**SLOVENSKI STANDARD**
# SIST EN ISO 21922:2022/oprA1:2023

**01-april-2023**

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**Hladilni sistemi in toplotne črpalke - Ventili - Zahteve, preskušanje in označevanje - Dopolnilo A1 (ISO 21922:2021/DAM 1:2023)**

Refrigerating systems and heat pumps - Valves - Requirements, testing and marking - Amendment 1 (ISO 21922:2021/DAM 1:2023)



Kälteanlagen und Wärmepumpen - Ventile - Anforderungen, Prüfung und Kennzeichnung - Änderung 1 (ISO 21922:2021/DAM 1:2023)

Systèmes de réfrigération et pompes à chaleur - Robinetterie - Exigences, essais et marquage - Amendement 1 (ISO 21922:2021/DAM 1:2023)



**Ta slovenski standard je istoveten z: EN ISO 21922:2021/prA1**

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**ICS:**

| | | |
|---|---|---|
| 23.060.20 | Zapirni ventili (kroglasti in pipe) | Ball and plug valves |
| 27.080 | Toplotne črpalke | Heat pumps |
| 27.200 | Hladilna tehnologija | Refrigerating technology |

**SIST EN ISO 21922:2022/oprA1:2023 en,fr,de**

DRAFT AMENDMENT
# ISO 21922:2021/DAM 1

ISO/TC **86**/SC **1** Secretariat: **ANSI**

Voting begins on: **2023-02-20**

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# Refrigerating systems and heat pumps — Valves — Requirements, testing and marking

## AMENDMENT 1

*Systèmes de réfrigération et pompes à chaleur — Robinetterie — Exigences, essais et marquage*

*AMENDEMENT 1*

ICS: 27.200; 27.080



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**ISO/CEN PARALLEL PROCESSING**



Reference number
ISO 21922:2021/DAM 1:2023(E)

# Refrigerating systems and heat pumps — Valves — Requirements, testing and marking

## AMENDMENT 1

*In Table 1 replace variable description of $K_{VS}$, $L$, and $Q_M$ with:*

**Table 1 — List of symbols**

| | | |
|---|---|---|
| $K_{VS}$ | Flow coefficient of the valve | $m^3/h$ |
| $L$ | Leakage percentage | % |
| $Q_M$ | Valve leakage mass flow rate measured with air | kg/h |

*In Table 1 replace $Q_V$ with $Q_{V1}$ and insert a new variable description of $Q_{V2}$:*

**Table 2 — List of symbols**

| | | |
|---|---|---|
| $Q_{V1}$ | Valve leakage volume flow rate measured upstream with air | $m^3/h$ |
| $Q_{V2}$ | Valve leakage volume flow rate measured downstream with air | $m^3/h$ |



*Replace 7.6 and Table 3 with:*

### 7.6 Seat tightness



#### 7.6.1 General

This clause applies to components where internal seat tightness is a design feature. The seat tightness shall be classified according to Table 3.

The maximum leakage percentage, $L$, is calculated as described in 7.6.2. The maximum values of $L$ for a given seat tightness class are listed in Table 3.

**Table 3 — Type test requirements for seat tightness**

| Seat tightness class | Maximum leakage percentage, $L$[a] | Maximum leakage volume flow, $Q_V$, measured downstream[a] |
|---|---|---|
| A | — | Zero bubbles or equivalent measured during one minute[b] |
| B | — | Zero bubbles or equivalent measured during one minute |
| C | 0,002 % | —[c] |
| D | 0,01 % | —[c] |
| E | 0,025 % | —[c] |
| F | 0,05 % | —[c] |

[a] For type test the manufacturer shall measure the leakage at ambient temperature covering the whole differential pressure range. For manual valves, see Table 4 for suggested upper limits to the maximum differential pressure.

[b] For safety valves the manufacturer shall measure the leakage up to 0,9 × set pressure of the valve.

[c] The maximum downstream leakage volume flow corresponding to the maximum leakage percentage can be calculated using equation (3) in clause 7.6.2.

[d] For seat tightness class H, testing shall be conducted to verify the seat tightness specified in the technical literature.

 

**Table 3** *(continued)*

| Seat tightness class | Maximum leakage percentage, $L$[a] | Maximum leakage volume flow, $Q_V$, measured downstream[a] |
|---|---|---|
| G | 0,1 % | —[c] |
| H | — | —[d] |

[a] For type test the manufacturer shall measure the leakage at ambient temperature covering the whole differential pressure range. For manual valves, see Table 4 for suggested upper limits to the maximum differential pressure.

[b] For safety valves the manufacturer shall measure the leakage up to 0,9 × set pressure of the valve.

[c] The maximum downstream leakage volume flow corresponding to the maximum leakage percentage can be calculated using equation (3) in clause 7.6.2.

[d] For seat tightness class H, testing shall be conducted to verify the seat tightness specified in the technical literature.

The required seat tightness class depends on the intended application of the valve:

a) Valves leading to the atmosphere permanently shall be seat tightness class A.

b) Valves leading to the atmosphere during service shall be seat tightness class A or B.

c) For other valves, seat tightness classes with lower requirements are allowed.

NOTE 1 Components with several valve seats, may have several seat tightness classes.

NOTE 2 Safety valves are examples of valves where seat tightness class A is required, while most stop valves will require seat tightness classes A or B.

For manually closed valves, when testing the seat tightness, the seat shall be closed before the test applying the prescribed closing force.

For valves of the double seating type such as many gate, plug, and ball valves, the test pressure shall be applied successively to each end of the closed valve and tightness to the opposite end checked.

As alternate methods for valves with independent double seating (such as double disc or split wedge gate valves), at the option of the manufacturer, the pressure may be applied inside the bonnet (or body) of the closed valve and each seat checked for tightness at the valve ports, or the pressure may be applied to the valve ports and the sum of seat leakage measured at the bonnet (or body). These alternate methods may be used at the option of the manufacturer for valves with single discs (such as solid or flexible wedge gate valves) provided a supplementary closure member test across the disc is performed.

For other valve types, the test pressure shall be applied across the closure member in the direction producing the most adverse seating condition. For example, a globe valve shall be tested with pressure under the disc. A check valve, or other valve type designed, sold, and marked as a one-way valve, requires a closure test only in the appropriate direction. A stop check valve requires both tests.

### 7.6.2 Seat tightness: type test

The leakage percentage $L$ is specified for the flow directions for which the valve is designed to shut off the flow.

The manufacturer shall measure the leakage covering the whole differential pressure range for which the valve is designed using gas (for instance air or nitrogen). The leakage percentage $L$ shall not exceed the limits given in Table 3 for type test.

For seat tightness class H the leakage percentage, $L$, shall be specified in the technical literature.

Two-directional valves shall be measured in both directions.

The greatest value measured is used for calculating the leakage percentage, $L$, by means of equation (1), (2) or (3).

 

When measuring the leakage rate of the seat, the seat shall be closed before the test applying the prescribed closing force.

The leakage percentage, $L$, is determined using one of the following equations:

$$L = 100\,\% \times \frac{Q_{\mathrm{M}}}{31{,}6 \cdot K_{\mathrm{VS}} \cdot \sqrt{\Delta p \cdot \rho_1}} \tag{1}$$

or

$$L = 100\,\% \times \frac{Q_{\mathrm{V1}}}{31{,}6 \cdot K_{\mathrm{VS}} \cdot \sqrt{\dfrac{\Delta p}{\rho_1}}} \tag{2}$$

or

$$L = 100\,\% \times \frac{Q_{\mathrm{V2}} \cdot \rho_2}{31{,}6 \cdot K_{\mathrm{VS}} \cdot \sqrt{\Delta p \cdot \rho_1}} \tag{3}$$

where

$Q_{\mathrm{M}}$ is the valve leakage mass flow rate measured with air in kilograms per hour;

$Q_{\mathrm{V1}}$ is the valve leakage volume flow rate measured upstream with air in cubic metre per hour;

$Q_{\mathrm{V2}}$ is the valve leakage volume flow rate measured downstream with air in cubic metre per hour;

$K_{\mathrm{VS}}$ is the flow coefficient of the valve in cubic metre per hour;

$\Delta p$ is the test pressure difference across the valve in bar;

$\rho_1$ is the upstream density in kilograms per cubic metre;

$\rho_2$ is the downstream density in kilograms per cubic metre.

NOTE 1 The above equations are equivalent to the equations from EN/IEC 60534-2-1 ignoring the piping geometry factor, the Reynolds number factor and the expansion factor. The factor 31,6 is the constant $N_6$ in EN/IEC 60534-2-1.

EXAMPLE 1 A valve has $K_{\mathrm{VS}}$ equal to 0,5 m³/h and the maximum seat leakage $Q_{\mathrm{V2}}$ measured downstream of the valve is 0,02 liter air per minute (= 0,001 2 m³/h) with air inlet temperature of 20 °C, inlet pressure of 6,5 bar and outlet pressure of 1 bar. The upstream density of air at 20 °C and 6,5 bar is 7,742 kg/m³, and the downstream density is 1,194 kg/m³. $L$ can then be calculated to $L$ = 100 % × 0,001 2 × 1,194 / (31,6 × 0,5 × (5,5 × 7,742)$^{0,5}$) = 0,001 4 %.

EXAMPLE 2 A valve has $K_{\mathrm{VS}}$ equal to 300 m³/h and the maximum seat leakage $Q_{\mathrm{V2}}$ measured downstream of the valve is 1 m³/h. Conditions are as in Example 1. $L$ can then be calculated to $L$ = 100 % × 1 × 1,194 / (31,6 × 300 × (5,5 × 7,742)$^{0,5}$) = 0,001 9 %.

NOTE 2 Given a leakage percentage and flow coefficient of a valve, the corresponding leakage flow for a different fluid than air can be estimated for a given set of operating conditions using equation (1), see Annex N for more information.

NOTE 3 A simplification is made by omitting the piping geometry factor and the expansion factor in equations (1) to (3). This means that the true leakage rates in some cases will be somewhat lower than calculated. Leaving out the expansion factor in equations (1) to (3) will give a small deviation towards smaller true leakage rates at small $\Delta p$. With rising $\Delta p$ the true leakage rate can decrease to 50 % of the calculated leakage rate.

Valves with back seat shall be checked to ascertain that the back seat is sufficiently tight to allow change of valve packing without danger to the operator.

*Replace 9.3 and Table 5 with:*

**9.3 Seat sealing capacity**

This clause applies to components where internal seat tightness is a design feature.

For seat tightness class A to G as defined in 7.7.1 each valve shall be tested.

For a given seat tightness class, the calculated leak percentage (calculated from the measured leakage rate) or the measured bubbles per minute shall be less than the limits given in Table 5.

The differential pressure, $\Delta p$, given in Table 5 shall be used in the tests, and the measuring equipment shall be able to detect a leakage rate corresponding to the limits given in Table 5.

If any changes are made to the test procedure as described above (e.g. in the case of a helium leakage detection test) the test shall be carried out in a manner which ensures a reliable assessment.

**Table 5 — Production test requirements for seat tightness**

| Seat tightness class | Test pressure | Maximum leakage percentage, *L*, or maximum bubbles per minute |
|---|---|---|
| A | Maximum differential pressure[a, b] | Zero bubbles or equivalent measured during one minute [c] |
| B | Maximum differential pressure[a] | Depends on seat diameter[c, d] |
| C | 5,5 bar | 0,002 % |
| D | 5,5 bar | 0,01 % |
| E | 5,5 bar | 0,025 % |
| F | 5,5 bar | 0,05 % |
| G | 5,5 bar | 0,1 % |
| H[e] | — | — |

[a] For manual valves, see Table 4 for suggested upper limits to the maximum differential pressure.

[b] For safety valves, the maximum differential pressure shall be the to 0,9 × set pressure of the valve.

[c] For tightness classes A and B the number of bubbles or equivalent is measured downstream of the valve

[d] Leakage rates for seat tightness class B (1 bubble/min = 1 $mm^3$/min):

| **Seat diameter** (mm) | **Allowable leakage rate** (bubbles/min) |
|---|---|
| < 25 | 5 |
| 25 | 5 |
| 32 | 7 |
| 40 | 9 |
| 50 | 11 |
| 65 | 14 |
| 80 | 18 |
| 100 | 22 |
| 125 | 28 |
| 150 | 34 |
| 200 | 45 |
| > 200 | 45 |

[e] For seat tightness class H, testing should be carried out to verify the seat tightness specified in the technical literature. For instance by statistical methods.



*Add a new Annex N:*

# Annex N
(informative)

# Estimating seat leakage rate knowing leakage percentage

When the seat tightness class of a valve is known, then the maximum expected mass flow rate of leaked refrigerant can be calculated using Equation 1 from clause 7.6.2:

$$Q_{\mathrm{M}} = \frac{L}{100} \cdot 31{,}6 \cdot K_{\mathrm{VS}} \cdot \sqrt{\Delta p \cdot \rho_1} \qquad \text{(N.1)}$$

In Equation N.1 the density of the refrigerant in the system and the differential pressure across the valve shall be used.

EXAMPLE 1 A solenid valve with a $K_{\mathrm{VS}}$ of 0,7 $m^3/h$ is placed in the liquid line of a refrigeration system. The refrigerant is R290 (propane), the condensation temperature is 30 °C and the evaporation temperature is -10 °C. The solenoid valve is rated in class C, meaning the maximum leakage percentage is 0,002 %.

The inlet density to the solenoid valve is (saturated propane liquid at 30 °C) 484,1 kg/m3 and the pressure difference across the valve when it is closed is 7,338 bar (difference between condensation and evaporation pressure).



When the solenoid is closed, the maximum expected refrigerant leakage rate can be calculated as:



$$Q_M = \frac{0{,}002}{100} \cdot 31{,}6 \cdot 0{,}7 \cdot \sqrt{7{,}338 \cdot 484{,}1} = 0{,}026 \frac{kg}{h} \qquad \text{(N.2)}$$

This leakage mass flow rate corresponds to a cooling capacity of approximately 2 W.