Packings for pumps
The myth of high friction!

Compression packings have tended to suffer from the image of an old fashioned technology not really suited to modern industrial processes and, in the case of rotating equipment, largely superseded by mechanical seals. In particular, it is widely believed that packings are inefficient in terms of energy consumption due to high frictional losses.
As early as 2004, the European Sealing Association (ESA) along with its US counterpart, the Fluid Sealing Association (FSA), formed a joint Task Force to develop a realistic performance-based test method for compression packings when used in rotary applications. The driving force for this project was to enable manufacturers to publish true comparative data on packing performance and thus allow end users to better differentiate between products when selecting them for their applications.

By David Edwin-Scott, Technical Director, European Sealing Association and Henri Azibert, Technical Director, Fluid Sealing Association

The methodology employed was a series of round robin tests where several laboratories would test the same product using the same test procedure and the results compared. At each iteration the procedure was refined until a point at which very close correlation of results was achieved.

The resulting "Specification for a Test Procedure for Packings for Rotary Applications" was published jointly by the ESA and FSA. The specification was taken up by CEN and has subsequently been adopted, with minor modification, as EN 16752:2015 – "Centrifugal Pumps – Test procedure for seal packings".

Whilst the final test procedure proved to produce very good correlation of results in terms of packing leakage, temperature and post-test condition of the packing, the one aspect of performance which continued to cause debate was that of frictional level and power consumption. Throughout the round robin test program the results reported for frictional torque or absorbed power showed significant variability, not least because of the variety of methods used to measure it. The CEN standard addresses this and only allows direct measurement by means of an in-line torque meter.

This uncertainty about packing friction leads to concern, as the generally accepted 'wisdom' is that packings are inefficient in terms of power consumption. This perception is based to a large extent on historical evidence stretching back several decades to traditional packings of asbestos and other natural fibres, lubricated with greases and mineral oils. Very little research has been carried out on the much more sophisticated products currently available utilising exfoliated graphite, ePTFE, aramid and other synthetic yarns and modern lubricant systems.

However, the perception of inefficiency remains, to the extent that there are fears that packings may be precluded from use in many pump systems, either through standards or regulation, by specifying that only mechanical seals are permissible.

In order to obtain definitive information on packing friction the joint Task Force commissioned the French "Technical Centre for Mechanical Industries – CETIM" to carry out a follow-up project. This consisted of the design and manufacture of a dedicated test rig to carry out testing in accordance with EN 16752, including highly accurate systems to directly measure the frictional force of the packing alone.

The test rig is designed to be capable of testing both compression packings and mechanical seals so that direct comparison can be made under the same conditions. The shaft diameter is

Figure 1 *– Friction test rig.*

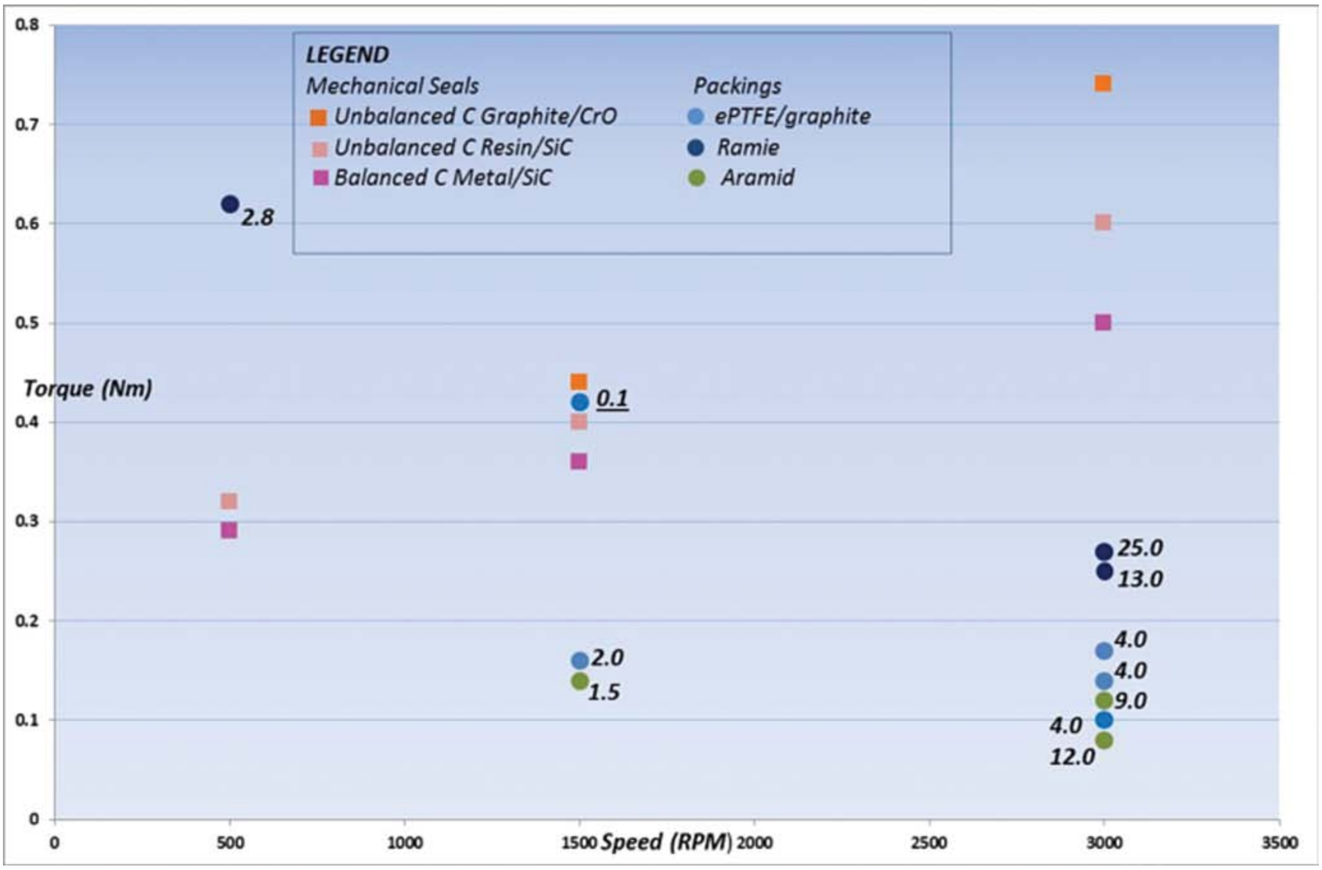

Figure 2 *– Initial test results.*

50 mm and the packing cross-section 10mm. All tests were carried out using water as the test medium.

After initial trials to validate the equipment functionality and accuracy of the monitoring devices, the first tests were carried out on 3 commonly used packing types and compared with several mechanical seal types.

Testing was carried out at 6 bar pressure and a variety of rotational speeds with varying target leak rates. The results of these tests are shown graphically in Figure 2.

The measured torque is plotted against speed, in the case of the packing with the associated shaft leak-rate in millilitres per minute shown.

These results were extremely unexpected. Not only were the figures for packing much lower than predicted, they were certainly of the same order of magnitude as, and generally lower, than the mechanical seal. Of course, a degree of leakage must be tolerated when using packings and the lubrication afforded by the leaking fluid will reduce the friction. But even when the leak rate is extremely low, as in the case of the ePTFE/graphite packing at 1500 rpm, the friction recorded was at the same level as the mechanical seals.

These unexpected results have led to a reconsideration of the traditional methods for calculating packing friction that have been widely accepted in the past.

Theoretical Considerations

The formula which has long been used to calculate power consumption from compression packing systems is

$$P = Pp \times RPM \times D \times \mu \times Ap \times F$$

where:

- **P:** Power (Horsepower or KW depending on units used)
- **Pp:** is the sealed pressure
- **RPM:** rotational speed
- **D:** Shaft diameter
- **μ:** Coefficient of friction between the packing and the shaft
- **Ap:** Packing contact area
- **F:** Factor depending on units used

Whilst recognized to be approximate, this formula has been generally adopted and used in the industry including tools such as the 'Life Cycle Cost Calculator' that is found on the FSA and ESA websites. The formula is similar to the one used for mechanical seals which has been shown to give a good approximation to power consumption levels.

Acknowledged approximations in the packing formula are that it does not take account of lubricant levels, actual packing compression, type of liquid sealed and viscosity and temperature. However, it has been used to provide a figure for the amount of energy consumed by

Leakage Class ml/min	Average Leakage ml/min	Packing Type	Torque Value Nm			
			Actual vs. Calculated	1500 RPM 6 bar	1500 RPM 10 bar	3000 RPM 6 bar
0 - 5	2,5	ePTFE/graphite	Actual	0.17	0,67*	0,15
			Calculated	0,11	0,19	0,23
5 - 15	10	Aramid	Actual	0,17	0,15	0,10
			Calculated	0,06	0,09	0,11
15 - 30	22,5	Ramie	Actual	0,34	0,47	0,25
			Calculated	0,04	0,07	0,08
0 - 5	2,5	Carbon/graphite	Actual	0,91	0,60	1,19
			Calculated	0,19	0,31	0,38

≤2:1 ≥2:1

Figure 3 *– Actual v calculated (Initial).*

the packing. It tends to give power consumption levels that are approximately 10 greater than that of a balanced mechanical seal used under the same conditions.

It is clear from the test results that the approximations in the formula are not sufficient to explain the deviations from the calculated values. The differences in calculated results from the test measurements reported here vary by factors from 25 to a massive 200 times.

Revising the formula

The formula needs to be adjusted to give results more in line with actual observations. The main areas to be addressed are:

- A pressure drop coefficient to recognize that the sealed pressure does not apply over the entire contact area of the packing
- The speed must be taken into account. From a typical Stribeck curve the highest torque is at low speeds (200 RPM), lowest torque around 750 RPM to 1500 RPM and increases at 3000 RPM
- Leakage – high leakage lowers torque due to lubricating effects

A further series of tests was initiated to complete a matrix of conditions, including a higher pressure and an additional packing type manufactured from carbon/graphite yarns.

By taking these factors into account and selecting constants to be included in the formula for pressure drop factor, speed, coefficients of friction for the various materials and applying the actual leakage rate, the test

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			Calculated	0,11	0,18	0,14
15 - 30	15	Ramie	Actual	0,34	0,47	0,25
			Calculated	0,18	0,29	0,24
5 - 15	10	Carbon/graphite	Actual	0,60	0,65	0,5
			Calculated	0,35	0,69	0,47

≤1.5:1 ≤2:1

Figure 4 *– Actual v calculated (Final).*

results were compared with the newly calculated figures as shown in Figure 3.

The calculated values are much closer to the experimental results and within the same order of magnitude, but there are still discrepancies. Particularly the Ramie result, which under-estimates by a factor of three and unexpectedly higher figures for the carbon/graphite packing.

A further carbon/graphite test was carried out with an amended installation technique which created additional pre-compression on each ring, resulting in improved performance. Further adjustment to the speed factors applied and closer attention to coefficient of friction result in an even better correlation of calculated and actual torque, as shown in the comparison table, Figure 4.

Therefore, the finalized formula takes the form:

Torque = area of contact x process pressure x coefficient of friction x moment arm radius x pressure drop factor x speed factor x size factor/ leakage factor

This expression, combined with the figures evolved for the various factors and for a range of packing types and their known leakage characteristics, will be incorporated in the 'Life Cycle Cost Calculator', available on both the ESA and FSA websites.

About the Authors

David Edwin-Scott graduated with a BSc (Hons) in Chemical Engineering from Queen's University, Belfast in 1973. He has gained extensive experience in the sealing industry throughout the years and currently serves as a consultant to the ESA as a Technical Director, in the fields of compression packings and elastomeric sealing technology.

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