



Labyrinth seal flow

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LABYRINTH SEAL FLOW

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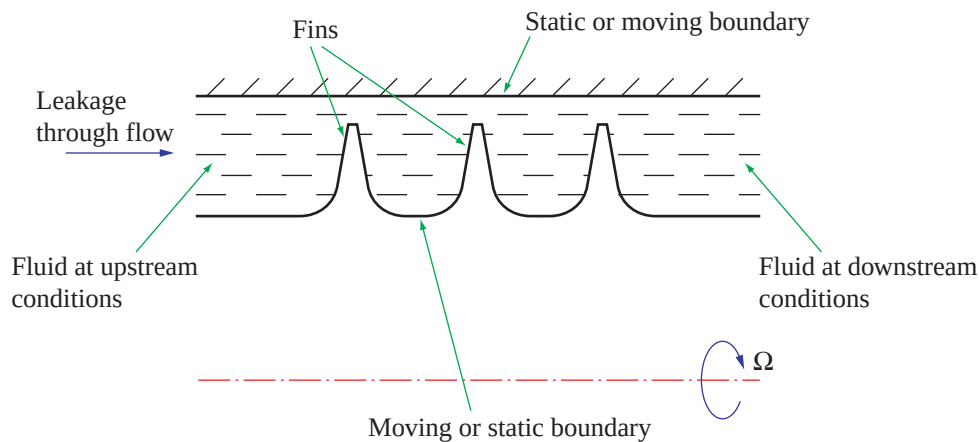
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LABYRINTH SEAL FLOW

1. PURPOSE AND SCOPE OF THIS DATA ITEM

Labyrinth seals are widely used in industry to limit the flow of fluid between components. They are commonly used in rotating machines such as pumps, compressors, turbines and food processing machinery, where they are typically selected in applications with high relative speed between rotating components.

In its simplest form, a labyrinth seal consists of a series of radial fins forming a restriction to an annular flow of fluid as shown in Sketch 1.1.



Sketch 1.1 Generic labyrinth seal configuration

The aims of this Data Item are to:

- introduce a variety of seal configurations relevant to rotating machinery;
- assist engineers in the selection of a labyrinth seal adequate for their specific rotating flow application;
- provide methods and recommendations for the flow performance assessment and design of straight-through and stepped labyrinth seals by quantifying the leakage flow rates; and
- introduce methods for quantifying the leakage flow rates in staggered and radial labyrinth seals.

2. NOTATION AND UNITS

		<i>SI Unit</i>
A	annular cross-sectional flow area of the seal, Sketch 3.1	m^2
C_d	discharge coefficient, defined in Section 4.2.1	–
$C_{d,r}$	discharge coefficient for a labyrinth fin with rounded tip edge, in Figure 8	–
$C_{d,se}$	discharge coefficient for a labyrinth fin with a sharp tip edge, in Figure 8	–
c	seal clearance, defined in Sketch 3.6	m
D	diameter of outer fin, defined as $D = 2r_i$	m
D_h	annular hydraulic diameter, defined as $D_h = 2(r_o - r_i)$, Equation (4.16)	m
e	groove depth, defined in Sketch 4.3	m
F	function	–
g	acceleration due to gravity	m^2/s
H	step height defined in Sketch 5.1	m
h	fin height, defined in Sketch 3.6	m
k	carry-over factor defined in Sections 4.2.1 and 4.2.4	–
k_1	Hodkinson's carry-over factor defined in Equation (4.10)	–
k_2	Zimmermann's carry-over factor defined in Equation (4.11)	–
k_i	carry-over factor defined in Equation (4.12)	–
k_s	correction factor for C_d , defined in Equation (5.1)	–
L	total length of seal	m
L_{cell}	honeycomb cell dimension, defined in Sketch 4.4	m
l_{pitch}	pitch, defined in Sketch 3.6	m
m	mass flow rate across the seal clearance area A , defined as $m = \rho u_{mean} A$	kg/s
m_{actual}	actual mass flow rate, in Section 4.2.1	kg/s

m_{ideal}	ideal mass flow rate, in Section 4.2.1	kg/s
Ma	Mach number	–
n	number of labyrinth fins	–
p	static pressure	Pa
p_t	total pressure	Pa
p_{t0}	upstream total pressure	Pa
p_n	downstream static pressure	Pa
Q	mass flow function, defined in Table 3.1 and Equation (4.2)	kg K ^{0.5} / (N s)
Q_{smooth}	mass flow function without honeycomb surface, Equation (4.17)	kg K ^{0.5} / (N s)
$Q_{honeycomb}$	mass flow function with honeycomb surface, Equation (4.17)	kg K ^{0.5} / (N s)
R	characteristic gas constant	J/(kg K)
Re	axial Reynolds number, defined in Table 3.1	–
Re_{D_h}	Reynolds number based on hydraulic diameter, defined in Equation (4.15)	–
Re_ϕ	rotational Reynolds number, defined in Table 3.1	–
r	radial coordinate or local radius	m
r_{fillet}	fillet radius, defined in Sketch 3.6	m
r_{round}	rounding radius, defined in Sketch 4.5	m
r_i	radius of labyrinth fin tip, defined in Sketch 3.6	m
r_o	inner radius of outer cylinder, defined in Sketch 3.6	m
S_i	factor defined in Equation (4.14)	–
T	fluid static temperature	K
T_t	fluid total temperature	K
T_{t0}	upstream fluid total temperature	K
Ta	Taylor number, defined in Table 3.1	–
t	fin tip land width, defined in Sketch 3.6	m

t_G	groove width, defined in Sketch 4.3	m
u_{mean}	mean throughflow leakage velocity over the seal clearance	m/s
u_ϕ, r_i	fin tip angular velocity component or fin tip speed	m/s
x	axial coordinate	m
x_0	steam quality	—
Δm	mass flow change induced by honeycomb, defined in Equation (4.17)	kg/s
Δp	static pressure difference	Pa
α	flow coefficient in Table 4.2	—
ε	carry-over correction factor in Table 4.2	—
ε_1	carry-over correction factor in Table 4.2	—
λ	friction factor, defined as $\lambda = \Delta p / [(L/D)0.5\rho u_{mean}^2]$	—
λ_c	critical pressure ratio, defined in Equation (4.7)	—
v	fluid specific volume	m ³ /kg
v_0	upstream fluid specific volume	m ³ /kg
γ	isentropic index, also known as specific heat ratio	—
ϕ_{fin}	fin included angle, defined in Sketch 3.6	degree
μ	fluid dynamic viscosity	Pa s
Ω	shaft angular velocity	rad/s
ρ	fluid density	kg/m ³
ρ_0	upstream fluid density	kg/m ³
ζ_m	compressibility function	—

Subscripts

0	refers to flow conditions upstream of a labyrinth seal
i	refers to flow conditions in the i -th chamber or i -th fin
n	refers to flow conditions in the n -th chamber or n -th fin or downstream of a labyrinth seal

3. INTRODUCTION

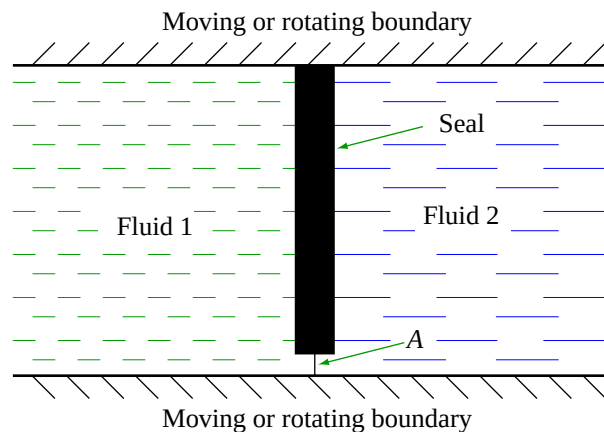
A general introduction to sealing for rotating machinery applications is presented in Section 3.1, where the function of seals and their principles of operation are provided. An overview of the selection of an appropriate seal for a particular function is presented in Section 3.2. The operating and design principles of labyrinth seals are introduced in Section 3.3, where performance and design parameters are also discussed.

3.1 Introduction to Sealing

Seals are an important aspect of rotating machinery design where pressurised fluids must be contained within an area of a machine, such as a hydraulic cylinder, contaminants must be excluded or lubricants retained. In more general flow applications, the function of seals is to prevent or limit the flow of fluids or particulates between components. There is a wide range of seals available and they can be classified into two general categories:

- (a) **static seals**, where sealing takes place between two surfaces that are not in relative motion to each other; and
- (b) **dynamic seals**, where sealing takes place between two surfaces that are in relative motion to each other, for example, produced by rotating or reciprocating motion.

The subject of sealing for fluid applications is introduced by Zuk²⁶, Neale⁵⁰, Steinetz and Hendricks⁵⁶, Muller and Nau⁵⁷, SAE⁵⁸ and Flitney⁷⁶.



Sketch 3.1 Basic sealing configuration

The basic sealing problem is illustrated in Sketch 3.1, where either boundary shown may be stationary or moving, and the fluid in each region may be of the same or of a dissimilar substance. Any clearance between two flow components permits the passage of fluid molecules in either direction, the direction depending on the relative static pressure and momentum associated with the fluid regions. Fluid can move between different regions of space by means of diffusion, free convection or forced convection. The diameter of a typical gas or vapour molecule is of the order of 10^{-9} m. Molecules can, therefore, diffuse through very small gaps such as pores in a machine casing or a seal component. Convection involves the mixing of one portion of fluid with another by means of gross movements of the mass of fluid. If the fluid motion is caused by external mechanical means, such as a fan or pump, the process is called forced convection. If the fluid motion is caused by local density differences, due to, for instance, temperature differences in buoyancy-driven flows, the process is called natural or free convection.

Designing to limit the leakage flow in a seal typically involves the direct opposite of designing conventional flow in pipes, ducts and nozzles, where it is usually desirable to have a system with efficient, low pressure-loss flow processes²⁶. In seals, it is normally beneficial to have an inefficient flow process because this results in lower flow rates and therefore less leakage. A trade-off of seal complexity with associated power loss for the overall machine system must also be considered.

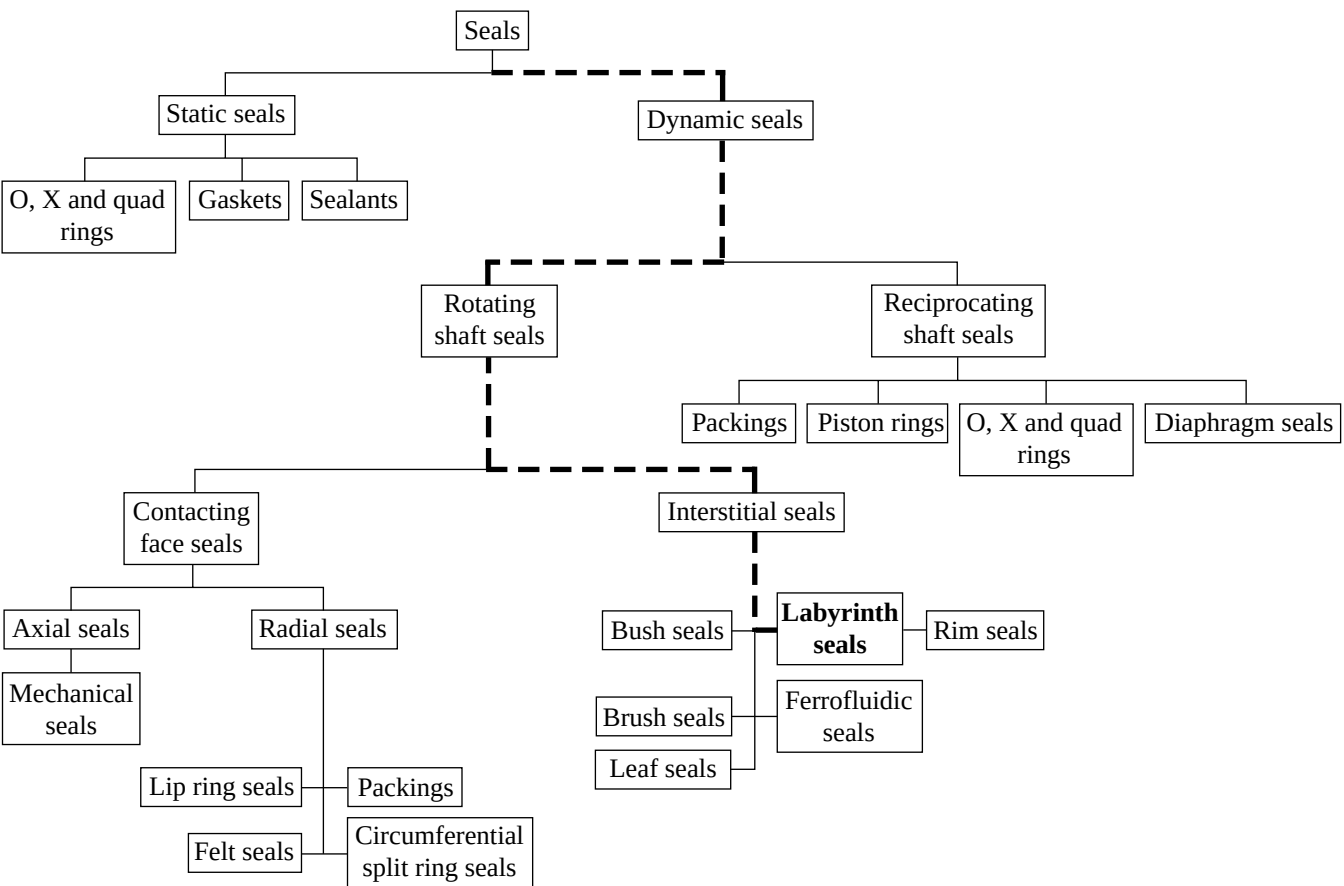
3.2 Seal Selection Overview

In the selection of seal type, the following considerations are typically taken into account:

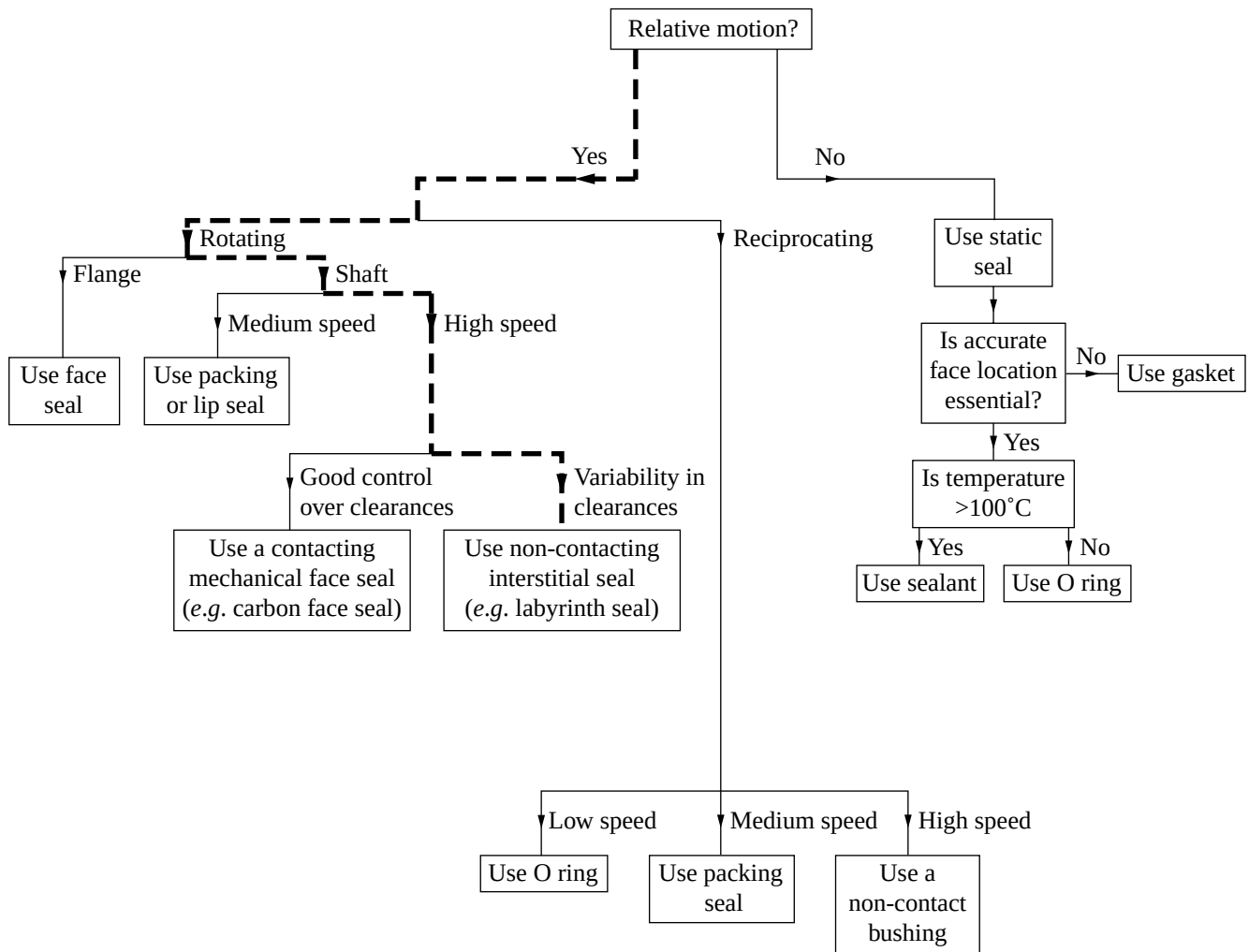
- (a) the nature of the fluid to be contained or excluded, whether liquid, gaseous, two-phase, multi-phase, multi-component flow or particulate laden;
- (b) the static pressure levels on each side of the seal;
- (c) the nature of any relative motion between the seal and mating components, *e.g.* rotating, reciprocating, contacting or non-contacting, high speed or low speed;
- (d) the level of sealing required, for example, whether some leakage can be tolerated or can a contacting seal be used;
- (e) the operating temperatures;
- (f) the life expectancy;
- (g) the need for serviceability; and
- (h) the total lifetime cost, considering components and lifetime service and replacement costs.

The variety of seals and sealing systems available is extensive although can be broadly classified as static or dynamic seals. A general classification diagram for the principal types of seals is shown in Sketch 3.2.

In selecting a seal, the principal criterion is the level of mass of flow leakage that can be tolerated. Practical considerations of geometric form, such as run-out need to be taken into account. If a low leakage is required and close control of geometric form is possible, then a contacting mechanical face seal, for example, may be possible. If geometric conditions are less well defined with variability in, for example, radial and axial location, then an interstitial seal may be the only practical solution. A general guide to the selection of seal type is provided in Sketch 3.3.



Sketch 3.2 Seal taxonomy



**Sketch 3.3 Guide to the selection of a particular type of seal (adapted from Hamilton⁴⁸).
Definition for low, medium and high speeds depend on the specific application**

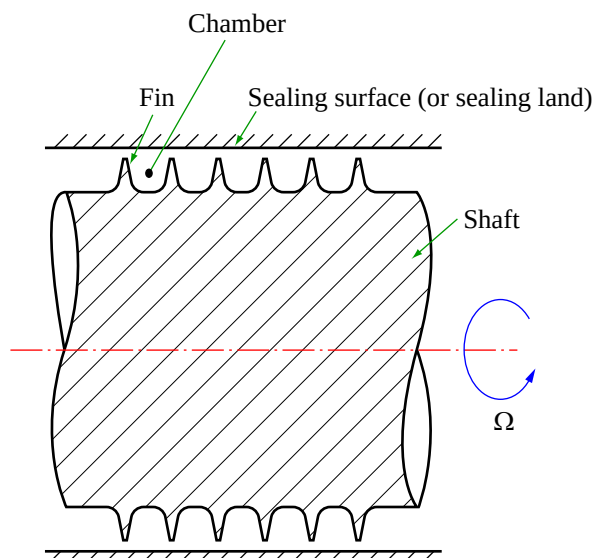
Labyrinth seals are dynamic seals for rotating shafts and belong to a seal category known as interstitial seals. The term interstitial, or non-contacting, seal is used for seals that allow unrestricted relative motion between the stationary and moving components (*i.e.* no seal to shaft contact). Types of interstitial seals include labyrinth, brush and bush seals. The merits of labyrinth seals are primarily their simplicity and reliability in providing effective leakage resistance over long periods of time, at elevated temperatures and in rotating applications. In general, however, brush seals offer improved sealing characteristics in comparison to labyrinth seals^{68,72}. Measurements of the flow coefficient in brush seals have shown up to 24% reduction in comparison to the equivalent labyrinth seals⁶⁸. Brush seals, however, cause higher power consumption due to the friction contact between the brushes and the rotor. In addition, it is possible for fluid to blow through part of the brush packing if the stiffness of a region of the brush seal, at a location around the circumference, is not sufficient. Bush seals are geometrically simpler than labyrinth and brush seals but, typically, permit greater leakage rates.

The selection of seals for rotating and reciprocating applications is considered in ESDU 80012³³ and ESDU 83031³⁸ respectively. Labyrinth seals, the focus of this item, are typically selected in applications where there is high relative speed between rotating components and there is a requirement to restrict the leakage flow. Labyrinth seals can be used for axial applications where the axis of the fins is coincident with the axis of the shaft, and in radial sealing applications, where the fins are typically perpendicular to the shaft axis.

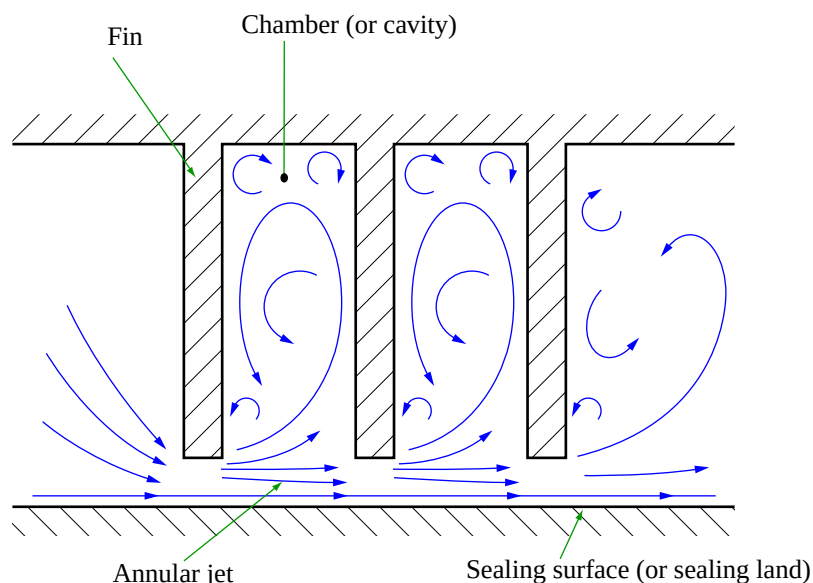
3.3 Labyrinth Seal Operating and Design Principles

It is believed that the use of a tortuous path, between high and low static pressure regions, incorporating a series of non-contacting restrictions and chambers in the form of a labyrinth seal, was first developed and implemented by Parsons for his steam turbine concept in 1892^{1,8}. Labyrinth seals therefore have a long history of proven reliability in a wide range of applications, including gas turbine engines⁷², with robust operation and developed technology. Labyrinth seals have been reviewed by Sneek²³, are the subject of the book by Trutnovsky and Komotori³⁶, and have been the focus of many research studies. In the older literature, labyrinth seals may also be referred to as “labyrinth packings” or “labyrinth glands”.

In its simplest form, a labyrinth seal consists of a series of fins and corresponding chambers or cavities forming a restriction to the leakage flow and a volume for expansion, respectively, as illustrated in Sketch 3.4. Labyrinth seals work by throttling the flow through successive openings in series. In each throttle, static pressure difference accelerates the flow to form an annular jet, and some of the kinetic energy associated with the jet flow is dissipated by turbulence induced by the intense shear stress, recirculation and eddy motions in the next chamber as illustrated in Sketch 3.5.

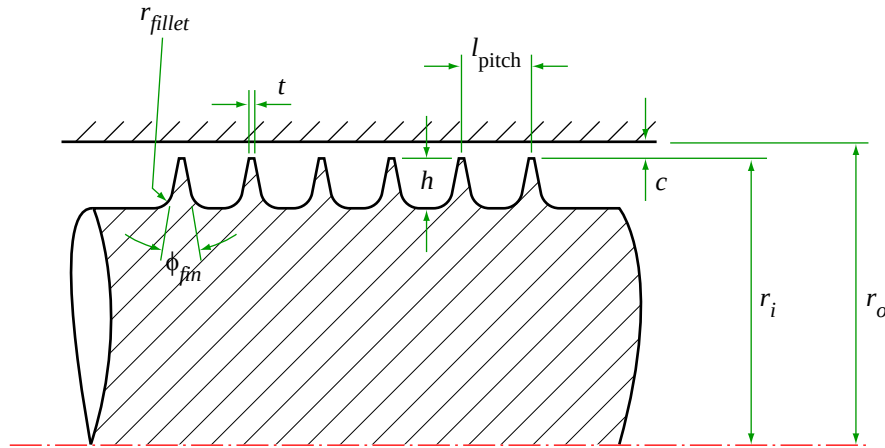


Sketch 3.4 Typical axial straight-through labyrinth seal

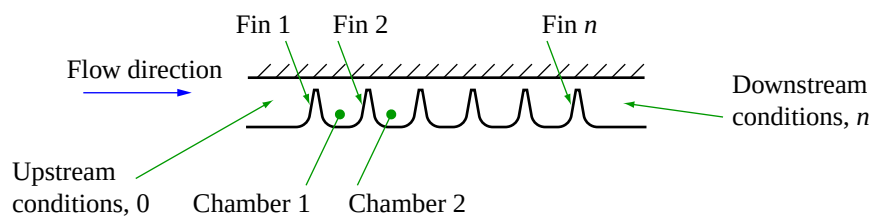


Sketch 3.5 Characteristic flow through a labyrinth seal (after Tipton *et al.*⁴¹)

The principal geometric parameters of a typical axial straight-through labyrinth seal are defined in Sketches 3.6 and 3.7. It should be noted that the terms fin, knife, tooth and blade are used interchangeably in industry and in the technical literature. In this Data Item, the term fin is principally used.



Sketch 3.6 Principal labyrinth seal geometric parameters

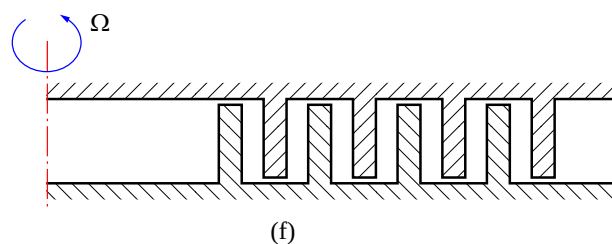
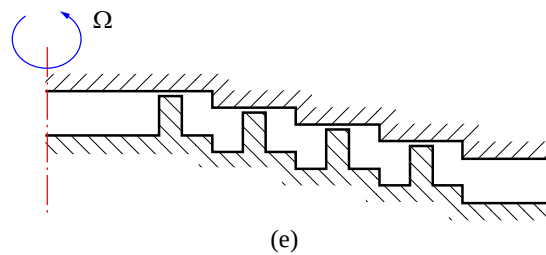
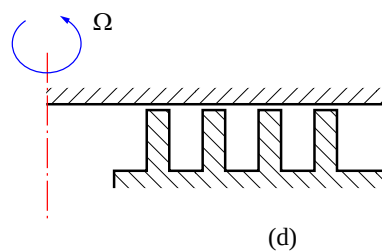
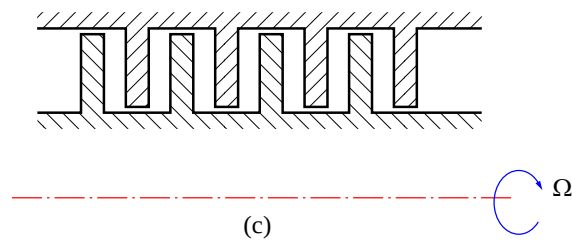
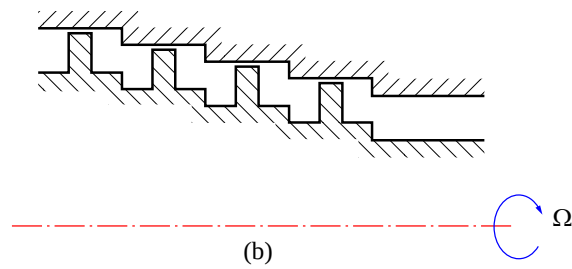
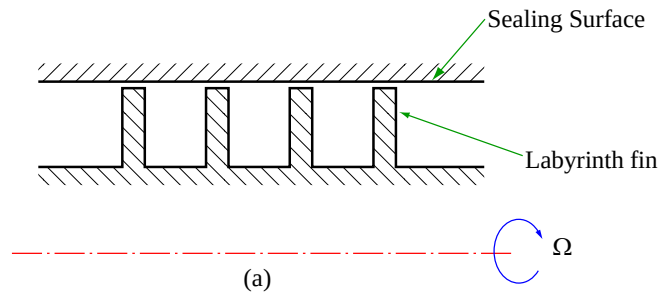


Sketch 3.7 Labyrinth fin and chamber notation

The most common forms of labyrinth seal include:

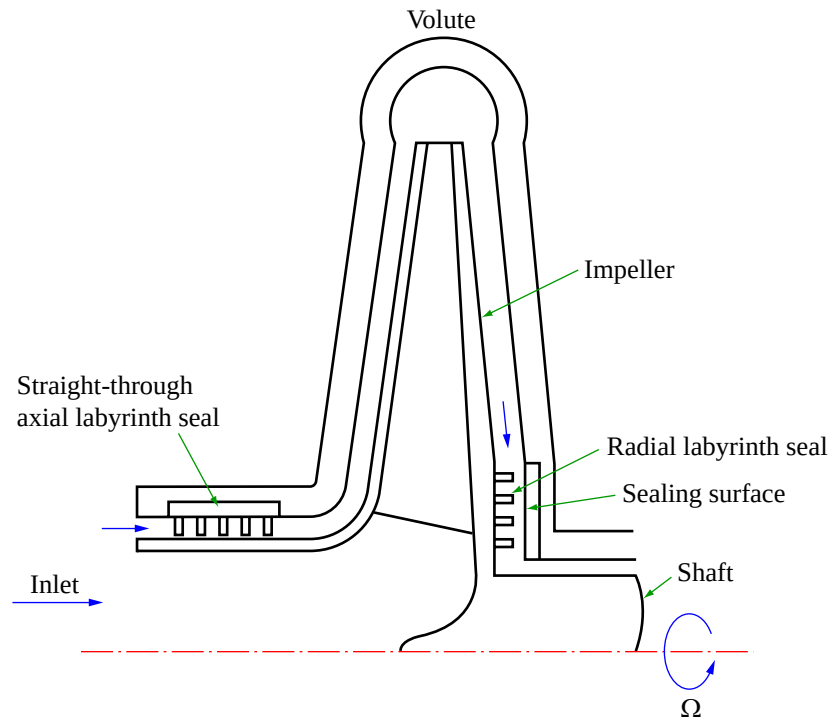
- straight-through;
- stepped; and
- staggered.

Each of these forms of labyrinth seal can be used in axial rotating flow applications, where the axis of the fins is coincident with the axis of the shaft, or in radial sealing applications, where the fins are perpendicular to the shaft axis. Schematic diagrams for each of these are illustrated in Sketch 3.8. Fins may rotate, move in a reciprocating motion or be stationary. In general, for the case of an axial application, it is desirable to locate the fins on the inner component as this improves the stability of the system and heat pick-up due, for example, to windage.



Sketch 3.8 Schematic diagrams illustrating a selection of labyrinth seals.
Axial applications: (a) straight-through (b) stepped (c) staggered.
Radial applications: (d) straight-through (e) stepped (f) staggered.

It should be noted that labyrinth seals do not provide a complete seal. If there is a static pressure difference across the seal, then there will be a net flow through the seal. A typical application for a labyrinth seal for limiting the leakage of flow in a centrifugal pump is illustrated in Sketch 3.9. In this application a straight-through labyrinth seal is used for limiting the flow leakage on the inlet side of the impeller shaft and a radial labyrinth is used on the rear face of the impeller.



Sketch 3.9 Typical application of labyrinth seals showing an axial straight-through seal at the pump inlet and a radial straight-through labyrinth seal on the pump rear face

3.3.1 Performance and design parameters

The design of a straight-through labyrinth seal generally involves a compromise between the number of fins and a pitch that is large enough to ensure that the kinetic energy of the flow is dissipated in the cavity (chamber). However, there are a significant number of performance and design parameters to consider, including (see Sketch 3.6):

- clearance, c ;
- pitch, l_{pitch} ;
- the number of fins, n ;
- fin tip land width, t ;
- rotational speed, Ω ;
- cavity aspect ratio;
- fin included angle, ϕ_{fin} ;
- sealing land surface roughness and porosity;
- fin height, h ;
- eccentricity;
- pressure ratio p_{t0}/p_n ; and
- axial Reynolds number, Re .

For a stepped seal, additional parameters (see Sketches 3.8(b) and (e) and 5.1) include:

- step height, H ;
- step configuration;
- distance from seal fin to step face; and
- flow direction.

The principal fluid-dynamic parameters that determine labyrinth seal performance are listed in Table 3.1. The corresponding geometric parameters are given in Table 3.2. In terms of modelling, the parameters identified do not operate independently and, therefore, a significantly large matrix of parameter combinations affects labyrinth seal performance.

TABLE 3.1 Principal fluid-dynamic parameters for labyrinth seals (after Tipton *et al.*⁴¹)

Parameter	Definition	Characteristic Effect
Mass flow rate function	$Q = \frac{m\sqrt{T_{t0}}}{Ap_{t0}}$	Dependent variable
Pressure ratio	p_{t0}/p_n	Strong
Axial Reynolds number	$Re = \frac{(m/A)c}{\mu}$	Moderate
Fin tip speed	u_{ϕ, r_i}	Moderate
Rotational Reynolds number	$Re_{\phi} = \frac{\rho\Omega rc}{\mu}$	Weak
Taylor number	$Ta = \frac{\rho c u_{\phi, r_i}}{\mu} \sqrt{\frac{c}{r_i}}$	Weak

TABLE 3.2 Principal parameters for labyrinth seals
(adapted from Tipton *et al.*⁴¹)

Parameter	Associated parameter	Characteristic Influence on Relevant Seals			
		Straight-through Seals	Stepped Seals	Staggered Seals	Radial Seals
Number of fins, n	Number of throttles	Strong	Strong	Strong	Strong
Fin included angle, ϕ_{fin}	Orifice geometry	Moderate	Moderate	Moderate	Moderate
Fin tip land width, t	Throat length	Moderate	Moderate	Moderate	Moderate
Clearance, c	Fin sharpness	Moderate	Moderate	Moderate	Moderate
Fin shape	Orifice geometry	Weak to moderate	Weak	Weak	Weak to moderate
Pitch, l_{pitch}	Throttle spacing	Moderate	Weak	Moderate	Moderate
Fin height, h	Chamber aspect ratio	Weak	Weak	Weak	Weak
Land surface roughness	Land relative roughness	Moderate	Weak	Moderate	Moderate
Land surface porosity	Land porosity	Moderate	Weak to moderate	Moderate	Moderate
Step height	Step height	—	Weak	—	—
Distance to contact	Rotor axial location	—	Weak to moderate	—	—
Flow direction	Flow down stator step/ Flow up stator step	—	Weak	—	—

A particular challenge in determining labyrinth seal performance parameters arises with in-service and worn seals which can involve variations due to:

- erosion of coatings on either the fin land or on the sealing surface;
- grooves in abradable coatings caused by wear^{*}
- uneven fin wear;
- fin rounding;
- manufacturing errors;
- thermal distortions; and
- axial movements.

Care must be taken in the design of labyrinth seals. If the cavity volume following the radial fin is too small the flow will pass straight-through without expanding and thus without the desired pressure drop. Despite careful labyrinth design and specification of a labyrinth seal parameters, some flow velocity will inevitably be carried straight-through from one fin to another. In order to reduce this kinetic-energy “carry-over” effect, a stepped geometry can be incorporated into a labyrinth seal design and a wide range of labyrinth variants are possible.

^{*} Note that grooves may be deliberately formed as an integral part of a labyrinth seal design.