5 Haulage dynamics

Hopefully the buffers are rarely called into play. However, the haulage system is in constant use, and it is to this that I shall now turn.

The lift is raised and lowered by the wire ropes, which run down from the motor and brake equipment at the head of the lift well. Figure 25(a) shows the simplest arrangement for this, with the wires being wound on a rotating drum. Though this is the simplest drive it has several disadvantages of varying seriousness. One is that the drum becomes heavy and cumbersome as the number of floors served increases. The effect is made worse by the Standards requirement that at least two ropes be used. The ropes must be wound into grooves to stop rubbing, and the rope must not be wound layer upon layer or the strands would damage each other. We shall see shortly the advantage of employing a counterweight which descends as the lift car is pulled up; accommodating the counterweight rope on the drum would make it even longer, even if we do something ingenious, as in Figure 25(b).

The last and most devastating drawback to drum haulage, and one which serves to limit its use to lifts whose speed does not exceed 0.63 m s⁻¹, is a safety consideration. In the safety requirements a situation is envisaged in which the lift reaches the top of its travel but the drum is not brought to rest, owing to malfunction of the limit switches, the motor control and the brakes. If that were to happen the lift car would be drawn inexorably into the head-gear. Equipment would be damaged, the ropes broken and possibly even lives endangered. Hence on lifts running at a speed of 0.63 m s⁻¹ or more, a drive called 'traction drive' is used.

In traction drive the set of ropes rising up from the car merely passes over a special pulley called a sheave, at the head, and then runs down to the counterweight (Figure 26). The drive and the braking are therefore not positive as when a drum is used — rather the motion of the sheave is transmitted to the rope by friction. The friction is arranged so as to be sufficient for normal drive or braking, but to slip if the need arises, such as overwinding. We shall examine this slipping later, but as we shall find it to be dependent on the size of counterweight, that ought to be looked at next.

The counterweight serves to reduce to a minimum the traction and braking effort required. Conventionally its mass is chosen so as to balance the mass of the empty lift plus 40–50% of the rated load. The precise figure that is adopted for a particular installation is seemingly derived from the desire to make the power requirement the same in the following two cases: (1) raising a full lift from the lowest level, (2) raising the counterweight when the empty lift is at the top. Another argument which could be advanced is that the passenger load in the lift varies between zero and 90% of the rated load, so it would seem logical to balance by the counterweight the empty lift and the average load — 45% of the rated load. We shall adopt this result, if not the argument.

Movement of the car and counterweight, by control of velocity, acceleration and retardation, stem from the machine room at the head of the lift well. There a motor and brake share a common shaft, on which the sheave is also mounted. Control of the rotating sheave effects control of the car and counterweight through the interconnecting medium of the tensions in the respective sets of ropes. What are these tensions? They are sufficient to provide accelerations of the car and counterweight masses, through Newton's Second Law. So we follow the familiar procedure of separating the parts of the system into free-body diagrams and inserting the appropriate quantities that we know or seek. Figure 27 shows free-body diagrams for the car and counterweight, using the following information (the rope mass is omitted for simplicity):
Mass of rated load 1200 kg denoted by $m$
Mass of empty car, sling and fittings 1680 kg is equal to $1.4m$
Mass of counterweight 2220 kg, given by $1.4m + 0.45m = 1.85m$
Upward acceleration of car $a$ (equals downward acceleration of counterweight)

Rope tension at car $T_1$
Rope tension at counterweight $T_2$

Applying $\mathbf{R} = m\mathbf{a}$ (Block 4 Dynamics procedure):

For the car along $y$:
$$T_1 - 2.4mg = 2.4ma$$
$$T_1 = 2.4m(g + a)$$

Counterweight, along $y$:
$$T_2 - 1.85mg = -1.85ma$$
$$T_2 = 1.85m(g - a)$$

For our lift $m = 1200$ kg, $a = 1.5 \text{ m s}^{-2}$ maximum.

Taking $g = 9.81$ m s$^{-2}$ we have
$$T_1 = 2.4 \times 1200 (9.81 + 1.5) = 32 573 \text{ N} \approx 32.6 \text{ kN}$$
$$T_2 = 1.85 \times 1200 (9.81 - 1.5) = 18 448 \text{ N} \approx 18.4 \text{ kN}$$

The motor must supply sufficient torque to its shaft and to the sheave, and the sheave must not slip under the rope when the rope carries these tensions. I shall look at the torque later. Meanwhile what are the conditions for the rope slip? We shall need to extend the sliding friction analysis that you have done in earlier Units. The 'dry' limiting friction model will still be applied, but the contact is between surfaces that are far from flat.

The sheave cannot be a simple cylinder. It must have grooves, to locate the rope and provide enough friction. Here we shall examine only the simple V-groove but more complicated groove profiles are used too. Figure 28(c) shows the sheave with the circular-section rope lying in the V-groove. Imagine it for yourself — try it even — a rope or string around a convenient chair leg (or tree!). The tensions determine whether the rope slips, along with the other factors such as the ‘wrap’ of the rope — the angle subtended by the part of the circumference over which contact is made ($\theta$ in Figure 28(a)). We need to quantify this relationship to ensure that the haulage dynamics are acceptable, and also of course to design the sheave.

The rope tensions at each side of the sheave are $T_1$ and $T_2$ ($T_1 < T_2$). $\theta$ is the ‘wrap’. We need to examine a very short length of rope — the piece BC. The tension at C I shall call $T$. At B it is a little greater, $T + \delta T$.

In the plane of the sheave there is a force of magnitude $R$ exerted on the small piece of rope by the sheave (Figure 28(b)), and also a friction force opposing the extra bit of tension $\delta T$. We will expect the rope to slip if $\delta T$ exceeds the limiting friction force $F_{max}$. The magnitude of this limiting friction force is given by $\mu N$ where $\overline{N}$ is the normal reaction. This is not just $\mathbf{R}$, because the rope is in the V-groove (Figure 28(c)). $\mathbf{R}$ is the resultant of the two normal forces $\overline{N}$ (Figure 28(d)).

$$R = 2N \sin \alpha$$
so
$$N = \frac{1}{2}R/\sin \alpha$$
$$F_{max} = 2\mu N = \mu R/\sin \alpha$$  (two $N$ forces)

We still do not know $R$, but this can be found from the FBD of Figure 28(e).
Along \( y \):

\[
R - T \sin \frac{1}{2} \delta \theta - (T + \delta T) \sin \frac{1}{2} \delta \theta = ma,
\]

\( a \) is the acceleration radially, which is the centripetal acceleration \( \omega^2 r \). This is not zero, but it is insignificant in this case and can safely be neglected (this is not true of all pulleys by any means).

You should remember from Block 1 that for a small angle

\[
\sin x \approx x
\]

So our equation can be simplified to

\[
R - T \frac{1}{2} \delta \theta - (T + \delta T) \frac{1}{2} \delta \theta = 0
\]

so

\[
R - T \delta \theta - \frac{1}{2} \delta T \delta \theta = 0
\]

The \( \delta T \delta \theta \) term is the product of two very small quantities, so it can be neglected compared with the others, leaving

\[
R - T \delta \theta = 0
\]

so

\[
R = T \delta \theta
\]

which is what we were looking for.

The equation \( F_{\text{max}} = \mu R / \sin \alpha \) becomes

\[
F_{\text{max}} = \mu T \delta \theta / \sin \alpha
\]

and the rope will be on the verge of slipping when

\[
\delta T = F_{\text{max}} = \mu T \delta \theta / \sin \alpha
\]

so

\[
\frac{\delta T}{\delta \theta} = \frac{\mu}{\sin \alpha} T
\]

Expressing this as a proper differential

\[
\frac{dT}{d\theta} = \frac{\mu}{\sin \alpha} T
\]

which is a satisfyingly simple equation to derive from the seemingly rather complicated physical arrangement. The solution can be looked up in any mathematics handbook. It is

\[
\frac{T_1}{T_2} = e^{\mu \theta / \sin \alpha}
\]

where \( e = 2.718 \). This is the relationship we need in order to estimate the performance of the sheave.

As might be expected from common experience, the higher the value of \( \theta \) (the wrap), the higher the ratio of tensions. The whole principle of capstans, bollards for ships, and even knots is to use a high value of \( \theta \) so that a small \( T_2 \) can sustain a large \( T_1 \) without slip. If \( \theta \) is doubled, the tension ratio limit is squared. The ratio of the tensions also increases when \( \alpha \) is decreased, but if \( \alpha \) is reduced too much the rope is squashed, worn and finally broken by the pinching action of the groove, so there is a definite practical limit.

Returning to the drive problem, we see that for a particular lift installation the values of \( \theta \) and \( \alpha \) are fixed, and so is \( \mu \) unless the lubrication between rope and sheave is changed, so \( e^{\mu \theta / \sin \alpha} \) can be calculated. The ratio of the tensions in the ropes at the sheave cannot exceed this and no larger ratio than this ‘available traction’ should be demanded from the system or slip will begin and positive control of lift position and motion will be lost. In particular the acceleration profiles that we desired (Section 1) required a certain rope tension and hence traction requirement. For our lift (Figure 27), the traction required is

\[
\frac{T_1}{T_2} = 32.573
\]

\[
\frac{T_1}{T_2} = 18.448 = 1.77
\]

24
We can now investigate whether this ratio is available from a simple sheave. If not, the lift won't work. The value of $\mu$ for a steel cable on a cast iron pulley varies a good deal, from 0.06 if the cables are greasy, to 0.15 for lightly loaded dry cables and 0.40 for heavily loaded dry cables. From this it seems that rope friction is not quite independent of load, presumably on account of the flexibility of the cables. We will not follow up this thought here. The value of $\alpha$ cannot be much less than 20° or the rope will be worn badly through being 'pinched'. The angle $\theta$ will be $\pi$ rad (180°).

$$e^{\theta \sin \alpha} = e^{0.06 \times \pi \sin 20°} = e^{0.55} = 1.73$$

is the largest traction that can be guaranteed. It is inadequate. The solution adopted is to increase $\theta$ (Figure 29). The ropes from the lift pass up the well and over the sheave as before but then pass around a loosely running idler pulley. The ropes next run over the sheave again, using grooves between those used on the first pass. After the second pass over the sheave the ropes run over the same idler pulley again and thence to the counterweight. The angle $\theta$ is now the sum of 0.75$\pi$ radians (approximately, depending on layout) for the first pass, and $\pi$ for the second, giving $\theta = 1.75\pi$. The 'traction available' is now

$$e^{0.06 \times 1.75\pi \sin 20°} = e^{0.964} = 2.62$$

Slip is eliminated, even with a greasy rope. Double wrap subjects the rope to more flexure, and would shorten its life, but a small increase in the diameters of the sheave and pulley will make up for this. The angle of wrap on the idler pulley is of no special significance because the idler pulley does not affect the tension — it is not driven separately.

So far in this section we have only been concerned with obtaining sufficient traction from the sheave. Now we must consider a design case in which the traction must be low, so low in fact that the sheave slips on the rope surface — that is why the rope is not simply wrapped several times around a drum. This is a key safety feature of the traction drive, giving it a clear advantage over drum winding. The idea is that if all the electrical safeguards should fail when the lift is approaching the upper limit of its travel, then the counterweight bottoms and the tension in the counterweight ropes falls to such a low value that traction is lost. The rope slips on the sheave and the lift comes to a halt. Although the counterweight is now ineffective there is still a tension in the top of the counterweight ropes: the rope's own weight. In this analysis we have often implicitly set aside the rope weight as being too small for inclusion in our rough calculations. Now we must include it.

The counterweight rope is virtually 30 m long. The ropes for a lift like this would be typically six in number, each of 13 mm diameter. The corresponding mass would be 112 kg. Opposing this is the mass of the empty lift, 1680 kg. The equations of motion are, for a car acceleration $a$ as shown in Figure 30:

For the lift, along $y$:

$$T_4 - 16481 = 1680a$$

so

$$T_4 = 16481 + 1680a$$

For the counterweight rope, along $y$:

$$T_3 - 1099 = -112a$$

so

$$T_3 = 1099 - 112a$$

The traction ratio is:

$$\frac{T_4}{T_3} = \frac{16481 + 1680a}{1099 - 112a}.$$
If $a$ is positive, the lift is being driven on unchecked. $T_a/T_3$ would be required to exceed $16.48/1099 = 15$. We could call this the traction ratio required for a disaster. Now the ropes are only lightly loaded and if we assume they are dry the coefficient of friction is 0.15. So the largest traction ratio available with double wrap and $2a = 40^\circ$ is

$$e^{0.15 \times 1.75 \times 2.02} = e^{2.41} = 11.1$$

This represents the available traction; no more can be provided by the sheave because slipping begins. The available traction is less than that needed for a disaster to occur. Good.

Now we have a design of sheave which can safely drive the lift. However, we still need to make it move. Power is obviously needed. This is one of the features specified at purchase, so we need to be able to calculate it. The motor driving the sheave will need a supply of electric current which must be supplied and paid for. Also the motor will heat up and so will the brake because it converts kinetic energy from the system into thermal energy, which must be expelled from the machine room and into the atmosphere. Not only might equipment be damaged by excessive heat, but maintenance engineers will need to enter the machine room.

An estimate of the motor power is conventionally obtained as follows. Essentially the motor is simply regarded as a source of power for raising the out-of-balance load (tension difference) at the constant rated speed. For the lift in this study the maximum out-of-balance tension on the sheave is (from Figure 31):

- Fully loaded lift at bottom of travel:
  $$T = 16.48 + 11.77 + 1.10 - 21.78 = 7.57 \, \text{kN}$$

- Empty lift at top of travel:
  $$T = 21.78 + 1.10 - 16.48 = 6.40 \, \text{kN}$$

The rated speed is 1.5 m s$^{-1}$ so the higher power requirement is

$$7.57 \times 10^3 \times 1.5 = 11 \, 355 \, \text{W} \approx 11.4 \, \text{kW}$$

This is by no means the whole story. At the cage and counterweight further power is needed to overcome guide friction and air resistance. Power is lost in bending and unbending the rope as it passes over the sheave and other pulleys. There will be aerodynamic drag on the sheave, brake drum, ropes, car and so on. Bearings will possess friction and the car guide rollers will have rolling resistance. The net result of all these and other losses is that the lift needs some 18 kW to run at rated speed.

It may seem odd that we have not yet made mention of the power required to provide acceleration in the moving parts. Briefly the answer is that when running at rated speed there is no acceleration and hence no kinetic energy change, so the power that has been calculated is a figure for steady operation at the highest speed. As regards heat build-up and dissipation, this continuous rating power is the important figure. Nevertheless, power is needed to overcome the inertia of the moving parts and give them the requisite kinetic energy. A glance at the velocity and acceleration profiles in Section 1 shows that the acceleration is needed when the speed is below the rated value. So in effect a swap takes place. As the motor builds up speed, power is used mainly for acceleration. At rated speed power is used to maintain speed only. Actually an exact swap will not occur, and for brief periods of less than a second the power demanded from the motor will exceed the continuous rating. Electric motors cope with this.

Power is only one of the mechanical parameters that must be examined for the motor. The other is torque. To evaluate the torque we could draw a whole set of free-body diagrams for the components of the lift. But
rather than do this I shall try to talk you through the system rather as an experienced engineer might think it through informally.

Consider a full car being accelerated from the lowest floor (Figure 31a). The unbalanced load is obtained from the weights of the car and sling, the rated load, the counterweight and hanging portion of rope:

\[ 16.48 + 11.77 + 1.10 - 21.78 = 7.57 \text{ kN} \]

The masses to be accelerated at a maximum of 1.5 m s\(^{-2}\) are those items just mentioned, except that we now need the full length of the rope, which includes the portion around the sheave and pulley, say 40 m in all. The total mass is 1680 + 1200 + 149 + 2220 = 5249 kg. The force needed for acceleration is \( F = ma = 5249 \times 1.5 = 7.87 \text{ kN} \). The torque required at the sheave shaft to handle these loads and accelerating masses in the well will depend on the sheave radius. This would be typically 0.26 m for our lift, so the torque is

\[
M = F \times r = (7.57 \text{ kN} + 7.87 \text{ kN}) \times 0.26 \text{ m} = 4.01 \text{ kN m}
\]

Guide friction, air resistance and other losses total some 10% of this, so we need about 4.5 kN m. The sheave has a second moment of mass and so do other components on the same shaft. A gearbox must be interposed between the sheave and the motor so that when the sheave is running at a speed to suit our kinematic profiles, the motor runs at its best speeds. The gearbox used is the 'worm' type (Figure 32). Here a worm resembling a coarsely threaded screw shares the motor shaft with the brake drum. The worm meshes with a large gear wheel on the sheave shaft. The sheave and the gear wheel on the sheave shaft together have a second moment of mass \( I = 30 \text{ kg m}^2 \). These parts have an angular acceleration

\[
a = \frac{a}{r} = 1.5/0.26 = 5.77 \text{ rad s}^{-2}
\]

The idler pulley (Figure 28) has the same angular speed and acceleration with \( I = 12 \text{ kg m}^2 \). The sheave shaft torque required to accelerate these will be

\[
M = Iz = (30 + 12) \times 5.77 = 242 \text{ N m}
\]

Suppose the motor shaft runs at 20 \( \times \) the speed of the sheave shaft, then the input torque to the gearbox would be

\[
M = (4500 + 242)/20 = 237 \text{ N m}
\]

The worm and its shaft alone have \( I = 1.3 \text{ kg m}^2 \) and an acceleration

\[
a = 5.77 \times 20 = 115 \text{ rad s}^{-2}
\]

They need a torque of \( Iz = 150 \text{ N m} \) giving a total 387 N m. Unfortunately a worm-style gearbox is not very efficient, the conditions of contact between the gear teeth reduce the efficiency to say 0.87, so the torque needed from the motor rises to

\[
M = 387/0.87 = 445 \text{ N m}
\]

Armed with this analysis we could return to the velocity and acceleration profiles and calculate power and torque profiles. For example, consider the condition one second after a loaded lift leaves the bottom. Our kinematic profiles derived for constant jerk show the acceleration to be still 1.5 m s\(^{-2}\) so the motor torque is 445 N m. Lift speed is 1.0 m s\(^{-1}\), sheave angular velocity

\[
\omega = 1/0.26 = 3.85 \text{ rad s}^{-1}
\]

and the motor angular velocity is

\[
\omega = 20 \times 1/0.26 = 77 \text{ rad s}^{-1}
\]
The power input to the motor is

\[ 445 \text{ N m} \times 77 \text{ rad s}^{-1} = 34 \text{ kW} \]

This analysis could be extended so as to find how the torque and power must vary with speed. An electrical engineer could then go on to design the control gear.

Before we leave this analysis with these loose ends, notice how what has gone before is now tied together. Human considerations of passenger comfort led us to particular kinematic profiles, and now the kinetics might force us to reconsider the whole design if insufficient electrical power were available in the machine room.

6 Conclusion

If you have followed the Unit to this stage you will have renewed your acquaintance with many of the concepts and techniques introduced and taught earlier in the course. What I have done here is to begin at one of the possible starting points for analysing or designing a lift installation. A succession of explorations and calculations showed how each area of investigation, the lower beam of the sling say, became the input to another investigation. So we see that this tour of the mechanics of lifts provides examples of the strong links between the different branches of mechanics. To those who only learn about mechanics and are not regular practitioners, these branches may appear as isolated treatments. This Unit has set out to show that for a purpose-oriented task such as lift engineering a succession of mechanics activities is engaged in, each linked to others. Applied to real analysis or design, Engineering Mechanics is always like this, linked together as a system.

Engineering Mechanics is principally about making things do what you want them to do. It is a purposeful activity in which material objects are made to serve human needs. Whoever you are — engineer, client or just a passenger in a lift — mechanics is worth knowing about.

Reference

Further details and information on electric lifts can be found in the British Standards document BS5655: Part 1: 1986 'Lifts and service lifts. Part 1 Safety rules for the construction and installation of Electric Lifts'.

Acknowledgement

The Course Team wishes to acknowledge with grateful thanks the assistance received from informal discussions between the Express Lift Company of Northampton and members of the T232 Course Team. This account should not, however, be taken to represent the practice of any particular manufacturer.
## Data Sheet

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Floors served</td>
<td>8</td>
</tr>
<tr>
<td>Levels served</td>
<td>9</td>
</tr>
<tr>
<td>Total travel</td>
<td>26.67 m</td>
</tr>
<tr>
<td>Interfloor distance</td>
<td>3.33 m</td>
</tr>
<tr>
<td>Rated speed</td>
<td>1.5 m s⁻¹</td>
</tr>
<tr>
<td>Uniform acceleration</td>
<td>0.9 m s⁻²</td>
</tr>
<tr>
<td>Rated load</td>
<td>1200 kg</td>
</tr>
<tr>
<td>Passenger load</td>
<td>16</td>
</tr>
<tr>
<td>Expected occupancy 90%</td>
<td>14.4</td>
</tr>
<tr>
<td>Average number of stops</td>
<td>6.9</td>
</tr>
<tr>
<td>Door open + close time</td>
<td>4 s</td>
</tr>
<tr>
<td>Passenger entry or exit</td>
<td>1.2 s</td>
</tr>
<tr>
<td>Round trip</td>
<td>115 s</td>
</tr>
<tr>
<td>Round trip + 10%</td>
<td>127 s</td>
</tr>
<tr>
<td>Journeys in 5 min</td>
<td>2.36</td>
</tr>
<tr>
<td>Handling capacity in 5 min (1 lift)</td>
<td>34</td>
</tr>
<tr>
<td>Number of lifts</td>
<td>4</td>
</tr>
<tr>
<td>Handling capacity in 5 min (system)</td>
<td>136</td>
</tr>
<tr>
<td>Departure interval</td>
<td>32 s</td>
</tr>
<tr>
<td>Average wait</td>
<td>16 s</td>
</tr>
<tr>
<td>Figure of merit</td>
<td>62 s</td>
</tr>
<tr>
<td>Jerk limit</td>
<td>2 m s⁻³</td>
</tr>
<tr>
<td>Peak acceleration</td>
<td>1.5 m s⁻²</td>
</tr>
<tr>
<td>Car floor area</td>
<td>2.8 m²</td>
</tr>
<tr>
<td>Interior height</td>
<td>2.2 m</td>
</tr>
<tr>
<td>Interior width</td>
<td>2.0 m</td>
</tr>
<tr>
<td>Interior depth (front to back)</td>
<td>1.4 m</td>
</tr>
<tr>
<td>Car mass + platform</td>
<td>1130 kg</td>
</tr>
<tr>
<td>Sling mass</td>
<td>350 kg</td>
</tr>
<tr>
<td>Safety gear mass</td>
<td>200 kg</td>
</tr>
<tr>
<td>Estimated BM in buffer beam</td>
<td>10.33 kN m</td>
</tr>
<tr>
<td>Buffer beam safety factor</td>
<td>5</td>
</tr>
<tr>
<td>Buffer beam channels</td>
<td>152 × 76 mm</td>
</tr>
<tr>
<td>Rope set</td>
<td>6 × 13 mm</td>
</tr>
<tr>
<td>Rope hanging length</td>
<td>30 m</td>
</tr>
<tr>
<td>Rope total length</td>
<td>40 m</td>
</tr>
<tr>
<td>Rope hanging mass</td>
<td>112 kg</td>
</tr>
<tr>
<td>Rope total mass</td>
<td>149 kg</td>
</tr>
<tr>
<td>Empty car + sling</td>
<td>1680 kg</td>
</tr>
<tr>
<td>Full car + sling</td>
<td>2880 kg</td>
</tr>
<tr>
<td>Counterweight</td>
<td>2220 kg</td>
</tr>
<tr>
<td>Traction required (drive)</td>
<td>1.77</td>
</tr>
<tr>
<td>Traction required (overwind)</td>
<td>15</td>
</tr>
<tr>
<td>Traction available (drive)</td>
<td>2.62</td>
</tr>
<tr>
<td>Traction available (overwind)</td>
<td>11.1</td>
</tr>
<tr>
<td>Sheave diameter</td>
<td>0.52 m</td>
</tr>
<tr>
<td>Maximum tension imbalance</td>
<td>7.57 kN</td>
</tr>
<tr>
<td>Sheave speed</td>
<td>5.77 rad s⁻¹</td>
</tr>
<tr>
<td>Sheave shaft 1</td>
<td>30 kg m²</td>
</tr>
<tr>
<td>Sheave shaft acceleration</td>
<td>5.77 rad s⁻²</td>
</tr>
<tr>
<td>Idler pulley 1</td>
<td>12 kg m²</td>
</tr>
<tr>
<td>Gear ratio</td>
<td>20</td>
</tr>
<tr>
<td>Wormshaft 1</td>
<td>1.3 kg m²</td>
</tr>
<tr>
<td>Wormshaft acceleration</td>
<td>115 rad s⁻²</td>
</tr>
<tr>
<td>Worm gear efficiency</td>
<td>0.87</td>
</tr>
<tr>
<td>Motor torque requirement</td>
<td>445 N m</td>
</tr>
<tr>
<td>Power for rated speed</td>
<td>11.4 kW</td>
</tr>
<tr>
<td>Installed motor power</td>
<td>18 kW</td>
</tr>
<tr>
<td>Power input peak</td>
<td>34 kW</td>
</tr>
</tbody>
</table>