} = Manual force
$F_{HaOStandard}$ = Manual opening force for standard handwheel
$F_{HaOOption}$ = Manual opening force for optional handwheel

Torque [Nm]

T	=	Torque
T_{max}	=	Maximum required torque
T_A	=	Actuator torque
T_{Amax}	=	Maximum actuator torque
T_{AO}	=	Actuator opening torque
$T_{AO+25\%}$	=	Actuator opening torque +25 %
T_{AC}	=	Actuator closing torque
$T_{AC+25\%}$	=	Actuator closing torque + 25 %
$T_{sluice\ gate\text{-}EHW}$	=	Sluice gate torque curve for EHW
$T_{sluice\ gate\text{-}MHW/MW}$	=	Sluice gate torque curve for MHW/MW
$T_{A\text{-}ØEHW}$	=	Mean actuator torque for EHW
$T_{A\text{-}ØMHW/MW}$	=	Mean actuator torque for MHW/MW
$T_{Duty\ type}$	=	Duty type run torque
$T_{Lifetime}$	=	Lifetime run torque
T_{GK}	=	Lifetime run torque of bevel gearbox
$T_{GKØ}$	=	Mean torque at bevel gearbox
T_{GS}	=	Torque at worm gearbox
$T_{GSØ}$	=	Mean torque at worm gearbox
$T_{GS\text{-}HSS2}$	=	Lifetime run torque of worm gearbox for HSS2
R_A	=	Actuator revolutions
R_A/stroke	=	Actuator revolutions per stroke
R_A/35y	=	Actuator revolutions in 35 years
R_{AHSS2}	=	Actuator revolutions for HSS2
R_{GK}/stroke	=	Bevel gearbox revolutions per stroke
$R_{GK\text{-}HSS}$	=	Bevel gearbox revolutions at run torque
$R_{GK\text{-}nominal}$	=	Bevel gearbox revolutions according to load
R_{GS}/stroke	=	Worm gearbox revolutions per stroke
$U_{GS\text{-}HSS}$	=	Worm gearbox revolutions at run torque
$U_{GS\text{-}nominal}$	=	Worm gearbox revolutions according to load

General parameters

μ_0	=	Static friction coefficient
μ	=	Sliding friction coefficient
ρ	=	Density of steel
ρ_{Wat}	=	Density of water
g	=	Gravitational force
p_H	=	Hydrostatic pressure
p_{HD}	=	Hydrodynamic pressure
v	=	Flow velocity of water
σ	=	Normal stress
σ_d	=	Rated value of s
$\sigma_{R,d}$	=	Limit normal stress

γ_F = Partial safety factor for impacts
γ_M = Partial safety factor for resistance variables
$f_{Y,K}$ = Yield strength

d = Nominal diameter
d_2 = Pitch diameter
d_3 = Core diameter
P = Thread pitch
l = Length
α = Rising pitch angle
ρ = Friction angle
A_{TR80} = Core cross-section of trapezoidal thread TR80
$A_{Trequired}$ = Required core cross-section required for tensile strength

i = Reduction ratio
i_{GK} = Reduction ratio of bevel gearbox
i_{GS} = Reduction ratio of worm gearbox
i_{GSRG} = Reduction ratio worm gearbox/primary reduction gearing
i_{Ha} = Handwheel reduction ratio

f = Conversion factor from output torque to input torque
L = Lifetime factor
n = Speed
n_{Ha} = Typical handwheel speed
s = Travel of one stroke

DATA REQUEST SHEET OF AN ACTUATOR AND GEARBOX MANUFACTURER

Connecting elements	☐ Spindles ☐ Lantern gears ☐ Ropes ☐ Chains
Set-up (with sketch)	
Parameters Torques T_{max} T_{AO} incl. safety T_{AC} incl. safety $T_{A\text{-}ØEHW}$ (for operation mode) $T_{A\text{-}ØMHW/MW}$ (for lifetime) T/stroke (spindle) Subsequent strokes T/35 years Op. time/stroke	 ______Nm ______Nm ______Nm ______Nm ______Nm ______ ______ ______ ______min

Gearboxes	☐ Bevel gearboxes ______Size/red. ratio Valve attachment: ☐ B1 ☐ B3 ☐ B3-D ☐ D-D ☐ Spur gearboxes ______Size/red. ratio Valve attachment: ☐ B1 ☐ B3 ☐ B3-D ☐ D-D ☐ Worm gearboxes ______Size/red. ratio ☐ RR ☐ RL ☐ LR ☐ LL ☐ multi-turn ☐ Bronze worm wheel
Spindle nuts Electric actuator	☐ A ☐ AK ☐ Spindle lubrication ☐ without ☐ with actuator controls ______Type/size Speed: ________ Type of duty: ________ Voltage: ________ Frequency: ________ Valve attachment: ☐ B3-D ☐ D-D ☐ _________

Actuator controls	Actuator controls type______________________ ☐ On actuator ☐ On wall bracket Cable length: ______m ☐ Conventional ☐ Fieldbus Type:_____________
Required enclosure protection IP XX Required ambient temperature Special features	______ ______°C ________________________

LITERATURE

[1] Muhs, D.; Wittel, H.; Jannasch, D.; Voßiek, J.: Roloff/Matek: Maschinenelemente. 18. Auflage, 2007, Verlag Vieweg

[2] Kuhn, R.: Binnenverkehrswasserbau. 1985, Ernst & Sohn

[3] Wasserwirtschaft 83 (1993) 5, Vieweg+Teubner Verlag

[4] Schmaußer, G.; Nölke, H.; Herz, E.: Stahlwasserbauten. Kommentar zur DIN 19704, 2000, Ernst & Sohn

[5] DVWK Heft 249, 1998, Wirtschafts- und Verlagsgesellschaft Gas und Wasser mbH Bonn

[6] Stahlwasserbau, Stahlbau-Kalender 2006, Ernst & Sohn

[7] Uhlemann, H.-J.: Schleusen und Wehre. 2002, DSV-Verlag

[8] BAW-Brief Nr. 1, Nichtperiodische Beanspruchung. November 2001, Bundesanstalt für Wasserbau

[9] Kobus, H.: Hydrodynamische Belastung „Downpull". 2000, Institut für Wasserbau, Universität Stuttgart

[10] Decker, K.-H.; Kabus, K.: Maschinenelemente. 2004, Carl Hanser Verlag München Wien

[11] Beck, Th., Beiträge zur Geschichte des Maschinenbaus, 1899, Berlin Springer

[12] Niemann, G.; Winter, H.: Maschinenelemente. Band 2, 2. Aufl., 2003 Springer Verlag

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