A NEW METHOD TO QUANTIFY THE LEAKAGE SCENARIOS (FREQUENCIES AND FLOWRATES) ON HYDROGEN HIGH PRESSURE COMPONENTS

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ABSTRACT

This work is part of the MULTHYFUEL E.U. research program [1] aiming at enabling the implementation of hydrogen dispersers in refuelling stations. One important challenge is the severity of accidents due to a leakage of hydrogen from a dispenser in the forecourt. The work presented in this paper deals with the quantification of the leakage scenarios in terms of frequencies and severities.

The risk analysis exercise, although performed by experts, showed very large discrepancies between the frequencies of leakages of the same categories and even between the consequences. A large part of the disagreement comes from the failure databases chosen as shown in the paper. The mismatch between the components on which the databases have been settled and the actual hydrogen components may be responsible for this situation. However, as it stands, limited confidence can be laid on the outcome of the risk analysis.

A new method is being developed to calculate the frequencies of the leakage and the flowrate based on an accurate description of each component and of each hazardous situation. For instance, the possibility for a fitting to become untight due to pressure cycling is modelled based on the contact mechanics. Human errors can also be introduced by describing the tasks. In addition of the description of the method, the application to a disperser is proposed with some comparison to experiments. One of the outcomes is that leakage cross sections can be much larger than expected.

1.0 INTRODUCTION

The possibility to use hydrogen outside the traditional industrial fields of metallurgy, electronics, chemistry, … emerged with the concept of "hydrogen economy" about 50 years ago [2]. This was mostly an idea but already targeting the limitation of the pollution by vehicles. The concept was enlarged later defining more clearly the new usage of hydrogen as an "energy vector" complementary to electricity [3]. Today, market studies consider that this new economical field is developing [4] pushed by the pressure set by the population of developed countries to decrease the CO_2 emissions. People seem aware that a significant step toward the "neutral emission" target would only be possible if the transportation sector is deeply decarbonized. Fuelling vehicles using hydrogen (produced from renewables) appears as a solution.

Progressively, the elements of the value chain have been developed starting from the production of "green hydrogen" [5], extending to the transportation [6], the dispensing [7] and hydrogen fuel cell vehicles [8]. The deployment of Hydrogen Fuel Cell vehicles is particularly marked in Europe with the erection of tens of Hydrogen Refuelling Stations (HRS) in the last decade [7]. To a large extent, the available HRS are used by professional drivers like for buses, trucks, and some taxis. Not yet "people from the street". But if hydrogen fuelled vehicles are considered as essential contributors to greening mobility, then a large-scale deployment of HRS is critical.

The achieve this, it is recognized that HRS should be part of conventional refuelling stations where hydrogen dispensers will stand alongside with standard gasoline, electrical, GNV, LPG dispensers. Considering this, they are many challenges regarding the reglementary framework, the technical specifications, and the safety of the refuelling operations. The aim of MULTHYFUEL E.U. sponsored project is to provide data to fill the gaps and propose answers.

In the present work, the specific issue of the safety of the refuelling of a private hydrogen vehicle is addressed targeting the potential leakage scenarios during the operation. In the following sections, some technical details of the dispenser are first given, then how the leakage scenario would be evaluated using standard approaches, third a novel way of investigating the leakage scenario is presented and lastly how this new approach could be checked.

2.0 HYDROGEN DISPERSERS IN REFUELLING STATIONS

If hydrogen as a fuel present many interesting properties – clean, high specific heat of combustion, flexibility – some significant drawbacks shall be recalled, justifying extensive research activities especially about safety. In table 1 are shown the major standard hazardous properties of hydrogen as compared to more traditional fuels likely to be used in refueling stations [9]. The flammability range of hydrogen-air is 5-10 times larger; the minimum ignition energy is 5-10 times smaller, and the maximum burning velocity is about 5-10 times larger. In short, the hazards (explosions and fires) due to accidental releases of hydrogen in air might be 5-10 larger as compared to hydrocarbons.

Table 1 : Physical properties of hydrogen and other fuels

Properties:	unit	Hydrogen	Methane	Propane	Gasoline
Molecular weight	g/mol	2.016	16.043	44.10	~110
Ignition range in air (LFL – UFL)	[% v/v]	4 – 75	5.3 - 15	2.2 - 9.6	0.79 – 8.1
Detonation range in air (LDL – UDL)	[% v/v]	13 – 65	6.3 - 13.5	3.1 – 7	1.1 - 3.3-
Stoichiometric composition in air	[% v/v]	29.5	9.5	4	1.8
Minimum ignition energy	[mJ]	0.02	0.28	0.25	0.23 - 0.46
Combustion heat	[MJ/kg]	120	50	46	45
Auto-ignition temperature	[°C]	585	537	470	228 – 501
Vapour/air density		0.1	0.6	1.6	~3

The way hydrogen is to be used also matters. Today, the industrial route which seems favored is to feed and store hydrogen in vehicles under high pressure, 35 or 70 MPa, inside composite reservoirs [10]. The higher the pressure, the larger the flowrate of any potential leakage and the higher the probability of ignition, either due to the size of the flammable cloud then able to reach more ignition sources or due to some spontaneous ignition mechanism [11, 12]. There is then a clear need to keep any leakage under control which requires quantifying the leakage frequencies and flowrates. This questioning covers all parts of the refueling surely but is more sensitive at the dispenser where end-users stand, which are not specifically trained, and where the number of solicitations of the high-pressure equipment is very large.

To investigate this "hot spot", technical details of the dispenser and dispensing process are needed. Whilst standards were developed for some components [13 to 16], there is some flexibility about the equipment and configuration of the dispensers [17]. Choices were made amongst the partners of the project after consulting various stakeholders. The focus is laid on high pressure dispensing of light duty vehicles because of the highest degree of solicitation of the dispensers (Figure 1). But later, other configurations could be analyzed.

Designation	Inlet diameter	Outlet diameter	Working pressure
Gas Detector	-	-	-
Check Valve	1/4" DRC	1/4" DRC	250 bar
Heat Exchanger	3/8" C&T	3/8" C&T	975 bar
Hose	3/8" C&T	3/8" C&T	875 bar
Flow Valve	1/4" C&T	1/4" C&T	975 bar
Flow Valve	9/16" C&T	9/16" C&T	975 bar
Double Block and Bleed	9/16" C&T	9/16" C&T	975 bar
Pressure Control Valve	9/16" C&T	9/16" C&T	975 bar
Pressure indicator and transmitter	1/4" C&T	1/4" C&T	975 bar
Pressure Safety Valve	-	-	975 bar
Restricted Orifice	1/4" C&T	1/4" DRC	975 bar
Solenoid Valve	6-8 mm	4-6 mm	10 bar
Temperature Transmitter	1/4" C&T	-	-
Shock Detector	-	-	-
Break-Away	3/8" C&T	3/8" C&T	875 bar
Nozzle	3/8" C&T	3/8" C&T	875 bar

Figure 1 : typical arrangement and equipment inside a H_2 dispenser (from [17])

Some details of the components can be found in the catalog of the suppliers but most of the components were procured to perform more accurate size measurements and to estimate the leakage flowrates (see later). Leakages may result from the everyday use of the device. A typical duration of a refueling is 2 mn. Assuming the customer will stay around the dispenser for 5 mn, the order of magnitude of the number of refuelings in a year could be as large as 10000 (on a highway for instance). Some maintenance would also be necessary, at least once a year requiring unmounting 10 fittings each time.

2.0 CURRENT APPROACHES

To estimate the leakage frequency and flowrate, it is common practice to use failure/leakage databases. Perhaps the most common ones are shown on Figure 2 highlighting the links between them. The raw data come from field observation mostly in the oil and gas industries and date back from the eighties. Only more recently, efforts have been done to incorporate data specific to hydrogen technologies.

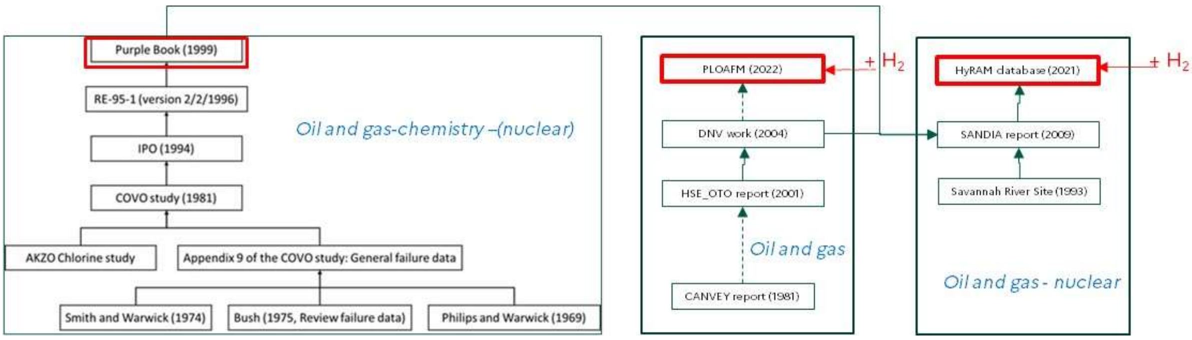
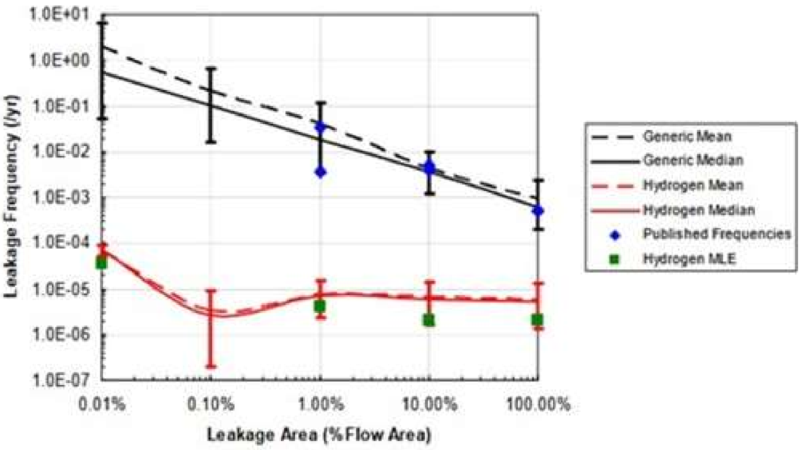


Figure 2 : history and filiations of some renown leakages databases [18, 19, 20]

It was shown earlier [21, 22], that even for case studies chosen in the field from which the databases come from, the choice of the scenarios and associated frequency assessment greatly varied, from one expert to another. The frequencies could vary by several orders of magnitude.

The situation could even be more difficult when applying such databases in the hydrogen energy sector where the products and the technologies are specific [23]. This was considered in the newest databases like PLOAFM and HyRAM in which equipment typically used to handle hydrogen are included. Since only scarce information about hydrogen specific equipment is available, a Bayesian approach was used to adapt generic data to the hydrogen field [20]. A significant twist of the generic data is obtained as shown on Figure 3. Note that the results are presented as a frequency versus the leakage area expressed as a fraction of the full flow area.



Results of Bayesian analysis for pipe joint leak frequency.

Figure 3: example of the application of a Bayesian treatment on generic data (from [20])

In MULHYFUEL, it was attempted to make the best use of this information for the hose of the system presented in Figure 1. It can be seen in Table 2, that very significant discrepancies remain even when using the newest version of the databases (work done by the experts of the project).

Table 2 : leakage scenario rating for the hose of Figure 1

Event	Pressure	DATABASE (leak/year)		
		BEVI (purple book)	Sandia (HyRAM)	Norskeolje&gass PLOAFM
Loss of H_2 containment (medium leak 10%) on hose	700 bar	10^{-3}	10^{-4}	10^{-5}
		10^{-3}	10^{-4}	10^{-4}
		10^{-3}	10^{-3}	10^{-4}
Loss of H_2 containment (medium leak 10%) on hose	700 bar	10^{-3}	10^{-4}	10^{-4}
		10^{-3}	10^{-4}	10^{-5}
		10^{-3}	10^{-3}	10^{-5}

One of the reasons for this, already pointed [20], is that the leakage frequency is very sensitive to generic databases used to develop, via the Bayesian method, the modified databases. In other words, technology is important, and some similarity is necessary between the equipment represented in the generic databases and those implemented in the hydrogen energy sector (Figure 4).

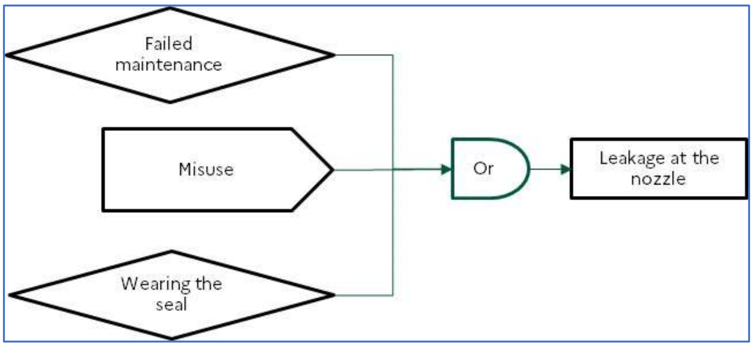
Figure 4 : typical kind of valve represented in the generic database (left) and in the dispenser (right)

In addition, the databases, as they stand today do not establish a link between the leakage scenario and the reasons for the leakage which is a significant difficulty when it comes to mitigate the risk.

3.0 ADOPTING ANOTHER PERSPECTIVE

The principle of the method was proposed in a preceding research work [24] and resembles that published earlier [25]. For each component of Figure 1, a fault tree is produced to describe as accurately as possible the conditions leading to a leakage. The tree is developed deep enough so that the "root" events are simple and well known or generic enough to be reasonably quoted.

A simple example is given on Figure 5 for the nozzle. A leakage (Figure 5 a) is possible either because the nozzle was "misused" (not properly plugged on the dispenser and an incoming car drives over it, or because a car drives our without removing the nozzle) or because of the seals in the nozzles were worn or because a mishap occurred during the plugging operation (dirt/object on the seal). The root events in the diamonds are generic although some argumentation is needed to quote them. The other causal event of the leakage (Misuse) is further developed (Figure 5-b) and broken down is "simple" root events – the circles – for which a probability could be found from the literature. Note the safety measures are not incorporated at this stage.



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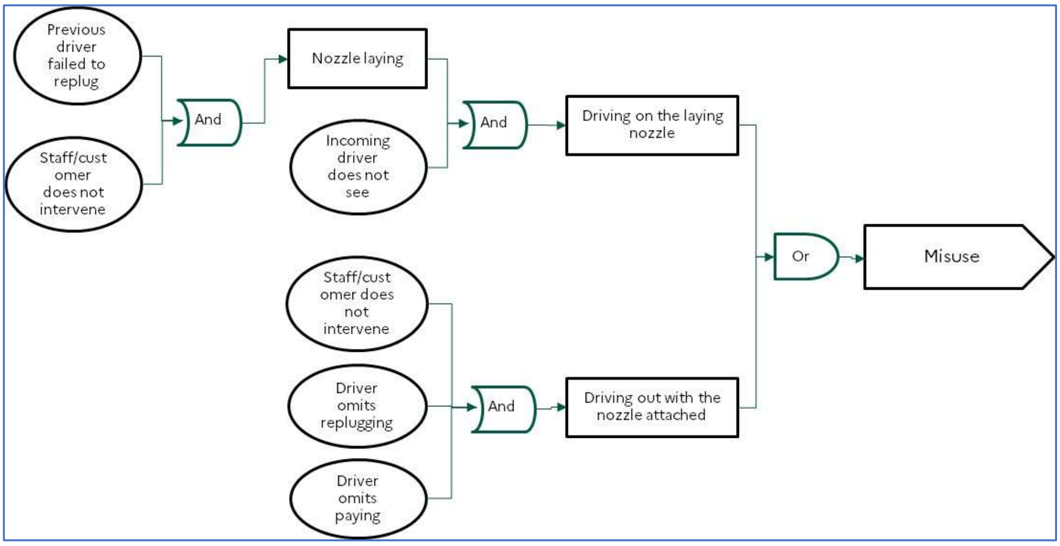


b

Figure 5 : fault tree for the event "leakage at the nozzle"

The reasoning concerning the "root" events is shown in table 3 (three first items). With this information and the frequency of refuelings (F_{ref}=10000 year^{-1}), the frequency of a leakage at the nozzle reads :

$$F_{leak-nozzle} = F_{ref} \cdot \frac{1}{100} + F_{ref} \cdot \left[\left(\frac{1}{100} \cdot \frac{1}{1000} \cdot \frac{1}{1000} \right) + \left(\frac{1}{100} \cdot \frac{1}{1000} \cdot \frac{1}{1000} \right) \right] + F_{ref} \cdot \frac{1}{1\,000\,000} \approx \frac{1}{100}$$

Table 3 : details about the “root” events.

Root event	Argumentation
Failed simple operation (circles Fig. 5)	1 operation failed/ 1000 similar ops (ex : opening a valve) [26]
Failed procedure (diamond Fig.5: maintenance)	For most components, a correct and tight mounting will require : checking the sealing zones (seats/seals), adjusting correctly the sealing pieces (seat, stem or pipe, seal or pressing piece, screw) and tightening according to the user guide. Thus typically, 4 pieces and 3 operations. The two first operations will be repeated 4 times and the last only one. Typically 9 independent operations. The probability that a component has not been correctly assembled is then about 1/100 (The component may not leak but this can be considered as the upper bound of the leakage probability)
Wearing of the seal (diamond: Fig. 5)	The tightness of the junction between the nozzles and the receptacle of the car is ensured by a toroidal polymeric seal (internal diameter 12 mm, 2 mm thick). The wear rate is 10^{10} $\mu m^3/(N.km)$ [27]. The force is at most that produced by the pressure (70 MPa) on the O ring and the sliding path at each refueling is 10 mm. Assuming that leakage is possible after 10% of wearing of the seal (elasticity cannot compensate anymore), the number of cycles before leaking is 750 000. So, the leakage frequency due to wearing is on the order of 1/1 000 000 $year^{-1}$.
Corrosion	The corrosion rate of a standard steel in a normally dry and protected environment (in house) is 5 µm/year. In a polluted environment (salty, acid) this rate is multiplied by 3. A stainless steel will corrode 8 times less [28]. The loss of confinement occurs when the residual thickness is not sufficient to withstand the internal pressure (standard mechanical laws). Metal pipes are much more vulnerable than other components, usually much more massive. For a 9/16’’ stainless steel pipe (316L, ultimate strength = 800 MPa) in a corrosive environment, the failure occurs after 5000 years so a failure frequency on the order of 1/1000 $year^{-1}$.
Fatigue	The Wohler model is applied : $\frac{N}{N_{end-lim}} = \left(\frac{\sigma_{end-lim}}{\sigma}\right)^{1/0.12}$ where N is the number of cycle before rupture, σ the amplitude of variations of the stress and the parameters labelled “end-lim” are characteristics of the material. $N_{end-lim}$ is about 10^7 cycles for steels [29] (10^5 for polymers [30]) and $\sigma_{end-lim}$ is half the ultimate strength of the material. For the stainless steel (316L) currently used the ultimate strength is about 800 MPa. The polymer used in the hose is POM with an ultimate strength of about 100 MPa.
Untightening	Screwed components (fittings, glands) are prone to loosen when submitted to variations of the load applied on the contact surface of the thread [31, 32, 33]. The stresses result from the pressure and temperature cycling (Appendix 1). The loosening action could either result from an axial loading or from a bending effort. A modelling is proposed in appendix 2 showing first that axial loading is predominant and second that thermal stresses are commensurate only for long pieces submitted to a longitudinal thermal gradient. A friction coefficient of 0.1 is retained (unlubricated contact [34]). The experimental observations can be correctly replicated assuming slight differences in the deformation properties of the materials (1% difference in the Young Modulus).

The other root events were issued from a preliminary analysis. This procedure can be applied to all components (Table 4). Note that some root events are specific to certain items. Wear concerns only valves and the seals of the nozzle. Fatigue and corrosion are not applied to valves or fitting since, because of their relatively large thicknesses of steel they seem much less vulnerable than pipes. To estimate the fatigue of the hose, the various components are assumed to behave as a composite with mixed mechanical properties except for the Wölher model (POM data).

Table 4 : Component failure frequency and leakage area (% full cross section)

Component	Scenarios	Leakage frequency (y^{-1})	Leakage area (%)
9/16’’20000 psi pipe	Fatigue corrosion	10^{-7} 10^{-4}	100 (rupture) 100 (rupture)
Hose (ID 4 mm)	Fatigue Misuse	10^{-2} 10^{-4}	100 (rupture) 100 (rupture)
Nozzle (3/8’’)	Procedure (plugging) Misuse Wear (seals)	10^{2} 10^{-4} 10^{-2}	9 (along receptacle) 100 (rupture) 9 (along receptacle)
Breakaway	Procedure (maintenance) Fatigue	10^{-2} 10^{-3}	9 (along receptacle) 9 (along receptacle)
Flow valve (9/16’’)	Procedure (maintenance) Wear (seals)	10^{-2} 10^{-4}	4 (along nut thread) 2 (along stem)
Flow valve (1/4’’)	Procedure (maintenance) Wear (seals)	10^{-2} 10^{-4}	24 (along nut thread) 15 (along stem)
Pres. saf. Valve (3/8’’)	Procedure (maintenance)	10^{-2}	1 (along stem)
9/16’’ fitting	Procedure (maintenance) Untightening	10^{-2} 10^{-2}	5 (along pipe) 5 (along pipe)
3/8’’ fitting	Procedure (maintenance) Untightening	10^{-2} 10^{-2}	8 (along pipe) 8 (along pipe)
1/4’’ fitting	Procedure (maintenance) Untightening	10^{-2} 10^{-2}	11 (along pipe) 11 (along pipe)

Going into the details of the events (maintenance for instance) made it possible to guess the leak flow path as for instance for the valves (Figure 6). If the root event is (failed) “maintenance procedure” (untightened for instance), the leaking gas will flow predominantly (largest section) through the outer diameter via the screw of the nut (Figure 6-left). If the root event is wear (of the seals), the leaking gas will flow along the stem through the driving screw (Figure 6-right). The data on the catalogues of the supplier may not be accurate enough and it is far better to procure the items and do the proper measurements. For pieces needing to move rather freely in another one, the gap between them when inserted is on the order of 0.1 mm. The leakage cross sections were determined by measuring the necessary dimensions on specimens, but a discharge coefficient was incorporated considering the head losses along the path.

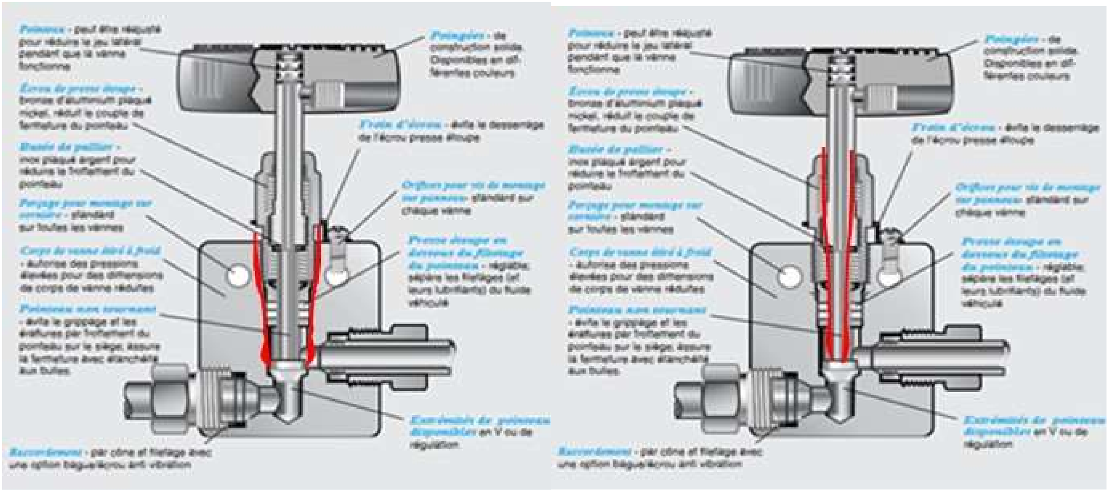


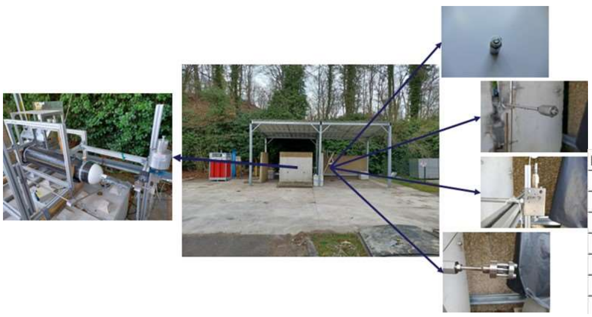
Figure 6 : example of leakage path in a valve (left : failed maintenance, right : wearing of the seals)

4.0 VERIFICATION & VALIDATION

Originally, it was intended to check this method based on existing data. The latter would need to be rather accurate concerning the circumstances of the leakage and the nature of the leaking equipment. We failed to find this kind of data.

Instead, the decision was made first to verify thoroughly the method, especially the basis data (human error frequency, parameters of the Wölher curves, ...) and the models. Then some key scenarios will be reproduced experimentally, for instance the wearing of the seals of the valves, the fatigue of the hose and the untightening of fitting due to pressure cycling.

Separately, an experimental setup was designed to measure the leakage flowrates and compare to the full-bore flowrate in order to verify the predicted leakage cross section. The equipment is shown on figure 7. A 50 litres-1000 bar type 2 vessel is filled with compressed H_2 up to 800 bars. The temperature and the temperature in the vessel are measured. The component to test is connected to the vessel via a high-pressure automatic valve. The mass flowrate is deduced from the pressure and temperature measurements using the Abel-Noble equation of state. The component is conditioned to mimic the targeted scenario. For a fitting which is loosening, the ultimate situation is obtained by applying the fitting on its seat manually. Some preliminary results are presented on Figure 7. The predictions are not unreasonable.



P (b)	component	mass flowrate (g/s)	% full cross section
800	full bore 1 mm	*19*	
800	full bore 2.6 mm (1/4'')	*128*	100
800	full bore 5 mm (3/8'')	*474*	100
800	full bore 7.8 mm (9/16"")	*1154*	100
800	Maximator U fitting 9/16''	30-50	3,5
800	Maximator U fitting 3/8''	15-30	4,2
800	Maximator U fitting 1/4''	10	7,8

Figure 7 : experimental device to measure the leakage flow rates (left) and some results (right)

6. CONCLUDING REMARKS

It is acknowledged that the uncertainties associated to the use of failure databases to estimate the leakages frequencies and flowrates are too large and raises doubts on the relevancies of safety studies. This difficulty is particularly significant for industrial field using new technologies for which the feedback from experience is very limited. Within the scope of MULTHYFUEL project sponsored by

E.U., a new method was developed to produce leakage frequencies and leakage cross sections as function of the constitution of components and usage modes. A combination of fault trees and physical models is used.

The method needs to be validated. This work is underway, but the partial results obtained about the potential leakage cross sections are encouraging. The main advantage of the method is that it seems fully predictive. But a lot of information is necessary, and the formatting of the results is different from that provided by the existing databases. The latter provides a frequency times a percentage of the full cross section on a piping for instance. To obtain this with the present method, the pipe needs to be described and in particular, the number of fittings and flanges has to be provided. Doing this for the components in the dispenser (5 valves, tens of fittings, pipes), the data from table 5 would tell that a "small" leak (5% full cross section) will occur 10^{-1}/y while a full bore only 10^{-4}/y.

On the safety side, the analysis may also clarify the needs. The weakest point is the plugging of the car on the dispenser, justifying at least a very robust checking procedure (SIL level 4 ?).

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Appendix 1 : Thermal stresses vs Pressure induced stresses

In the present context pressure cycling is a source of fatigue. Using standard mechanical rules, the circumferential and longitudinal stresses inside a piece of pipe may read respectively:

$$\sigma_{circum} = P \cdot \frac{D_{out}^2 + D_{in}^2}{D_{out}^2 - D_{in}^2} \quad \text{and} \quad \sigma_{long} = P \cdot \frac{D_{out}^2}{D_{out}^2 - D_{in}^2} \qquad \text{(A1-1)}$$

Where D_{out} and D_{in} are the outer and inner diameter of the pipe and P the internal pressure. The circumferential and longitudinal stresses in a 9/16'' piece of pipe (D_{out}=14 mm and D_{in}=8 mm) submitted to 70 MPa internal pressure would amount 140 MPa and 100 MPa respectively.

But temperature cycling may also induce stresses because of the thermal dilatation. Only at high/low enough temperature (hundreds of °C), a change in the mechanical nature of the steel would happen. Such a levels of temperature is not considered here, a few tens of degrees of temperature elevation is rather expected. (±40°C). Differential thermal dilatation inside the bulk of the material is responsible for the appearance of thermal stress. The stress due to the temperature may be roughly estimated using the expression below ([35]):

$$\sigma = \frac{E \cdot \alpha \cdot \Delta T}{1 - \nu} \qquad \text{(A1-2)}$$

Where E, $\boldsymbol{\nu}$ and α are respectively the Young Modulus, the Poisson coefficient and the thermal dilatation coefficient of the steel. Typical values for 316L are respectively: 200 GPa, 0.3 and 17. 10^{-6} 1/°C. ΔT is the maximum temperature difference inside the sample (between the inner and outer surface of a pipe for instance).

To estimate this some heat exchange calculation should be done. If a hot gas is flowing inside the pipe at temperature T_i, the inner and outer temperature of the wall are respectively Tp_i and Tp_e, the external ambient temperature is T_e then, as a first approximation:

$$\frac{\Delta T}{T_i - T_e} = \frac{Tp_i - Tp_e}{T_i - T_e} = \frac{e/\lambda}{1/h_i + e/\lambda + 1/h_e} \qquad \text{(A1-3)}$$

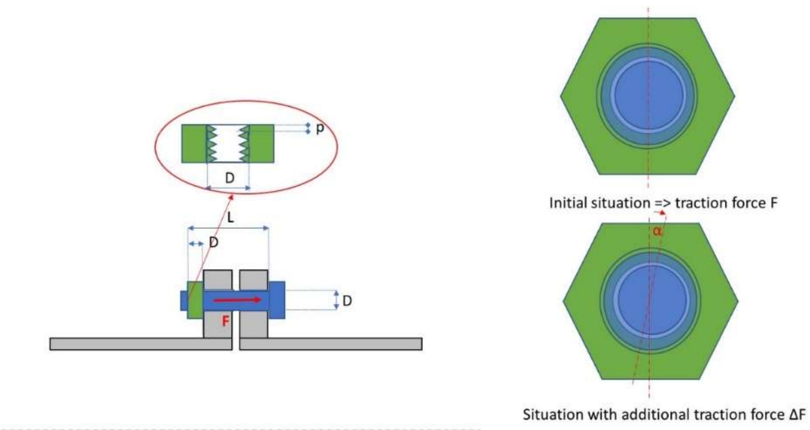
Where λ is the heat conductivity of the steel (typically 15 W/m.°C), h_i and h_e the internal and external heat exchange coefficients. At the denominator, the term in h_e dominates since only natural convection is expected there with a rather low value for h_e (typically 15 W/m^2). So, with e=3 mm, the ratio of temperatures in the above expression is less than 1% ! So even admitting a large temperature difference between the internal and external fluid, typically ±40°C, ΔT may not exceed 1°C. It can easily be shown that the induced thermomechanical stress is below 10 MPa. If now, a piece of pipe (length L) is standing in the open atmosphere T_e with one end held at a significantly different temperature T_i, then the portion of the pipe submitted to the temperature difference $\Delta T = T_i - T_e$ reads :

$$L_{\Delta T} = \sqrt{\frac{\lambda}{2 \cdot h_e} \cdot \frac{D_{out}^2 - D_{in}^2}{D_{out}}} \qquad \text{(A1-4)}$$

And the expression above giving the thermal stress is to be modified by multiplying by $L_{\Delta T}/L$ (only if <1). This situation may concern the heat exchanger zone with L on the order of tens of cms and ΔT on the order of 40°C. $L_{\Delta T}$ is typically a few cms so that $L_{\Delta T}/L$ is on the order of 0.1. The thermal stress due to this situation then amounts about 25 MPa.

Appendix 2 : Loosening by cyclic loading of screwed items

To illustrate the method, the situation addressed is that of flat flanges assembled and tightened by screws as shown below.



Usually, screws are stressed up to 80% of their yield limit (600 MPa for 316L stainless steel). There are several ways to describe loosening. A rather "natural" one is to consider that loosening occurs when the energy stored in tightening the pieces has been dissipated by the frictional movement along the threads. But since both energies are proportional to the forces (tightening force and sliding force respectively) times the products of the nominal diameter and of the angles (tightening angle and sliding angle respectively), it is simpler and equivalent to consider the cinematics: untightening would occur once the tightening angle would have been absorbed but the sum of the small deformation angles produced by the cycles.

Loosening occurs when the surfaces in contact start moving relative to each other. The movement may occur whenever the additional force overcomes the frictional forces or when submitted to the same additional stress the subsequent deformations of the two contact surfaces are different. The first mechanism would require an additional force on the same order of magnitude than the tightening force which is not, by far, the situations under investigation. The second mechanism is modelled below.

A simplified approach is proposed below.

Tightening angle

If σ is the stress applied on the screw at the end of the tightening, L_{sc}, the length of the threaded zone engaged in the nut and p the thread size, then the variation of L_{sc}, ΔL_{sc} is given by the Hooke's law (E is the Young modulus) :

$$\sigma = E \cdot \frac{\Delta L_{sc}}{L_{sc}} = E \cdot \frac{n \cdot p}{L_{sc}} \qquad \text{(A2-1)}$$

Where n is the fraction of thread engaged in the nut to produce ΔL_{sc}. In one turn (2. Π Rad) a screw progresses a distance p. So the crewing angle corresponding to n is β=2.π.n. thus :

$$\beta = 2 \cdot \pi \cdot \frac{\sigma \cdot L_{sc}}{E \cdot p} = E \cdot \frac{n \cdot p}{L_{sc}} \qquad \text{(A2-2)}$$

Loosening due to axial loading [31, 32]

Let G_i be the shear modulus of the screw and nut materials (i=1 or 2), D the nominal diameter of the screw/nut system, ΔF the additional force due to the variations of the pressure/temperature, α_i the average angles of deformation in material i and M the momentum of friction. Then the equilibrium of forces (torsion) reads :

$$M = N \cdot p \cdot \pi \cdot \frac{D}{2} \cdot \tau = f \cdot \Delta F \cdot \frac{D}{2} \qquad \text{(A2-3)}$$

Where τ is the strain on the thread surfaces in contact, f is close to a friction coefficient (incorporates the angles of the thread), p the thread size and N the number of threads in the nut ($N.p=L_{sc}$). The deformations of the contact surfaces can be expressed as :

$$\tau = G_1 \cdot \alpha_1 = G_2 \cdot \alpha_2 \qquad G_1 = \frac{E_1}{2 \cdot (1+\nu)} \qquad \Delta\alpha = \alpha_1 - \alpha_2 \quad \text{(A2-4)}$$

The sliding angle is given Δα and is responsible for the dissipation of the screwing energy. An expression for Δα can be obtained (under the assumption of small deviations between the materials) and then an estimate of the number of cycles required to loosen the screw-nut system :

$$\Delta\alpha = \tau \cdot \left(\frac{1}{G_1} - \frac{1}{G_2}\right) \approx \tau \cdot \frac{\Delta G}{G^2} \approx \frac{\tau}{G} \cdot \frac{\Delta E}{E} \approx \left(f \cdot \frac{\Delta E}{E}\right) \cdot \frac{f \cdot \Delta F}{G \cdot \pi \cdot D \cdot L_{sc}} \qquad N_{axial-cycles} = \frac{\beta}{\Delta\alpha} \qquad \text{(A2-5)}$$

Loosening due to radial loading [33]

The same formulation than above can be used but a link is required to derive ΔF from the radial force F_r or bending moment M_{bend}. If L_r is the normal distance between the nut and the point of action of F_r then :

$$M_{bend} = F_r \cdot l_r = \Delta F \cdot D \qquad \text{(A2-6)}$$

Most parameters can be known rather accurately except f and ΔE/E. The friction coefficient in a unlubricated steel-stell contact should amount about 0.1. Looking at the various values given for the Young modulus for the same material suggests that ΔE/E could be on the order of 1%.

Comparing the present models to the experiments ([33]) reveals that the best fit is obtained when f.ΔE/E is about 0.001 in line with the preceding estimate.