Design of an actuator for an anti surge control valve

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Preface

This report “Design of an actuator for an anti surge control valve” covers the short Internship of the author. The internship is performed within the Dynamics and Control group of the department Mechanical engineering at the Technische Universiteit Eindhoven.

By taking into account which motion must be constrained, the design of the components can be determined. Creative thinking and producing new design concepts are the main subjects during this training period.

In order to get a proper design within a short time span I would like to thank Nick Rosiëlle and Jan van Helvoirt for giving me this interesting assignment and the support. For all tribological aspects I would like to thank Cees Meesters. Further I would like to thank Erwin Dekkers for producing the new actuator-concept at the GTD and Maarten Steinbuch for his enthusiasm and confidence to realize it.

Eindhoven, February 2007
Summary

The subject of this traineeship is the development of an actuator for a valve, which is applied for compressor surge control. The control design requires that the valve make a full-stroke movement of at least 20 Hz.

Before this project started, there already was a prototype available. This prototype consists of a commercial valve body and a dedicated actuator. In order to improve the actuator a new project has been started. The goal of this project is to improve the design of the actuator with respect to speed, durability and a better accuracy.

During the first part of the project, emphasis was on getting familiar with the related forces and power demands. A complete dynamic model of the existing design has been made to get a better understanding of all stiffness terms and the translating masses and the associated natural frequency. Because the transmission consists of an eccentric drive we are dealing with a certain variable stiffness. This phenomenon is simulated and the natural frequency is derived.

By doing some tests on a test-rig, several shortcomings of the existing prototype became clear. All of these aspects have been analyzed and they formed a basis for the new design. In order to get a proper result, several design concepts have been made. The mono-housing principle of the final design circumvents all the problems which occurred during the described tests. To test the new system it is applied at the test-rig in Duisburg/Siemens. At this time, the test results are not yet available.
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1 Introduction

Stable operation of axial and centrifugal compressors is limited towards low mass flows by
the occurrence of two aerodynamic compressor flow instabilities: rotating stall and surge.
These instabilities can cause severe damage to the machine due to large mechanical and
thermal loads in the blades and restrict its performance and efficiency. One way to cope with
these instabilities is active control. The dynamics of the compression system are modified by
feeding back perturbations into the flow field. This results in an extension of the stable
operating region, and the performance and efficiency of the compression system can be
improved.

The feedback is realized by a control system in combination with a sliding gate valve.
The surge frequency is about 0,2 Hz. However, control design gave a specification of 20 Hz
in order to damp all disturbances that cause instability. The sliding gate valve has to operate
with at least this frequency. In order to realize this, a fast actuator must be invented to serve
the sliding gate control valve. The focus of this study is to improve the prototype-actuator.

1.1 Research goals

As already mentioned, there is a need to actuate a control valve with a certain flow and a
frequency of 20 Hz.

After some research it becomes clear that there are not many operational valves available on
the market with such a flow and operating frequency. The most solutions are dealing with a
“trade-off” between flow and actuating frequency. (Either a low flow-rate valve which can
operate within a wide frequency-band or a high flow-rate valve can operate in a limited
frequency band).

The first solution, the prototype, consists of a commercial valve body and a dedicated
actuator. From research, several shortcomings became clear, which formed a starting point of
this project. The main goal of this project is to improve the design of the actuator with respect
to speed, durability, repeatability and accuracy.

1.2 Outline of this report

Chapter 2 gives an introduction to the phenomenon surge to provide some insight in this
topic. This chapter also addresses surge-control and describes the need of a control valve.
Chapter 3 describes the research and the tests which are done on the prototype actuator. Also
the conclusions after these tests are described in this chapter. In chapter 4 the new actuator
design is described and all components are discussed. The last chapter contains the conclusion
and some future recommendations are formed.
2 Surge and surge control

A lot of research has been done in order to prevent surge in centrifugal and axial compressors. In this chapter the phenomenon surge will be explained and also several techniques to prevent surge will be treated. One of these techniques is to apply surge control. By doing this, some requirements on the control valve become clear to apply surge control on a full scale compressor.

2.1 Surge

Ever since compressors have been invented people have had to deal with the phenomenon surge. In this case the compressor, which is an example and is depicted in Figure 2.1, is a centrifugal one. Compressor surge is a large amplitude oscillation of the total annulus averaged mass flow through the compressor. Notice that the surge frequency is around 0.2 Hz. This unstable axisymmetric flow pattern and the associated pressure fluctuations limit the stable operating range of the compressor towards low mass flows (see Figure 2.2). Surge not only reduces performance and efficiency, it can also damage the compressor due to large mechanical and thermal loads.

Figure 2.1: An industrial centrifugal compressor

Several actions are known to increase the stable operating range of the compressor system. Without going too much into detail, these actions can be grouped in three categories:

- Techniques that attempt to suppress these instabilities by control. (So called surge control) Main advantage of this approach is that this technique can be used for a wide range of machines and it can result in a considerable performance improvement compared to the other techniques. Moreover, it can easily be added to existing machines and designs.
- Techniques that are focused on better compressor interior design.
- Techniques that use variable geometry.

For the first option, many promising results of this approach have been reported, but a real breakthrough in the practical application of active surge control was not achieved. A critical
technological barrier is the limited actuation of the valve and other components. Implementing a real surge controller on a full scale centrifugal compressor is the main goal of the project of Jan van Helvoirt.

### 2.2 Surge control

When applying surge control, controllers are used which stabilize the operating points by feeding perturbations back into the flow field. Based on information from sensors, which detect medium fluctuations, the controller computes the desired perturbations to stabilize the system. These perturbations are introduced by a control valve. By feedback control, the compression system dynamics are modified such that the stable operating region is enlarged beyond the surge line. Note that if small fluctuations are fed back into the medium, small control actions are required. Because other disturbances than surge must be suppressed too, the control valve must operate with at least 20 Hz. To act on this frequency a very fast actuator must be designed and implemented in the test line up, which is depicted in Figure 2.3.

![Schematic representation of compressor rig](image)

**Figure 2.3:** Schematic representation of compressor rig

In this test line up we are dealing with a sliding gate control valve. (See appendix A) The reason to choose for this sliding valve is the fact that there is a linear relation between flow and lifting displacement of the plate. This valve has a stationary lamellar-plate and a moving lamellar-plate. The moving plate, which is connected to a rod, has to operate with a frequency of at least 20 Hz. Originally the valve was actuated by a pneumatic cylinder, but unfortunately it could not meet this demand. An actuator prototype was invented in order to meet this requirement. In Chapter 3 this design will be discussed.
3 The prototype of the actuator

In order to meet the minimal control valve actuation frequency, a new actuator concept is designed and mounted on the existing valve. In this paragraph the complete construction of the prototype and all experimental analysis will be treated. First the construction will be analyzed and after that the complete dynamic behavior of the actuator is simulated. At the end of this chapter some tests with the prototype are discussed.

3.1 Construction and working principle

As mentioned in chapter 2.2 the current design (as depicted in Figure 3.1) is based on a sliding gate control valve (4), see also Appendix A. In order to operate at at least 20 Hz, an alternative actuator (eccentric disc (2)) is added and connected to the rod (3). Because we are dealing with a control unit, due to the fact that anti-surge control is applied, a servomotor (1) is used. The datasheet is shown in Appendix B.

The actuator consists of a motor and transmission (see figure 3.2). In this case de transmission is an eccentric disc, which converts a rotating motion in a translating motion. The stroke of the valve is 8 mm. Initially the first aim was to realize a valve actuator with an eccentricity of 4 mm and a rotation of 180°, after producing the actuator, the eccentricity of the transmission was 4.5 mm and does not rotate entirely. The engine is served with a start-stop principle, so first it turns 125° and than turns back to 0°. By this operation the valve has gone open and close. By applying this principle a fast actuator is realized.
3.2 Analysis of the prototype

To analyze the system, and possibly later compare it with other systems, it is advisable to start with a dynamic model. Before the prototype is modeled, a better insight in the dynamics and the coherent forces and power demands is needed. Therefore a script (Appendix D) is written and some simulations are done. In Figure 3.3 some simulation results are depicted.

![Simulation of dynamics](image1)

Figure 3.3: Simulation of dynamics

From this figure it becomes clear that, to translate the sliding gate valve at 20 Hz, a maximum force of 350 N is needed. To translate with a certain speed we end up with a maximum needed power of 165 W (bottom part of Figure 3.3).

Because all the related quantities are now known, the stiffness of all components can be determined. For example the radial stiffness of bearings can easily be determined by using a bearing-stiffness-graph, which is depicted in Appendix C.

The dynamic behavior of a mechanism, like the natural frequency, can be predicted by using a dynamic model (Figure 3.4). The dynamic model can be divided into three parts. The first part (the electrical part) is the servo motor with a certain inertia and angular speed. The second part is the rotating part of the mechanical system and the third part is the total translating part of the actuator.
Figure 3.4: Dynamic model of the actuator and connected valve
To get a clear overview of how to get one specific stiffness for the complete system, the dynamical model is reduced (figure 3.5a-b). In order to add all 'springs' the next formulas are used:

Adding the angular stiffness

\[
\frac{1}{k_{\text{tot}}} = \frac{1}{k_{\text{ds}}} + \frac{1}{k_{\text{b}}} + \frac{1}{k_{\text{ms}}}
\]  

(1)

Where:

- \( k_{\text{ds}} \) = Radial stiffness of driveshaft (Nm/rad)
- \( k_{\text{b}} \) = Radial stiffness of bellow (Nm/rad)
- \( k_{\text{ms}} \) = Radial stiffness of motor shaft (Nm/rad)

Adding the translating stiffness

\[
\frac{1}{C_{\text{c}}} = \frac{1}{C_{\text{i-e}}} + \frac{1}{C_{\text{bt}}} + \frac{1}{C_{\text{at}}} + \frac{1}{C_{\text{l-cr}}} + \frac{1}{C_{\text{cr}}} + \frac{1}{C_{\text{angle joint}}} + \frac{1}{C_{\text{r}}}
\]  

(2)

Where:

- \( C_{\text{r}} \) = Rod stiffness (N/m)
- \( C_{\text{angle joint}} \) = Radial angle joint bearing stiffness (N/m)
- \( C_{\text{cr}} \) = Connecting plate stiffness (crank) (N/m)
- \( C_{\text{l-cr}} \) = Radial bearing stiffness (crank) (N/m)
- \( C_{\text{i-e}} \) = Radial bearing stiffness (eccenter) (N/m)
- \( C_{\text{bt}} \) = Eccenter shaft bending stiffness (N/m)
- \( C_{\text{at}} \) = Eccenter shaft shearing stiffness (N/m)

In this case \( k_{\text{tot}} \) is the total angular stiffness and \( C_{\text{c}} \) is the total translating stiffness. These added stiffnesses are coupled variably to each other by the eccentric transmission. Note that the transmission will act quadratic in order to add this summated stiffness. (See formula (3))

Because the moment of inertia, \( J_{\text{tot}} \), is coupled variably to the remaining mechanism, by means of the transmission, the equivalent mass in the reduced model will not be constant. In other words the mass is dependent of the angular position of the eccentric disc. In practice it is acceptable to neglect these small mass fluctuations. [6] The stiffness of the reduced model, as depicted in figure 3.5b, is variable and dependent of the angular position \( \beta \).

\[
\frac{1}{C(\beta)} = \frac{1}{C_{\text{c}}} + \left( \frac{dh}{d\beta} \right)^2 \cdot \frac{1}{k_{\text{tot}}}
\]  

(3)
To derive the stiffness for each position of the eccentric disc, a script (appendix E) has been written to visualize the stiffness. In figure 3.6 the result of this derivation is depicted.

In order to calculate the natural frequency at a fixed angular position we use Figure 3.6 and take the lowest stiffness value. (The lowest stiffness is present at $\pi/2$ rad and is also the value where certain resonances first will occur.) The natural frequency at this fixed point will be:

$$\omega_e = \sqrt{\frac{C_e}{m_{eq}}} = \sqrt{\frac{2.149 \cdot 10^5}{0.435}} = 703 \text{ rad/sec}$$  \hspace{1cm} (4)

$$f_{eig} = \frac{\omega_e}{2\pi} = \frac{703}{2\pi} = 112 \text{ Hz}$$  \hspace{1cm} (5)

This natural frequency is sufficiently high compared to the compressor surge dynamics. It can be concluded that this will not form a limitation for compressor surge control.
3.3 Testing the prototype

The valve and actuator which are depicted in Figure 3.1 were tested at Siemens AG, Duisburg. See also the lay-out of the test rig in Fig 2.3 & 3.7. The aim of two days testing was to test if the actuator was able to operate at a certain frequency and of course to apply an anti-surge control on the compressor to investigate its working principle. During these tests several shortcomings of the actuator became clear. The most important shortcomings are summed up below:

1. Bended (and after a while a broken) connecting plate (Illustrated in Figure 3.8);
2. After some test period no position of the eccentric disc could be determined;
3. The motor can not follow the input signal;
4. There is a certain “overshoot” which give some vibrations.

For every one of these phenomena there is a certain cause. The failure of the connecting plate is the result of multiple actions. The area moment of inertia of the plate, which connects the eccentric to the valve, is low due to a hole made in the plate, this in contrast with the analysis in the previous chapter. (Is discussed in chapter 4.1.2 in detail) In practice the forces, acting on the connection plate are much higher than expected, due to friction. Also keeping in mind that the tests were done at a higher frequency, the related forces are much higher. To mount the actuator to the valve a pipe is screwed on it. Because there is no stop present, a certain pre-stress is introduced in the connecting plate. There is also a position error of the housing with respect to the connection of the valve. To cancel this error a ball bearing is mounted in the connecting plate. Due to friction, present in this type of bearing, there is a certain torque with its related stresses acting on the connecting plate. With repeated motion, fatigue will occur. All these aspects leads to a broken crank which is depicted in figure 3.8.

The second point has all to do with de connection between the motor shaft and the eccentric disc. In the prototype design the connection is a bellow which is based on force-closed connection. Due to the high torques and accelerations there was a certain slip in the bellow present and a position error was created between the motor and eccentric disc.
Friction is the main cause for the third point. For sealing reasons there is a lot of friction and generated heat, which give non-linearity’s. It is very hard to control these kinds of aspects and give as result a certain position error.

The last point has to do with the fact that the eccentricity is 4.5 mm. To complete the stroke of 8 mm the eccentric disc has to rotate over 125°. To realize this some “start-stop”-effects will occur. Because of this motion the systems constantly suffers from high accelerations. There is energy storage in the shaft which causes an overshoot when the motor shaft stops. Due to control aspects some vibrations will arise to compensate the position error.

By analyzing the prototype and summarizing all conclusions made in the previous chapters, a new design for a valve actuator can be made. At first the natural frequency was believed to be the most important problem. By analyzing the dynamical part of the actuator, by means of a dynamic model, there was more insight in this aspect. All components were calculated and checked on the stiffness. Ending up and translating all stiffness to a simple mass-spring model, the derived natural frequency is 112 Hz. It is clear that this aspect is not a restricting parameter for the control valve.

As described in this chapter, more aspects become clear. For all enumerated points there is a solution, which will be discussed now:

1. Bended/broken connecting plate. Solutions:
   - Raising the stiffness;
   - Avoid align-errors between the shafts, so that there is no torsion present;
   - When there are no alignment-errors, a stiffer bearing can be applied. In the new design a little error will be canceled by the elasticity of the plates;
   - Prevent pre-stress, due to mounting effects.

2. No position of the eccentric shaft could be determined after a while. Solution:
   - The linking system between the motor shaft en eccentric drive shaft must be carried out with a key. (form closed-connection)

3. Motor can not follow the input signal. Solution:
   - Due to the friction (for sealing reasons) there are a lot of non-linearity’s. The new model makes use of a simple metallic sealing principle which is a proper solution for this application.

4. There is a certain “overshoot”. Due to the start-stop motion the construction has to deal with accelerations and decelerations. This give some high force and torque peaks and will load the construction more heavy. Solution:
   - In the new model the eccentricity will be 4 mm so that the shaft has to rotate entirely (360°) en start stop effects will be avoided.

All these points and related solutions will be used as a basis for the new actuator design.
4 New actuator design

In the previous chapter it was concluded that several changes must be made, so a new actuator is designed. In the first paragraph the complete construction will be analyzed and all fundamental design changes will be treated. Finally, the prototype design and the new actuator design will be compared in order to check the improvements.

4.1 Construction

In the new construction, the actuator and valve are mounted in one housing. In order to decrease the translating mass the eccentric disc and the connecting plates are directly connected to the moving disc. As discussed in the previous chapter the connecting plate was the weakest part of the construction. In this design the stiffness is increased by applying a sandwich construction. The two housing parts are positioned very precisely with respect to each other, by applying fitting pins, which will contribute to avoid possible align-errors. By first assembling the moving disc and the connecting plates (and the eccentric disc) and subsequently mount this in the right housing, no pre-stress is introduced in the dynamical parts, see also Figure 4.1.

Figure 4.1: New valve design

By decreasing the translating mass and raising the stiffness, the natural frequency of the system is increased. In order to avoid the start-stop principle, which is discussed in paragraph 3.3, the eccentricity was chosen equal to half of the stroke, 4 mm. To open and close the
valve, the eccentric disc has to rotate entirely and no start-stop effects are present. This is not only energy-efficient but also contributes to the controllability of the anti-surge control valve.

4.1.1 Housing

The new housing principle consists of two halves. Because the construction no longer consists of a single actuator, but a combination of valve and actuator, the number of demands increases. Not only taking care of the reaction forces but also positioning for example the eccentric drive shaft with respect to the valve (which is discussed in the next paragraph) was part of the design requirements. In order to achieve this, the tolerances of the housing have to be very high. To mount the two halves precisely to each other, fitting pins are applied. Sealing aspects also belong to the requirements due to the fact that the sliding disc (valve) is integrated in the housing. All detailed sealing aspects of the housing will be treated in paragraph 4.1.4.

Figure 4.2: New housing principle

In order to meet these requirements, a new housing has been designed. As shown in Figure 4.2. By combining a groove (which is not depicted in Figure 4.2) and an O-ring wire, the two halves are sealed with respect to each other. This principle is depicted in Figure 4.3, also see appendix F.

Figure 4.3: Sealing principle of two houses
4.1.2 Connecting plates

The connecting plate was the weakest part of the construction, as shown in paragraph 3.3. The ball joint bearing (mounted in the connecting-“rod”) also had a low stiffness-value and a certain clearance space. In order to increase the stiffness, two parallel plates (sandwich construction) are mounted on a flange that results in a greater cross section surface, see Figure 4.4.

![Figure 4.4: Exploded view of sandwich construction](image)

![Figure 4.5: Xyz-axis with related revolutions](image)

In Appendix G, Figure A.9 illustrates the previous design and Figure A.10 depicts the new design of the connecting plates. By comparing the two constructions it becomes clear that the design has been improved on several points. First of all the buckling length is decreased in order to increase the critical buckling force and to have a compact construction.

Secondly, the new design makes use of two housing parts, which are positioned by fitting pins. Due to this measure, a possible position-error (θ) of the actuator with respect to the valve, see Figure 4.5 for orientation, is avoided. Because this position error is not present in the new design, the use of a needle bearing is possible. Possible alignment-errors, due to the machinery tolerance of the housing, can be compensated by the elasticity of the sandwich construction. (Note that in the previous connecting plate this problem was solved by applying an ball joint bearing in point D of Figure A.9) By applying a needle bearing the bearing-stiffness is increased and also the tilt stiffness.

By applying a needle bearing and making use of a sandwich construction, the rotation φ in point B, see figure A.10, is constrained. With all these measures the effective buckling-length (see figure 4.6-1)* is reduced with a k-factor of $1/\sqrt{2}$ and this contributes positively to the critical buckling-force. Furthermore by applying a sandwich construction, the second moment of inertia about the neutral axis $x$ has been increased, see section C-C in Figure A.10.

* note that in figure 4.6, the first construction with “1” illustrates the new concept and “2” illustrates the previous concept
In order to derive the critical buckling force we first have to derive the second moment of inertia:

\[ I_{xx} = \frac{1}{12} b \cdot h^3 \]  
(6)

Where:  
\( I_{xx} \) = second moment of inertia in x direction \((m^4)\)  
\( b \) = width of the construction \((m)\)  
\( h \) = height of the construction \((m)\)

In figure 4.6, situation (1) is typified as the previous layout and situation (2) is supposed to be the sandwich construction.

For situation (1) \( I_{xx(1)} = \frac{1}{12} b \cdot h^3 = \frac{1}{12} \cdot 0.01 \cdot 0.001^3 = 8.33 \cdot 10^{-13} m^4 \)

For situation (2) \( I_{xx(2)} = \frac{1}{12} b \cdot h^3 = \frac{1}{12} \cdot 0.035 \cdot 0.012^3 - \frac{1}{12} \cdot 0.035 \cdot 0.010^3 = 2.12 \cdot 10^{-9} m^4 \)

Figure 4.6: Two ways of clamping

The critical buckling-force will be:

\[ F_{cr} = \frac{\pi^2 \cdot E \cdot I}{(k \cdot l)^2} \]  
(7)

Where:  
\( F_{cr} \) = critical buckling force \((N)\)  
\( E \) = modulus of elasticity \((N/m^2)\)  
\( I \) = second moment of inertia \((m^4)\)  
\( k \) = constant factor \((-)\)  
\( l \) = unsupported length of column \((m)\)
\[ F_{cr(1)} = \frac{\pi^2 \cdot E \cdot I_{xx(1)}}{(l)^2} = \frac{\pi^2 \cdot 2.1 \cdot 10^{11} \cdot 8.33 \cdot 10^{-13}}{(0.0745)^2} = 311N \]

\[ F_{cr(2)} = \frac{\pi^2 \cdot E \cdot I_{xx(2)}}{(\frac{l}{\sqrt{2}})^2} = \frac{\pi^2 \cdot 2.1 \cdot 10^{11} \cdot 2.12 \cdot 10^{-09}}{(0.044)^2} = 4.5 \cdot 10^6 N \]

In the above made calculations, again the first (1) case is the previous critical buckling force and the second (2) case is the critical buckling force of the new design. (Sandwich construction)

By taking into account that the maximum reached force is 350 N in the first case, buckling will occur. In the second case the critical buckling force is much higher. Also by the fact that the maximum reached forces will be less than in the first case (no start-stop effects and a reduced translating mass), no buckling will occur in the new design.

**4.1.3 Moving disc**

The moving disc is translating with at least 20 Hz in the new construction. In the previous construction the moving disc was guided at the top by a rod and two contact-planes were guiding the left en right side which results in a lot of friction.

In order to prevent a rotation of the moving disc, four needles have been applied. By doing so also the friction is reduced in comparison to the previous lay-out, see figure 4.7. As represented in the bottom figure there is a line-contact between the surface of the moving disc and the guiding needle. It is known that a line-contact has a greater contact surface which result in a lower Hertz strain.

**Figure 4.7: Moving disc with guiding needles**
As described in the previous chapter, the connection plate length has been reduced to 44 mm. As a consequence, the forces in the x-direction ($F_x$) will increase. In order to determine these forces a simple model has been made in a program which can simulate moving masses which are connected to each other by mechanical connections, see Figure 4.8.

![Simulation model and simulation results](image)

**Figure 4.8: Simulation model (left) and the related simulation results (right)**

In this model, element (1) is the eccentric disc and element (2) is the connecting plate. There is a linear dependence between the elements. The mass of the moving disc is concentrated at the end of element (2) as a point-mass. Point (3) is supposed to be the guiding needle. In order to simulate this model, a motion of 30 Hz is applied at point (1). In the right-hand side of Figure 4.8 the simulation results are depicted. From this figure it can be concluded that $F_{x(max)} = 3.5 \times 10^3$ N. Because now the related force in the x-direction is known, the diameter of the guiding needles can be determined by a simple design-calculation.

$$d_{\text{min}} = k \cdot \sqrt{\frac{c_b \cdot F_{x(max)}}{\sigma_b}}$$  \hspace{1cm} (8)

Where:

- $d_{\text{min}}$ = minimal required diameter (mm)
- $k$ = clamp factor (-)
- $\sigma_b$ = strain-rate of the applied material (N/mm²)
\[ c_b = \text{safety value (\text{-})} \]
\[ F_{x(\text{max})} = \text{maximum acting force in x direction (N)} \]

The applied material is C45, whereby \( \sigma_b = 580 \text{ N/mm}^2 \). The safety value is determined according to table 3.6a of Roloff & Matek [7].

\[
d_{\text{min}} = 1.2 \cdot \sqrt{\frac{1.3 \cdot 3.5 \cdot 10^3}{580}} = 3.4 \text{ mm}
\]

In the final design, a diameter of 5 mm was selected because this was the first standardized size which corresponds with this calculation.
4.1.4 Bearings and sealing

Due to the fact that the valve is integrated into the actuator, there is a possible leakage flow from the valve to the actuator. In order to prevent this, a form of sealing is required. The housing of the actuator is not separated of the valve, which give some leakage flow from the valve towards the actuator. Most of these leakage-problems occur when the valve is closed and there is a certain pressure difference present. In this case, see Figure 4.9, the moving disc is pushed against the stationary disk and a certain space will form between the moving disc and the housing, see black arrows in Figure 4.9. The flow is moving from (1) to (2). Because the two halves are sealed by the principle which is depicted in Figure 4.3, the only way of leakage is along the eccentric-shaft in point (2). Therefore a proper sealing-ring must be applied. In order to have a way of sealing between the moving plates, the material choice of these two plates is a combination between hard- metal and through hardened steel. When the valve is closed for a certain period with a certain pressure difference present in the valve, this pressure difference will also occur in the actuator house. Because of this pressure, a special sealing must be applied (in point (2)) which can resist this pressure difference. Because the pressure difference in the test-rig at Duisburg doesn’t exceed 4 bar and taking into consideration that the maximum allowed pressure difference of the sealing is 10 bar, no problems will occur, see Appendix H.

![Figure 4.9: An overview of a possible leakage flow](image-url)
4.2 Comparison of designs

In order to compare the previous and the new prototype system, a comparison-table, Table 4.1, is made. As already mentioned in this report not only the natural frequency, but also other aspects, are important. The natural frequency of the previous system was, as derived in chapter 2.2, high enough to avoid resonances and therefore didn’t need anymore attention. In the new model this frequency has become higher still because of the construction differences. The reliability is also improved. By ruling out some assembly errors all components are better aligned and therefore no unnecessary stresses are present in the new system. By applying a new guiding principle, the amount of friction is reduced, see chapter 4.1.3, and therefore a lot of non-linearity’s are gone.

<table>
<thead>
<tr>
<th>Criteria ↓</th>
<th>System →</th>
<th>Previous system</th>
<th>New system</th>
<th>Quantity:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural frequency $\omega_e$ (derived)</td>
<td>112 (+)</td>
<td>263 (++)</td>
<td>[Hz]</td>
<td></td>
</tr>
<tr>
<td>Assembly</td>
<td>Easy</td>
<td>Easy</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Leakage-flow*</td>
<td>++</td>
<td>+/-</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Position error due to assembly* ($\theta$)</td>
<td>-</td>
<td>++</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Angular position error of driveshift ($\psi$)</td>
<td>After testing +/- 20°</td>
<td>No error possible due to mounted key (form-closed)</td>
<td>[°]</td>
<td></td>
</tr>
<tr>
<td>Implementing actuator in test rig</td>
<td>+</td>
<td>+</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Friction/heat loss*</td>
<td>--</td>
<td>+</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Costs (prototype)</td>
<td>2500</td>
<td>3400</td>
<td>[€]</td>
<td></td>
</tr>
</tbody>
</table>

Table 4.1: Comparing both systems

* No quantities are available due to the fact that no tests were done so far with the new system, a rough estimation (by an illustration of + or -) can be made.
5 Conclusions and recommendations

As mentioned in the introduction, there were not much operational valves available on the market with such a flow and operating frequency. From controlling aspects it became clear that the surge control valve has to operate with at least 20 Hz. In order find a solution, a commercial valve body and a dedicated actuator were applied. During some tests, some shortcomings became clear and formed a basis for the new design.

By maintaining the actuating frequency the system must become more operational safe. Also the durability, repeatability and the accuracy must become better.

It can be concluded that with the new design most of the targets are fulfilled. The durability has become better by a manner of positioning of components in one single house. By introducing a form closed connecting between motor and valve, similar measurements can be performed. Due to the fact that the eccentric rotates entirely, the valve can move precisely and also the controllability has been improved.

Figure 5.1: The end result

One remark must be made to the new solution. When the pressure difference across the two plates (moving and stationary disc) becomes more than 10 bar (for example when the valve is completely closed) the leakage flow is not acceptable. This has to do with the fact that the sealing at the rotating driveshaft (nr. 2 in fig. 4.9) is not resisted against this pressure. Eriks is able to develop such a sealing.

It can be concluded that with simple adaptations a proper solution has been found in order to contribute to the anti-surge phenomenon.
Appendix A  Valve datasheet

Figure A.1: Datasheet of the commercial valve, type: 8036
Source: Schubert & Salzer catalogue
Appendix B  Motor datasheet

Figure A.2: Servo motor datasheet; type AKM-24C
Source: Selection guide AKM

<table>
<thead>
<tr>
<th>Mounting Code</th>
<th>( T_{	ext{in}} ) (Nm)</th>
<th>( \tau_{	ext{in}} ) (Nm)</th>
<th>( T_{	ext{max}} ) (Nm)</th>
<th>( T_{	ext{max}} ) (Nm)</th>
<th>( T_{1} ) (Nm)</th>
<th>( T_{2} ) (Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AC</td>
<td>4.00 (1.90)</td>
<td>–</td>
<td>0.10 (0.10)</td>
<td>0.50 (0.50)</td>
<td>0.40 (0.40)</td>
<td>0.40 (0.40)</td>
</tr>
<tr>
<td>AE</td>
<td>4.00 (1.90)</td>
<td>–</td>
<td>0.10 (0.10)</td>
<td>0.50 (0.50)</td>
<td>0.40 (0.40)</td>
<td>0.40 (0.40)</td>
</tr>
</tbody>
</table>

Figure A.3: Servomotor datasheet; type AKM-24C
Source: Selection guide AKM
## AKM2x - Up to 640 VDC

See system data beginning on page 8 for typical torque-speed performance.

<table>
<thead>
<tr>
<th>PARAMETER</th>
<th>AKM21</th>
<th>AKM22</th>
<th>AKM23</th>
<th>AKM24</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max Rated DC In Voltage</td>
<td>320</td>
<td>320</td>
<td>320</td>
<td>320</td>
</tr>
<tr>
<td>Continuous Torque (ft-lb) for all voltage, no speed</td>
<td>4.4</td>
<td>4.4</td>
<td>4.4</td>
<td>4.4</td>
</tr>
<tr>
<td>Continuous Current (Amp) for all voltage, no speed</td>
<td>1.5</td>
<td>1.5</td>
<td>1.5</td>
<td>1.5</td>
</tr>
<tr>
<td>Continuous Torque (Nm) for all voltage, no speed</td>
<td>0.38</td>
<td>0.38</td>
<td>0.38</td>
<td>0.38</td>
</tr>
<tr>
<td>Max Acceleration of Speed</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Peak Torque (Nm)</td>
<td>13.6</td>
<td>13.6</td>
<td>13.6</td>
<td>13.6</td>
</tr>
<tr>
<td>Peak Current</td>
<td>6.3</td>
<td>6.3</td>
<td>6.3</td>
<td>6.3</td>
</tr>
<tr>
<td>Rated Torque (p.u.)</td>
<td>0.46</td>
<td>0.46</td>
<td>0.46</td>
<td>0.46</td>
</tr>
<tr>
<td>Rated Power (p.u.)</td>
<td>0.45</td>
<td>0.45</td>
<td>0.45</td>
<td>0.45</td>
</tr>
<tr>
<td>Rated Speed</td>
<td>2500</td>
<td>2500</td>
<td>2500</td>
<td>2500</td>
</tr>
<tr>
<td>Rated Power (p.u.)</td>
<td>0.59</td>
<td>0.59</td>
<td>0.59</td>
<td>0.59</td>
</tr>
<tr>
<td>Rated Torque (p.u.)</td>
<td>0.38</td>
<td>0.38</td>
<td>0.38</td>
<td>0.38</td>
</tr>
<tr>
<td>Rated Power (p.u.)</td>
<td>0.47</td>
<td>0.47</td>
<td>0.47</td>
<td>0.47</td>
</tr>
<tr>
<td>Rated Speed</td>
<td>3000</td>
<td>3000</td>
<td>3000</td>
<td>3000</td>
</tr>
<tr>
<td>Rated Power (p.u.)</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
<td>0.7</td>
</tr>
<tr>
<td>Rated Torque (p.u.)</td>
<td>0.53</td>
<td>0.53</td>
<td>0.53</td>
<td>0.53</td>
</tr>
<tr>
<td>Rated Power (p.u.)</td>
<td>0.65</td>
<td>0.65</td>
<td>0.65</td>
<td>0.65</td>
</tr>
<tr>
<td>Rated Speed</td>
<td>4000</td>
<td>4000</td>
<td>4000</td>
<td>4000</td>
</tr>
<tr>
<td>Rated Power (p.u.)</td>
<td>0.8</td>
<td>0.8</td>
<td>0.8</td>
<td>0.8</td>
</tr>
<tr>
<td>Rated Torque (p.u.)</td>
<td>0.72</td>
<td>0.72</td>
<td>0.72</td>
<td>0.72</td>
</tr>
<tr>
<td>Rated Power (p.u.)</td>
<td>0.85</td>
<td>0.85</td>
<td>0.85</td>
<td>0.85</td>
</tr>
<tr>
<td>Rated Speed</td>
<td>5000</td>
<td>5000</td>
<td>5000</td>
<td>5000</td>
</tr>
<tr>
<td>Rated Power (p.u.)</td>
<td>0.94</td>
<td>0.94</td>
<td>0.94</td>
<td>0.94</td>
</tr>
<tr>
<td>Rated Torque (p.u.)</td>
<td>0.81</td>
<td>0.81</td>
<td>0.81</td>
<td>0.81</td>
</tr>
<tr>
<td>Rated Power (p.u.)</td>
<td>0.97</td>
<td>0.97</td>
<td>0.97</td>
<td>0.97</td>
</tr>
<tr>
<td>Torque Constant (Amp)</td>
<td>0.36</td>
<td>0.36</td>
<td>0.36</td>
<td>0.36</td>
</tr>
<tr>
<td>Back EMF Constant</td>
<td>19.3</td>
<td>19.3</td>
<td>19.3</td>
<td>19.3</td>
</tr>
<tr>
<td>Resistance (Ohm)</td>
<td>13.8</td>
<td>13.8</td>
<td>13.8</td>
<td>13.8</td>
</tr>
<tr>
<td>Inductance (Henry)</td>
<td>16</td>
<td>16</td>
<td>16</td>
<td>16</td>
</tr>
<tr>
<td>inertia (kgm²)</td>
<td>2.31</td>
<td>2.31</td>
<td>2.31</td>
<td>2.31</td>
</tr>
<tr>
<td>Backlash backlash feedback (°)</td>
<td>1.55</td>
<td>1.55</td>
<td>1.55</td>
<td>1.55</td>
</tr>
<tr>
<td>Optical encoder (mm)</td>
<td>0.012</td>
<td>0.012</td>
<td>0.012</td>
<td>0.012</td>
</tr>
<tr>
<td>Weight (kg)</td>
<td>0.08</td>
<td>0.08</td>
<td>0.08</td>
<td>0.08</td>
</tr>
<tr>
<td>Judder friction (Nm)</td>
<td>0.002</td>
<td>0.002</td>
<td>0.002</td>
<td>0.002</td>
</tr>
</tbody>
</table>

**Figure A.4:** Servomotor datasheet; type AKM-24C
Source: Selection guide AKM
Appendix C  Bearing stiffness

Figure A.5: Needle-bearing stiffness (in Dutch)
Source: Philips
Figure A.6: Deep groove ball bearing stiffness (in Dutch)
Source: Philips
Appendix D  M-file for force and power demand

% VALVEDESIGN  Initial design calculations for valve actuator.
% This program performs the sizing calculations for a Kollmorgen
% brushless servomotor. The motor is needed for actuating the surge
% control valve that should achieve at least a 20 Hz bandwidth.

% CENTRIFUGAL COMPRESSOR SURGE
% Modeling and Identification for Control

clear
close all
clc

%%% PARAMETERS %%%
Fbw  = 25;        % Required bandwidth    (Hz)
Ts   = 1e-3;      % Sample time         (s)
rho  = 7.8e3;     % Density of steel   (kg/m3)

% Valve
DP   = 2;        % Pressure difference(bar)
Mv   = 0.353;     % Moving mass of valve(kg)
Mr   = 0.235;     % Moving mass of rod(kg)
Mc   = 0.054;     % Translating mass of transmission (kg)
Ml   = Mv + Mr + Mc;  % Total moving mass (load)(kg)
dv   = 0.008;     % Stroke of valve(m)

% Transmission
Sr   = dv / (0.697 * pi);  % Rotation -> translation   (m/rad)
deg  = Sr * 180 / pi;     % Eccentricity 4.5 mm (gives angle of 125.4679
eta  = 0.85;            % Transmission efficiency   (-)

rho  = 7.8e3;         % Density of steel   (kg/m3)
Dd   = 0.025;         % Diameter of crank     (m)
dd   = 0.007;         % Thickness of crank    (m)
Da   = 0.008;         % Diameter of axle      (m)
lA   = 0.070;         % Length of axle        (m)
Ib   = 2.5e-6;        % Inertia of bellow     (kg m2)

Md   = rho * pi * (Dd/2)^2 * dd;  % 'Exact' mass of disc  (kg)
Ma   = rho * pi * (Da/2)^2 * la;  % 'Exact' mass of axle  (kg)

% 'Exact' inertia of transmission
Ire  = 0.5 * Md * (Dd/2)^2 + 0.5 * Ma * (Da/2)^2 + Ma * (dv/2)^2;
Ire  = Ire + Ib;

% Estimated transmission inertia
Ir   = 2e-5;          % Transmission inertia  (kg m2)

% Motor inertia (from Danaher Motion catalogue)
Im   = 2.7e-5;        % Motor inertia AKM     (kg m2)
%%
%% CALCULATIONS LOAD %%

\[
F_k = 200 + 34.5 \times D_P; \quad \text{Dry friction force } (N)
\]

\[
F_b = 100; \quad \text{Viscous friction constant } (N)
\]

\[
A = \frac{dv}{2}; \quad \text{Position amplitude } (m)
\]

%% Trajectory

\[
t = (0:Ts:1); \quad \text{Time vector } (s)
\]

\[
x = A + A \times \sin(F_{bw} \times 2 \times \pi \times t); \quad \text{Position trajectory } (m)
\]

\[
x_d = [A \times F_{bw} \times 2 \times \pi, \text{diff}(x) / Ts]; \quad \text{Velocity trajectory } (m/s)
\]

\[
x_{dd} = [0, \text{diff}(x_d) / Ts]; \quad \text{Acceleration trajectory } (m/s^2)
\]

\[
F = M_l \times x_{dd} + F_k \times \text{sign}(x_d) + F_b \times x_d; \quad \text{Required force } (N)
\]

\[
P = F \times x_d; \quad \text{Required power } (W)
\]

%% PLOTTING %%

\[
\text{figure(1)}
\]

\[
\text{subplot(511)}
\]

\[
\text{plot}(t, x), \quad \text{xlabel('t (s)'), ylabel('x (m)')}
\]

\[
\text{grid on}
\]

\[
\text{title(['Eccentric disc, load side: f_{bw} = ', num2str(F_{bw}), ' Hz'])}
\]

\[
\text{subplot(512)}
\]

\[
\text{plot}(t, x_d), \quad \text{xlabel('t (s)'), ylabel('v (m/s)')}
\]

\[
\text{grid on}
\]

\[
\text{subplot(513)}
\]

\[
\text{plot}(t, x_{dd}), \quad \text{xlabel('t (s)'), ylabel('a (m/s^2)')}
\]

\[
\text{grid on}
\]

\[
\text{subplot(514)}
\]

\[
\text{plot}(t, F), \quad \text{xlabel('t (s)'), ylabel('F (N)')}
\]

\[
\text{grid on}
\]

\[
\text{subplot(515)}
\]

\[
\text{plot}(t, P), \quad \text{xlabel('t (s)'), ylabel('P (Watt)')}
\]

\[
\text{grid on}
\]

%% CALCULATIONS MOTOR %%

\[
\text{th} = x / Sr; \quad \text{Angular position } (rad)
\]

\[
\text{thd} = x_d / Sr; \quad \text{Angular speed } (rad/s)
\]

\[
\text{thdd} = x_{dd} / Sr; \quad \text{Angular acceleration } (rad/s^2)
\]

\[
\text{Tem} = F \times Sr \times (1 / \eta) + (I_r + I_m) \times thdd; \quad \text{Total motor torque } (Nm)
\]

\[
\text{Pem} = \text{Tem} \times thd; \quad \text{Total motor power } (W)
\]

\[
\text{Tem_max} = \text{max}(\text{Tem}); \quad \text{Peak motor torque } (Nm)
\]

\[
\text{Tem_rms} = \sqrt{\text{sum}(\text{Tem} \times 2) / \text{length}(\text{Tem})}; \quad \text{RMS motor torque } (Nm)
\]

\[
\text{thdd_max} = \text{max}(\text{thdd}); \quad \text{Peak acceleration } (rad/s^2)
\]

\[
\text{thdd_rms} = \sqrt{\text{sum}(\text{thdd} \times 2) / \text{length}(\text{thdd})}; \quad \text{RMS acceleration } (rad/s^2)
\]

\[
N_{max} = (\text{max}(\text{thd}) / (2 \times \pi)) \times 60 \times 1e-3; \quad \text{Maximum speed } (krpm)
\]
%%% PLOTTING %%%
figure(2)
subplot(511)
plot(t, th), xlabel('t (s)'), ylabel('\theta (rad)')
grid on
title(['Eccentric disc, motor side: f_b_w = ', num2str(Fbw), ' Hz'])
subplot(512)
plot(t, thd), xlabel('t (s)'), ylabel('\omega (rad/s)')
grid on
subplot(513)
plot(t, thdd), xlabel('t (s)'), ylabel('d\omega (rad/s^2)')
grid on
subplot(514)
plot(t, Tem), xlabel('t (s)'), ylabel('T (Nm)')
grid on
subplot(515)
plot(t, Pem), xlabel('t (s)'), ylabel('P (W)')
grid on

%%% DISPLAY %%%
disp('')
disp('VALVE DESIGN - MOTOR SIZING')
disp('----------------------------')
disp('')
disp('ACTUATOR SPECIFICATIONS')
disp('')
fprintf(1, 'Bandwidth : %3.0f Hz\n', Fbw)
fprintf(1, 'Stroke : %3.0f mm\n', dv * 1e3)
disp('')
disp('LOAD SIDE')
disp('')
fprintf(1, 'Mass of valve : %6.6f kg\n', Mv)
fprintf(1, 'Mass of crank : %6.6f kg\n', Md)
fprintf(1, 'Mass of shaft : %6.6f kg\n', Ma)
disp('')
fprintf(1, 'Bearing efficiency : %6.2f\n', eta)
disp('')
fprintf(1, 'Friction force : %6.2f N\n', Fk)
fprintf(1, 'Damping : %6.2f Ns/m\n', Fb)
fprintf(1, 'Total radial force : %6.2f N\n', max(F))
disp('')
disp('MOTOR SIDE')
disp('')
fprintf(1, 'Exact crank inertia : %3.3e kg m^2\n', Ire)
fprintf(1, 'Rounded crank inertia : %3.2e kg m^2\n', Ir)
fprintf(1, 'Motor inertia : %3.2e kg m^2\n', Im)
disp('')
fprintf(1, 'Required peak torque : %6.3f Nm\n', max(Tem))
fprintf(1, 'Required rms torque : %6.3f Nm\n', Tem_rms)
disp('')
fprintf(1, 'Required max speed : %6.0f rpm\n', Nmax * 1000)
fprintf(1, 'Required max acc. : %6.0f rad/s^2\n', thdd_max)
Appendix E  M-file for variable stiffness

%% RE-DESIGN OF AN ACTUATOR FOR ANTI SURGE CONTROL
%% Calculations of variable stiffness due to the eccenter transmission
%% placed in the actuator for an anti surge valve.

%% Short internship project
%% Dennis Leermakers
%% Technische Universiteit Eindhoven

disp('clear memory')
clc
clear
close all

disp('system parameters')
Cr= 7.3e7                  % N/m       Rod stiffness
Cbk= 2.57e5               % N/m       Radial bearing stiffness
Ccr= 3e6                    % N/m       Connecting rod stiffness
CLcr= 6.1e6              % N/m       Radial bearing stiffness
CL= 1.02e7                   % N/m       Radial bearing stiffness
Cbt= 2.09e8                 % N/m       Eccenter shaft bending stiffness
Cat= 8.02e6              % N/m       Eccenter shaft shearing stiffness
kds= 1.15e3              % Nm/rad    Radial stiffness of driveshaft
kb= 3e3               % Nm/rad    Radial stiffness of bellow
kms= 3.2e3              % Nm/rad    Radial stiffness of motorshaft
m= 0.435                 % kg        Total translating mass
e= 0.004                    % m         Eccentricity

Cc=1/(1/Cr + 1/Cbk + 1/Ccr + 1/CLcr + 1/CL + 1/Cbt + 1/Cat)
% N/m       Total translating stiffness (constant value)

k =1/(1/kds + 1/kb + 1/kms)
% Nm/rad    Total radial stiffness

disp('calculating the variable stiffness')
QA=[];                        % Empty matrix A

db=0:pi/100:0.697*pi;        % Angular vector

for w=db;
    var=e*sin(w);            % m/rad       Variable transmission
    Cb=1/(1/Cc + (1/(k*(1/var)^2)))); % N/m       Calculating variable stiffness
    QA=[QA, Cb];
end;

figure(1)
plot(db,QA,'-');
xlabel('Angle position of eccenter (rad)');
ylabel('Variable stiffness (N/m)');
title('VARIABLE STIFFNESS N/m');
grid
Appendix F  O-ring wire (house sealing)

Sponsrubberplaten
Sponsrubberplaten worden in diverse stramheden ("stevigheid", "hardheid") en kwaliteiten geleverd. De platen zijn aan weerszijden voorzien van een niet gladde huid, een z.g. doekafdruk. Kleur: zwart en of grijs. Plaatafmeting 1000 x 1000 mm. Zelfklevende uitvoering is eveneens leverbaar op bestelling.

<table>
<thead>
<tr>
<th>Technische gegevens</th>
<th>HOP/NR</th>
<th>HOH/NR</th>
<th>HOH/CR</th>
</tr>
</thead>
<tbody>
<tr>
<td>Soortelijk gewicht in kg/m²</td>
<td>300 - 500</td>
<td>300 - 600</td>
<td>450 - 800</td>
</tr>
<tr>
<td>Temperatuurbestendigheid in °C</td>
<td>-40 tot +60</td>
<td>-40 tot +60</td>
<td>-40 tot +110</td>
</tr>
<tr>
<td>Samendrukbaarheid in kPa vlg. ASTM D 1056</td>
<td>60 - 90</td>
<td>100 - 200</td>
<td>100 - 170</td>
</tr>
<tr>
<td>Bijvende drukvervorming in % vgl. DIN 53517</td>
<td>3 - 10</td>
<td>5 - 12</td>
<td>15 - 25</td>
</tr>
<tr>
<td>A) na 72 uur en 23 °C</td>
<td>5 - 25</td>
<td>20 - 30</td>
<td>25 - 35</td>
</tr>
<tr>
<td>B) na 24 uur en 70 °C</td>
<td>300 - 400</td>
<td>200 - 300</td>
<td>400 - 600</td>
</tr>
</tbody>
</table>

Speciaal voor etiketteermachines kunnen platen sponsrubber worden geleverd. Kleur: oranje. In stramheden van 140 - 500 kg/m² in de dikte 5 t/m 60 mm. De platen zijn zonder huid en zijn opencellig.

Pakkingen en of stroken op maat kunnen in eigen stanserij worden vervaardigd.

Sponsrubbersnoer (rond)
Sponsrubbersnoer (rond met huid) wordt in twee kwaliteiten in voorraad gehouden nl. Neoprene (CR) en natuurrubber (NR).
Neoprene rubber is ook olie- en verouderingsbestendig.

<table>
<thead>
<tr>
<th>Neoprene rubber (CR) zwart</th>
<th>Natuurrubber (NR) grijs</th>
</tr>
</thead>
<tbody>
<tr>
<td>diameter in mm</td>
<td>rol lengte in mtr.</td>
</tr>
<tr>
<td>3</td>
<td>100</td>
</tr>
<tr>
<td>4</td>
<td>100</td>
</tr>
<tr>
<td>5</td>
<td>100</td>
</tr>
<tr>
<td>6</td>
<td>100</td>
</tr>
<tr>
<td>8</td>
<td>50</td>
</tr>
</tbody>
</table>

Minimum afname: één rol.
Andere diameters en of kwaliteiten zijn op bestelling leverbaar, waarbij als minimum afname geldt: 100 à 200 meter.

Figure A.7:  Datasheet O-ring wire (in Dutch) -type 8x50
Source: Eriks
### Sponsrubberprofielen (rechthoekig en vierkant)

Sponsrubberprofiel (rondom met huid) wordt in twee kwaliteitlen in voorraad gehouden, nl Neoprene en Natuurrubber. Neoprene rubber is olie- en verouderingsbestendig, alsmede bestand tegen een temperatuur van ca. 90 à 100 °C.

<table>
<thead>
<tr>
<th>Neoprene rubber (CR) zwart</th>
<th>Natuurrubber (NR) grijs</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>b x d</strong></td>
<td><strong>b x d</strong></td>
</tr>
<tr>
<td>Rollengte in mtr.</td>
<td>Rollengte in mtr.</td>
</tr>
<tr>
<td>15 x 6</td>
<td>15 x 5</td>
</tr>
<tr>
<td>8 x 6</td>
<td>20</td>
</tr>
<tr>
<td>25 x 6</td>
<td>20</td>
</tr>
<tr>
<td>16 x 7</td>
<td>20</td>
</tr>
<tr>
<td>10 x 10</td>
<td>20</td>
</tr>
<tr>
<td>16 x 10</td>
<td>20</td>
</tr>
<tr>
<td>20 x 10</td>
<td>20</td>
</tr>
<tr>
<td>25 x 10</td>
<td>20</td>
</tr>
<tr>
<td>30 x 10</td>
<td>20</td>
</tr>
<tr>
<td>45 x 10</td>
<td>20</td>
</tr>
<tr>
<td>12 x 12</td>
<td>20</td>
</tr>
<tr>
<td>16 x 12</td>
<td>20</td>
</tr>
<tr>
<td>25 x 12</td>
<td>20</td>
</tr>
<tr>
<td>30 x 12</td>
<td>20</td>
</tr>
<tr>
<td>15 x 15</td>
<td>10</td>
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<tr>
<td>20 x 16</td>
<td>10</td>
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<tr>
<td>30 x 16</td>
<td>10</td>
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<td>20 x 20</td>
<td>10</td>
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<td>25 x 20</td>
<td>10</td>
</tr>
<tr>
<td>30 x 20</td>
<td>10</td>
</tr>
<tr>
<td>25 x 25</td>
<td>10</td>
</tr>
</tbody>
</table>

Speciaal voor de cartonnage-industrie houden wij in voorraad de afmeting 10 x 10 mm in de normale hardheid, kleur grijs, alsmede in een extra stramme uitvoering, kleur zwart. Beide in fabricage-lengten van ca. 3 meter.

**Figure A.8:** Datasheet O-ring wire (in Dutch)

*Source: Eriks*
Appendix G  Connecting plate design

Figure A.9: Previous connecting plate

Figure A.10: New connecting plate
Appendix H  Sealing

P/S® | [Model 61]
General purpose assembled seal for high pressure applications.

Features

- GYLON element offers excellent chemical resistance
- Dry running up to 700 fpm (3.5 m/s)

<table>
<thead>
<tr>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Minimum Shaft Diameter:</strong></td>
</tr>
<tr>
<td><strong>Maximum Shaft Diameter</strong></td>
</tr>
<tr>
<td><strong>Misalignment &amp; Runout</strong></td>
</tr>
<tr>
<td><strong>Pressure</strong></td>
</tr>
<tr>
<td><strong>Spring Configuration</strong></td>
</tr>
</tbody>
</table>

Figure A.11: Datasheet seal; type: P/S model 61
Source: Garlock
References:


