# KLINGER

guesswork out of gaskets

Published by
Klinger International Export Ges. m. b. H.
on behalf of the companies of the Klinger
Group.

# KLINGER

## takes the guesswork out of gaskets

The new Klinger dimensioning method gives you quick solutions to many gasket problems . . .

Successful gasket design involves three basic elements:

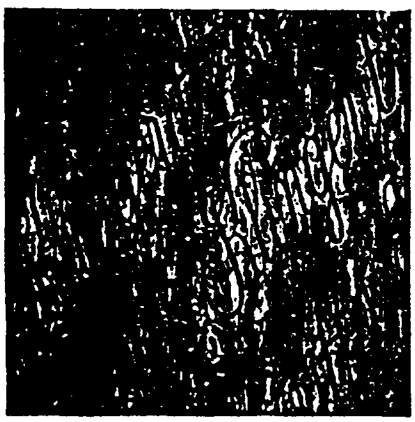
The right gasket material
The right gasket dimensions
The right gasket surface stress

Choosing the material is usually straightforward since your Klinger catalogue gives recommendations for most common media. Getting the dimensions and stresses right is more difficult and in some cases requires the help of specialist gasket engineers, but there are plenty of everyday problems that you can solve without being a gasket specialist, as this booklet shows.

The data in this book have been compiled by ISTAG AG, central research organization of the Richard Klinger group of companies. They apply solely to the Klinger materials KLINGERIT ®, KLINGERIT 400 UNIVERSAL® and KLINGEROILIT®.

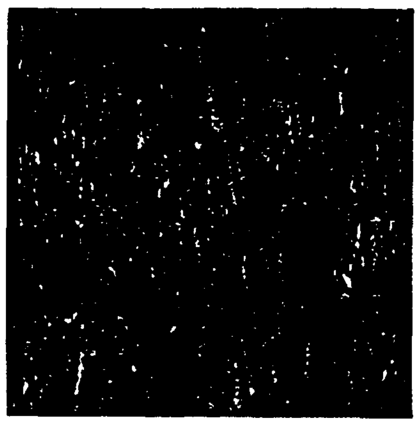
## Klingerit

A top quality sheet for general use, composed of asbestos fibre with SBR binder. Used mainly on steam services but has excellent resistance to many oils and chemicals. Meets many UK, US and other specifications including BS 1832, BS 2815 A, DIN 3754 (It 400) and ASTM D 1170 Grade P.1161 A. Normally recommended maxima: 550° C (1,000° F) and 1750–2,000 psi (120–140 atm).



## Klingerit 400 Universal

A high temperature/high pressure sheet with outstanding resistance to oils, fuels and lubricants. Composed of asbestos with NBR binder. Has controlled low chloride content making it especially suitable for stainless steel flanges. Specification compliance includes DTD 378 A, BS 2815 A, DIN 3754 (It C) and ASTM D.1170 Grade P.1141 A. Normally recommended maxima: 550° C (1,000° F) and 2,000 psi (140 atm).



## Klinger-Oilit

A high pressure sheet with excellent resistance to hot oils, hydrocarbons etc. Composed of asbestos with NBR binder. Klinger-Oilit is widely used in the oil and chemical industries and meets BS 2815 A, DIN 3754 (It-Oe) and ASTM D.1170 Grade P.1141 A. Normally recommended maxima: ca. 500° C (900° F) and 1,500 psi (100 atm).

# How gaskets seal

(Minimum surface stress at working pressure)

If flange surfaces mated perfectly there would be no need for gaskets.

In practice, flanges always have slight surface irregularities, and a compressible, resilient material – the gasket – is used to compensate for them. This provides an uninterrupted barrier against the medium and compensates for slight movement of the flanges during service.

Even at low internal pressures the gasket must be pressed against the flanges with a definite minimum surface stress. This "deformation stress" depends on the structure and compressibility of the gasket material; for Klingerit, Klingerit 400 Universal and Klinger-Oilit it is

750 psi (50 kp/cm²) for liquids
3,000 psi (200 kp/cm²) for gases

At higher internal pressure ($p_i$) additional gasket stress is necessary. This gives a total minimum surface stress ($p_{d_{min}}$) at working pressure:

(1) for liquids
$p_{d_{min}}$ = 750 psi (50 kp/cm²) + 2.5 $p_i$

(2) for gases
$p_{d_{min}}$ = 3,000 psi (200 kp/cm²) + 4 $p_i$

Values of $p_{d_{min}}$ calculated from these formulae are given in Tables 3a and 3b (pages 13 and 14).

The factors 2.5 and 4 are generally called "gasket factors".

Note:

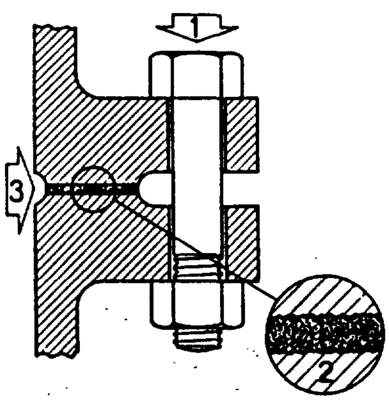
Formula 1 also applies to many saturated vapours which condense at ambient temperatures.

Formula 2 should be modified for highly toxic or dangerous gases by increasing the gasket factor (to a maximum of about 8).

With easily deformable flanges or where the available bolt load is low, a softer material such as Klingerflex-A 10, Klingerflex-CB 1 or Klingerpac may be advisable (low to moderate pressures only).

These formulae are not significantly dependent on gasket thickness within the range 0.020" – 1/8" (0.5 – 3 mm). There is, however, a definite relationship between the degree of flange surface roughness and the minimum gasket thickness (see Table 1, page 12).

Table 3 (pages 13 and 14) gives values of $p_{d_{min}}$ for various values of $p_t$. Use Table 3a for liquids, 3b for gases. Intermediate values can be interpolated.




1. Flange bolts provide gasket surface stress
2. "Deformation stress" is required even at zero internal pressure
3. Additional gasket stress is needed to withstand internal pressure



UTEX 001148

## The hydrostatic end thrust (bursting thrust)

In closed vessels the internal pressure exerts a thrust on the cover or lid – the hydrostatic end thrust. This applies also in closed pipelines, where the hydrostatic end thrust tends to pull the flanges apart. This reduces the stress originally applied to the gasket (assembly stress).

The assembly stress must therefore:
Compensate for the effect of the hydrostatic end thrust;
and maintain the minimum gasket surface stress needed to seal at working pressure ($p_{d_{min}}$).

i. e. minimum assembly stress $= p'_{d_{min}} = p_{d_{min}} + \dfrac{\text{hydrostatic end thrust}}{\text{area of gasket}}$

Hydrostatic end thrust = internal pressure ($p_i$) X internal area

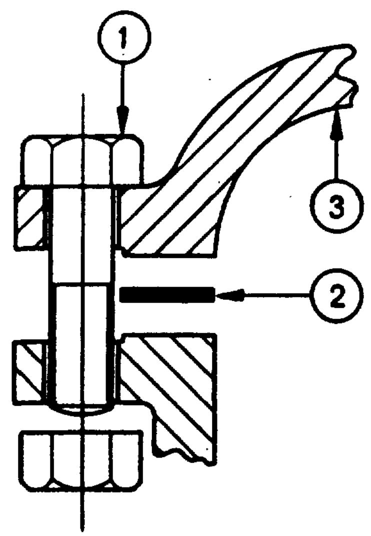
For circular gaskets (ignoring bolt holes):

$$\text{Minimum assembly stress} = p_{d_{min}} + p_i \cdot \frac{\pi d_i^2/4}{\pi b d_m}$$

$$= p_{d_{min}} + p_i \cdot \frac{d_i^2}{4 b d_m}$$

where $d_i$ = inside diameter
$b$ = radial breadth of gasket
$d_m$ = mean diameter ($= d_i + b$)

Note that the hydrostatic end thrust increases with the square of the inside diameter.




1. Assembly load produced by bolts or clamps
2. Radial force produced by internal pressure
3. Hydrostatic end thrust



UTEX 001149

## Choosing the right gasket thickness

Compressed asbestos materials have a slight porosity, so gaskets should be as thin as possible. This reduces diffusion losses and also the area exposed to attack by aggressive media (oils, chemicals etc.).

Since the gasket must compensate for the surface roughness of the flanges, the minimum thickness depends on:

Depth of flange surface roughness

Compressibility of the gasket

Gasket surface stress at working pressure.

The minimum thickness is:

$$\frac{2 \times \text{maximum depth of flange surface roughness} \times 100}{\text{compressibility (\%) at given surface stress}}$$

Depth of flange surface roughness is commonly expressed as a degree of surface finish ( ▽,▽▽ etc.). Using this value and the already calculated value of $p_{d_{min}}$, the minimum gasket thickness can be taken from Table 1 (page 12) without further calculation.

The values given in Table 1 take account of the roughness of both flange surfaces: as this is not entirely necessary they contain a safety margin. Gasket thicknesses are given as the nearest standard Klinger sheets, so inch and metric thicknesses do not correspond exactly. Thicker materials can of course be used but it is technically best as well as economic to keep to the minimum.

Turned flanges generally have peaked grooves which in effect reduce the area of gasket carrying the flange load. This gives an increased surface stress and increased gasket compression. In such cases the gasket thickness may be reduced to the values printed in BLUE (provided flanges are rigid and undistorted).

Note: For intermediate values of $p_{d_{min}}$ use the next THICKER gasket.

# Maximum gasket surface stress (minimum gasket area)

Too high a gasket surface stress can cause leakage. This is because the gasket loses the resilience needed to maintain its pressure against the flange surfaces. The surface stress on the gasket must never exceed the recommended maximum.

For a given material the maximum permissible surface stress depends mainly on the temperature and the thickness. Thin materials withstand higher stresses than thick ones, cold conditions permit higher stresses than hot. For maximum permissible surface stress on Klinger materials see Table 2 (page 12).

If the area enclosed by the gasket is large, the hydrostatic end thrust may be very large even at moderate internal pressure. This demands a correspondingly high bolt loading and the calculated minimum assembly stress (= bolt load/area of gasket) may exceed the permissible maximum for the given gasket thickness. There are three possible remedies:

1. Use a larger gasket. This is only possible when the gasket dimensions are not already fixed (e. g. by pipeline flange specifications such as BS, ASA, DIN etc).
2. Use a thinner material (and correspondingly better flange finish).
3. Use a material that withstands higher surface stresses e. g. wire-reinforced (Klingerit 1000) or spiral wound (Klinger-Metallit). Maximum permissible surface stress for Klingerit 1000 is about 25 % above that of Klingerit, Klingerit 400 Universal and Klinger-Oilit.

This complication is dealt with in Practical Example 2 (page 10).

With circular gaskets the minimum radial breadth can be found from Table 3 (pages 13 and 14). This table is in four parts: liquid cold, liquid hot, gas cold and gas hot. It gives the minimum radial breadth for various internal pressures as a ratio $b_{min}/d_i$.

**Example:**
Medium: hot gas
Internal pressure: 1,500 psi (ca 100 atm)
Inside diameter: 36in
Gasket thickness: $^1/_{32}$in

For these data Table 3b (gas hot) gives
$b_{min}/d_i = 0.05$
i. e. $b_{min} = d_i \times 0.05 = 1.8$in

Note that the radial breadth of the gasket should never be less than double the gasket thickness, to prevent the gasket from being crushed. For gases and other penetrating media the breadth should not be less than ½in (12 mm) or $^3/_8$in (9 mm) at very least. This is to prevent diffusion of the medium along the asbestos fibres.



UTEX 001151

# What prevents blow-out?

Gaskets are held in position by friction, not by their tensile strength. This is demonstrated by the following calculation.

The stability condition to resist blow-out is that the frictional force exceeds the radial force caused by the internal pressure i. e.

frictional force $>$ radial force

$2 \mu p_d \pi (d_i + b) b > p_i \pi d_i s$

where $\mu = 0.1$ = coefficient of static friction *

$s$ = thickness of gasket

$$2 \times 0.1 p \, \pi (d_i + b) b > p_i \pi d_i s$$

$$_d > p_i \frac{5 d_i s}{b (d_i + b)}$$

$$\text{or } p_d > p_i \frac{5 s}{b}$$

(if d is large compared to b.)

$p_d$ in fact always exceeds 2.5 $p_i$ (formulae 1 and 2, page 3). Also b is never less than 2 s, even in extreme cases. It follows that the stability condition is always fulfilled in a gasket assembly designed in accordance with the Klinger method. Only in the most extreme cases is a check-calculation of the frictional forces necessary. Note that the stability condition has been fulfilled without considering the tensile strength at all: for this purpose tensile strength is irrelevant.

* $\mu$ is in fact generally about 0.4; even with smooth flanges it is never less than 0.1. Gaskets should never be treated with oil or grease since this reduces the frictional force. If a non-stick material is required, Klinger jointings can be supplied with graphited surfaces.

# Practical example no. 1

This example shows how the Klinger dimensioning method speeds up calculation.

**Given**

| | |
|---|---|
| Pipeline diameter | : 5 in. (nominal) |
| Flanges | : B. S. 10 Table R (raised face) |
| Inside dia. of gasket | : $5^3/_8$ in * |
| Outside dia. of raised face | : 7 in * |
| (* laid down by BS 10) | |
| Flange finish | : ▽▽ (turned) |
| Medium | : Cold water |
| Working pressure | : 1,200 psi |
| Test pressure | : 1,800 psi |
| Gasket material | : Klingerit |

**Required**

Determine minimum gasket thickness and minimum assembly surface stress.

**Step 1**

Find the minimum surface stress required at test pressure. Table 3a (liquids), column 2, gives this as

$p_{d_{min}}$ = 5,250 psi (ca. 370 kg/cm²)

**Step 2**

Find the minimum thickness from Table 1 (page 12) for flange finish ▽▽ and $p_{d_m}$ = 5,250 psi. Since the value 5,250 psi lies between two of the values given in the table use the next THICKER material i. e. 0.040 in (1 mm), using the black figures. Since however the flanges are turned, the lower values of $^1/_{32}$" or 0.75 mm (blue figures) are permissible.

**Step 3**

Find the minimum assembly stress. As shown on page 5 this is given by the formula:

minimum assembly surface stress

$$= p_{d_{min}} + p_i \cdot \frac{d_i^2}{4\,b\,d_m}$$

Note: b = radial breath = ½ (outside dia − inside dia)

$d_m$ = mean diameter = (inside dia. + b)

Hence minimum assembly surface stress

= 5,250 + 2,570 psi

= 7,820 psi (ca. 555 kg/cm²)

**Check**

It is essential to check that the calculated minimum assembly surface stress does not exceed the maximum permissible stress for material of the selected thickness. Table 2 (page 12) shows that the maximum permissible surface stress (cold) on 0.040 in Klingerit is 20,000 psi (ca. 1,400 kp/cm²). This is well above the calculated minumum assembly stress of 7,820 psi so the design is acceptable.

The actual assembly stress (and hence the surface stress under working pressure) is normally chosen in excess of the minimum value and in such cases this higher stress must be checked against the permissible maximum.

Practical Example No. 2 deals with a case in which the calculated minimum stress exceeds the permissible maximum.

# Practical example no. 2

This example introduces the complication of a large hydrostatic end thrust.

**Given**

A pressure vessel is being designed as follows –
Internal diameter : 1,000 mm
The cover is attached by flanges having a raised face (gasket seating face) of radial breadth 20 mm. This surface will be milled to a finish of ▽. Medium is cold gas, internal pressure ($p_i$) = 
10 80 atm (ca 1,150 psi).

**Required**

Minimum gasket thickness and minimum assembly stress.

**Step 1**

Using Table 3b (gas) the minimum surface stress at working pressure is found to be $p_{d_{min}}$ = 520 kp/cm²

**Step 2**

For minimum stress at working pressure $p_{d\,min}$ = 520 kp/cm² and flange finish = ▽, Table 1 gives minimum gasket thickness = 3 mm.

**Step 3**

Minimum assembly stress

$$= p_{d\,min} + p_i \cdot \frac{d_i^2}{4\,bd_m}$$

$$= 520 + 960 = \text{ca } 1{,}500 \text{ kp/cm}^2$$

**Check**

Table 2 shows that for 3 mm material the maximum permissible surface stress is 800 kp/cm² (cold). Since the calculated minimum assembly surface stress is higher than the permissible maximum the design is not acceptable. The possible remedies are now considered in turn:

**Larger gasket**

If the vessel is still at the design stage this may be possible. To determine the minimum radial breadth required use Table 3b (cold gas). For 3 mm gasket thickness and $p_i$ = 80 atm this gives $b_{min}/d_i$ = 0.20
i. e. $b_{min}$ = 1,000 x 0.20 = 200 mm
If an even broader gasket is used, a further check must be made that the resulting stress at working pressure does not fall below the required minimum.

**Thinner gasket**

In this example the simplest remedy would be to use smoother flanges and a thinner gasket capable of withstanding a higher surface stress. Suppose the flanges are now machined to ▽▽. Table 1 shows that the minimum gasket thickness is now 0.75 mm. This material can accept surface stresses up to 1,600 kp/cm² (cold) so the gasket design is now satisfactory.

**Other materials**

Klinger gasket engineers will gladly give advice in cases where neither of the above remedies proves practicable.

# General notes

Gasket assemblies depend on many factors that are outside the control of the gasket manufacturer. These recommendations are therefore a guide — not a guarantee of success. Here are some other factors to consider.

### Flanges

These should be even parallel and sufficiently rigid not to be distorted by the bolt load. The bolts should be tightened (preferably with a torque spanner) working at diametrically opposite nuts alternately. First turn all bolts to about half the recommended torque, then follow up to full assembly torque. Follow up the bolts about 4 hours later or 1 hour after the gasket reaches its working temperature.

### Flange finish

Concentric grooving is ideal for high pressures. "Gramophone" finish (spiral grooving) gives a continuous path for leakage and is not recommended, especially for gases. A finish equivalent to ▽▽ is usually best for flat surfaces.

### Pipe expansions / contractions

Thermal expansions of the pipeline generate forces which can crush the gasket. Contractions can reduce the gasket surface stress below the minimum required for sealing. Pipe expansions and contractions must therefore be compensated by suitable expansion devices.

## Table 1

Minimum gasket thickness for various surface stresses (at working pressure) and degrees of flange finish. For intermediate surface stress values use the next THICKER material.

1 $\mu$ = 0.001 mm = 0.04/1,000 inch

Values are rounded upwards to nearest Klinger standard sheets, hence inch and metric thicknesses do not correspond exactly.

For intermediate surface finishes (e. g. 100 $\mu$), minimum gasket thickness may be interpolated.

| Surface Stress at Working Pressure | | Milled, Turned etc. $\nabla$ (= 160 $\mu$) | | Ground, Turned etc. $\nabla\nabla$ (= 40 $\mu$) | | Ground / Turned $\nabla\nabla\nabla$ (= 16 $\mu$) | |
|---|---|---|---|---|---|---|---|
| psi | kp/cm² | inch | mm | inch | mm | inch | mm |
| 1,400 | 100 | 1/4 1/8 | 5 2 | 1/16 0.040 | 1.5 1 | 0.020 | 0.5 |
| 2,800 | 200 | 1/4 1/8 | 4 2 | 0.040 1/32 | 1 0.75 | 1/64 | 0.5 |
| 7,000 | 500 | 1/8 1/16 | 3 1.5 | 1/32 1/32 | 0.75 0.75 | 1/64 | 0.3 |
| 10,500 | 750 | – 1/16 | – 1.5 | 1/32 0.020 | 0.75 0.5 | 1/64 | 0.3 |
| 14,000 | 1,000 | – 0.040 | – 1 | 0.020 0.020 | 0.5 0.5 | 0.008 | 0.3 |

TURNED flanges generally have peaked grooves which in effect reduce the area of gasket carrying the flange load. This gives an increased surface stress and increased gasket compression. In such cases the gasket thickness may be reduced to the values printed in BLUE (provided flanges are rigid and undistorted).

## Table 2

Maximum permissible surface stress

| Thickness | | cold | | 300° C | |
|---|---|---|---|---|---|
| inch | mm | psi | kp/cm² | psi | kp/cm² |
| 0.020 | 0.5 | 28,000 | 2,000 | 20,000 | 1,400 |
| 1/32 | 0.75 | 23,000 | 1,600 | 15,000 | 1,100 |
| 0.040 | 1 | 20,000 | 1,400 | 13,000 | 950 |
| 1/16 | 1.5 | 15,500 | 1,100 | 10,500 | 750 |
| – | 2 | 13,000 | 900 | 8,500 | 600 |
| 1/8 | 3 | 8,500 | 600 | 5,500 | 400 |

## Table 3 a (liquids)

| $p_i$ psi | $p_i$ kp/cm² | $p_{d_{min}}$ psi | $p_{d_{min}}$ kp/cm² | COLD 1/8" 3 mm | COLD — 2 mm | COLD 1/16" 1,5 mm | COLD 0.04" 1 mm | COLD 1/32" 0,75 mm | COLD 0.02" 0,5 mm | HOT 1/8" 3 mm | HOT — 2 mm | HOT 1/16" 1,5 mm | HOT 0.04" 1 mm | HOT 1/32" 0,75 mm | HOT 0.02" 0,5 mm |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Minimum ratio radial width / inside dia. of gasket | | | | | | | | | | | |
| 150 | 10 | 1.100 | 75 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| 300 | 20 | 1.500 | 100 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| 575 | 40 | 2.200 | 150 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.04 | 0.02 | 0.02 | 0.01 | 0.01 | 0.01 |
| 850 | 60 | 2.900 | 200 | 0.04 | 0.02 | 0.02 | 0.01 | 0.01 | 0.01 | 0.07 | 0.04 | 0.03 | 0.02 | 0.02 | 0.01 |
| 1150 | 80 | 3.600 | 250 | 0.06 | 0.02 | 0.02 | 0.02 | 0.02 | 0.01 | 0.12 | 0.05 | 0.04 | 0.03 | 0.02 | 0.02 |
| 1450 | 100 | 4.400 | 300 | 0.08 | 0.04 | 0.03 | 0.02 | 0.02 | 0.01 | 0.20 | 0.06 | 0.05 | 0.04 | 0.03 | 0.02 |
| 1700 | 120 | 5.000 | 350 | 0.11 | 0.05 | 0.04 | 0.03 | 0.02 | 0.02 | | 0.11 | 0.06 | 0.05 | 0.04 | 0.03 |
| 2000 | 140 | 5.750 | 400 | 0.15 | 0.06 | 0.05 | 0.03 | 0.03 | 0.02 | | 0.15 | 0.09 | 0.06 | 0.05 | 0.03 |
| 2275 | 160 | 6.450 | 450 | 0.22 | 0.08 | 0.06 | 0.04 | 0.03 | 0.03 | | 0.22 | 0.12 | 0.08 | 0.06 | 0.04 |
| 2550 | 180 | 7.100 | 500 | | 0.10 | 0.07 | 0.05 | 0.04 | 0.03 | | | 0.16 | 0.09 | 0.07 | 0.05 |
| 2850 | 200 | 7.900 | 550 | | 0.13 | 0.09 | 0.06 | 0.05 | 0.04 | | | 0.21 | 0.11 | 0.09 | 0.06 |
| 3125 | 220 | 8.550 | 600 | | 0.16 | 0.10 | 0.07 | 0.05 | 0.04 | | | 0.29 | 0.14 | 0.10 | 0.07 |
| 3400 | 240 | 9.250 | 650 | | 0.20 | 0.12 | 0.08 | 0.06 | 0.05 | | | | 0.17 | 0.12 | 0.08 |
| 3700 | 260 | 10.000 | 700 | | 0.26 | 0.14 | 0.09 | 0.07 | 0.05 | | | | 0.21 | 0.15 | 0.09 |
| 4000 | 280 | 10.750 | 750 | | | 0.17 | 0.10 | 0.08 | 0.06 | | | | 0.27 | 0.18 | 0.10 |
| 4300 | 300 | 11.500 | 800 | | | 0.21 | 0.11 | 0.09 | 0.07 | | | | | 0.21 | 0.11 |
| 4600 | 320 | 12.250 | 850 | | | 0.26 | 0.13 | 0.10 | 0.07 | | | | | 0.26 | 0.13 |
| 4900 | 340 | 13.000 | 900 | | | | 0.15 | 0.11 | 0.08 | | | | | | 0.15 |
| 5100 | 360 | 13.500 | 950 | | | | 0.17 | 0.13 | 0.09 | | | | | | 0.17 |
| 5700 | 400 | 15.000 | 1050 | | | | 0.23 | 0.16 | 0.10 | | | | | | 0.23 |
| 6300 | 440 | 16.500 | 1150 | | | | | 0.21 | 0.12 | | | | | | |
| 6900 | 480 | 18.000 | 1250 | | | | | 0.27 | 0.14 | | | | | | |
| 7400 | 520 | 19.250 | 1350 | | | | | | 0.18 | | | | | | |
| 7900 | 560 | 20.500 | 1450 | | | | | | 0.21 | | | | | | |
| 8500 | 600 | 22.000 | 1550 | | | | | | 0.26 | | | | | | |

Notes: Intermediate values should be interpolated. The radial width of the gasket should never be less than twice the thickness 

# Table 3 b (gases)

| $p_l$ psi | $p_l$ kp/cm² | $p_{d_{min}}$ psi | $p_{d_{min}}$ atm | COLD ¹/₈" 3 mm | COLD — 2 mm | COLD ¹/₁₆" 1,5 mm | COLD 0.04" 1 mm | COLD ¹/₃₂" 0,75 mm | COLD 0.02" 0,5 mm | HOT ¹/₈" 3 mm | HOT — 2 mm | HOT ¹/₁₆" 1,5 mm | HOT 0.04" 1 mm | HOT ¹/₃₂" 0,75 mm | HOT 0.02" 0,5 mm |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | Minimum ratio radial width / inside dia. of gasket | | | | | | | | | | | |
| 150 | 10 | 3.600 | 240 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| 300 | 20 | 4.200 | 280 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 |
| 375 | 25 | 4.500 | 300 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.01 | 0.06 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 |
| 450 | 30 | 4.800 | 320 | 0.03 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.09 | 0.03 | 0.02 | 0.01 | 0.01 | 0.01 |
| 525 | 35 | 5.100 | 340 | 0.03 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.13 | 0.03 | 0.02 | 0.01 | 0.01 | 0.01 |
| 575 | 40 | 5.300 | 360 | 0.04 | 0.02 | 0.01 | 0.01 | 0.01 | 0.01 | 0.20 | 0.04 | 0.03 | 0.02 | 0.02 | 0.01 |
| 710 | 50 | 5.850 | 400 | 0.06 | 0.03 | 0.02 | 0.01 | 0.01 | 0.01 | | 0.06 | 0.03 | 0.02 | 0.02 | 0.01 |
| 850 | 60 | 6.400 | 440 | 0.09 | 0.03 | 0.02 | 0.02 | 0.01 | 0.01 | | 0.09 | 0.04 | 0.03 | 0.02 | 0.02 |
| 1000 | 70 | 7.000 | 480 | 0.13 | 0.04 | 0.03 | 0.02 | 0.02 | 0.01 | | 0.13 | 0.06 | 0.04 | 0.03 | 0.02 |
| 1150 | 80 | 7.600 | 520 | 0.20 | 0.05 | 0.03 | 0.02 | 0.02 | 0.02 | | 0.20 | 0.08 | 0.05 | 0.03 | 0.02 |
| 1300 | 90 | 8.200 | 560 | | 0.06 | 0.04 | 0.03 | 0.02 | 0.02 | | | 0.11 | 0.06 | 0.04 | 0.03 |
| 1450 | 100 | 8.800 | 600 | | 0.08 | 0.05 | 0.03 | 0.03 | 0.02 | | | 0.15 | 0.07 | 0.05 | 0.03 |
| 1600 | 110 | 9.400 | 640 | | 0.09 | 0.06 | 0.04 | 0.03 | 0.02 | | | 0.20 | 0.08 | 0.06 | 0.04 |
| 1700 | 120 | 9.800 | 680 | | 0.11 | 0.07 | 0.04 | 0.03 | 0.02 | | | 0.30 | 0.10 | 0.07 | 0.04 |
| 1850 | 130 | 10.400 | 720 | | 0.14 | 0.08 | 0.06 | 0.04 | 0.03 | | | | 0.13 | 0.08 | 0.05 |
| 2000 | 140 | 11.000 | 760 | | 0.21 | 0.10 | 0.06 | 0.04 | 0.03 | | | | 0.16 | 0.10 | 0.05 |
| 2150 | 150 | 11.600 | 800 | | 0.30 | 0.12 | 0.06 | 0.05 | 0.03 | | | | 0.20 | 0.12 | 0.06 |
| 2275 | 160 | 12.100 | 840 | | | 0.14 | 0.07 | 0.05 | 0.03 | | | | 0.29 | 0.14 | 0.07 |
| 2550 | 180 | 13.200 | 920 | | | 0.21 | 0.09 | 0.07 | 0.04 | | | | | 0.21 | 0.09 |
| 2850 | 200 | 14.400 | 1000 | | | | 0.11 | 0.08 | 0.05 | | | | | | 0.11 |
| 3125 | 220 | 15.500 | 1080 | | | | 0.16 | 0.10 | 0.06 | | | | | | 0.15 |
| 3400 | 240 | 16.600 | 1160 | | | | 0.21 | 0.12 | 0.07 | | | | | | 0.21 |
| 3550 | 250 | 17.200 | 1200 | | | | 0.26 | 0.14 | 0.08 | | | | | | 0.26 |
| 3700 | 260 | 17.800 | 1240 | | | | | 0.16 | 0.08 | | | | | | |
| 4000 | 280 | 19.000 | 1320 | | | | | 0.20 | 0.10 | | | | | | |
| 4300 | 300 | 20.200 | 1400 | | | | | 0.30 | 0.12 | | | | | | |
| 4600 | 320 | 21.400 | 1480 | | | | | | 0.14 | | | | | | |
| 4900 | 340 | 22.600 | 1560 | | | | | | 0.17 | | | | | | |
| 5100 | 360 | 23.400 | 1640 | | | | | | 0.21 | | | | | | |
| 5400 | 380 | 24.600 | 1720 | | | | | | 0.27 | | | | | | |

14 Notes: Intermediate values should be interpolated. The radial width of the gasket should never be less than twice the thickness

# Some famous users of KLINGER gasket materials

Klinger gasket materials are used by leading industrial organizations in over [illegible] countries. We gratefully acknowledge permission to quote the names of the internationally famous companies given in the following list.

**AUTOMOTIVE INDUSTRY**

BSA Motor Cycles Ltd., England
Daimler-Benz AG, Germany
Fiat Concord S. A. I. C., Argentina
Ford Motor Co. Ltd., England
Hanomag-Henschel Fahrzeugwerke GmbH, Germany
Industria Automotriz Santa Fé S. A., Argentina
Norton Villiers Ltd., England
Reliant Motor Co. Ltd., England
Rheinstahl-Henschel AG, Germany
Rolls-Royce Ltd., England
The Rover Company Limited, England
Steyr-Daimler-Puch A. G., Austria

**AVIATION INDUSTRY**

British Aircraft Corporation (Operating) Ltd.
Deutsche Forschungs- und Versuchsanstalt für Luft- und Raumfahrt EV, Germany (German Aero-space Research and Testing Centre)
Dirección Nacional de Fabricaciones e Investigaciones Aeronáuticas, Argentina
Rolls-Royce Ltd. (Engines) Derby, England

**CHEMICAL & ALLIED INDUSTRIES**

The Associated Octel Company Limited, England
Australian Synthetic Rubber Co. Ltd.
BASF Badische Anilin- & Soda-Fabrik AG, Germany
Boehringer Mannheim GmbH, Germany
British Celanese Ltd.
Chemische Fabrik von Heyden AG, Germany
Chemische Werke Hüls AG, Germany
CIBA S. A., Switzerland
Compañía Química S. A., Argentina
Destilerías Hiram Walker & Sons (Argentina) S. A.
Deutsche Solvay-Werke GmbH
Dista Products Ltd., England
Dow Chemical GmbH, Germany
Dunlop, Germany
Dunlop Rubber Australia Ltd.
Etablissements Kuhlmann, Belgium
Farbwerke Hoechst AG, Germany
Hibernia-Chemie AG, Germany
Hoffmann-La Roche AG, Germany
Imperial Chemical Industries Ltd., England
Kali-Chemie AG, Germany
Imperial Chemical Industries of Australia & New Zealand Ltd.
Lever Brothers Pakistan Ltd.
Lonza A. G., Switzerland
E. Merck AG, Germany
Messer-Griesheim GmbH, Germany
Monsanto Chemicals (Australia) Ltd.
Montecatini, Italy
Österreichische Stickstoffwerke A. G., Austria
Pak-American Fertilizers Ltd., Pakistan
Pak Chemicals Ltd., Pakistan
Procter & Gamble Ltd., England
Procter & Gamble GmbH, Germany
Rhodiaceta, France
Ruhrchemie AG, Germany
Scholven-Chemie AG, Germany
Solvay & Cie, Belgium