IMPACT OF LIQUID SLOSHING ON THE VEHICLE TANK DESIGN

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Résumé

Dans la littérature, plusieurs normes existent pour la conception et les règles de sécurité pour des véhicules lourds transportant des marchandises dangereuses. Ces normes s'appliquent aux réservoirs des camions citernes utilisés pour le transport par route. L'augmentation de l'émission du gaz à effet de serre et les restrictions gouvernementales poussent les concepteurs à utiliser des matériaux plus légers (Aluminium, magnésium, plastiques, composites, etc.,) afin de réduire le poids des véhicules.

Ceci se traduit par de faible consommation du carburant et de faible émission de CO2. Ces normes représentent la base pour de la conception, la construction, l'essai et l'inspection des réservoirs. Les formes les plus utilisées dans les camions citernes sont le réservoir rectangulaire, le cylindre horizontal, la sphère, le réservoir cylindrique de section transversale trapézoïdale, le paraboloïde et le réservoir conique pour les véhicules spéciaux. Les normes sont valides également pour la conception des joints, des ouvertures, tuyauterie, des valves et des garnitures, des appuis, des renforts circulaires et des protections de dommages d'accidents. Cependant, il n'existe pas de normes pour l'influence défavorable du liquide générant des forces dans les réservoirs partiellement remplis et leurs effets sur la stabilité et le comportement des camions citernes. Généralement les réservoirs sont conçus en se basant sur leur intégrité structurale plutôt que sur les considérations de la stabilité du camion. Les forces et les moments résultant des interactions entre le liquide et le réservoir, dans des manœuvres tel que le freinage dans un virage peuvent générer des variations considérables dans le décalage du liquide et causeront des pressions locales élevées. Ceci peu affecter la structure du réservoir.

Dans cette étude, des modèles analytiques et numériques du mouvement du liquide sont formulés. Ces modèles sont basés sur les équations de Navier-Stokes avec quelques approximations pour le modèle analytique. Les forces de pression sont calculées et comparées à la résistance à la traction pour plusieurs matériaux.

***Mots clés** : Conception, véhicule, réservoir, stabilité, modélisation, matériaux*

Abstract

Several standards exist for the design and safety regulations for vehicles carrying dangerous goods. These standards apply to cargo tanks used for highway transportation. The increase in the Greenhouse gases emission and governmental restrictions, research takes towards the use of lighter materials (Al, Mg, plastics, composites, etc.,) to reduce weight, fuel consumption and CO2 emission.

These standards describe basic requirements for design, construction, testing, inspection, re-testing, qualification and maintenance, and identification aspects of such tanks. The container shapes most commonly associated with road tankers are the rectangular tank, the horizontal cylinder, the sphere, the cylindrical tank of trapezoidal cross section, the paraboloid and the conical tank for special vehicles. The standards also address the design requirements for joints, manholes, openings, piping, valves and fittings, supports, circumferential reinforcements and accident damage protections. However, the standards do not address the adverse influence of liquid sloshing forces in partially-filled tanks on the stability and handling of tank vehicles. In general, the tanks are designed based on their structural integrity rather than on vehicle system stability considerations. The forces and the moments resulting from the interactions between the liquid and the vehicle, in several maneuver situations as turning and braking in turning can make considerable variations of the liquid load shift and will cause high local pressures and dangerous stress on the tank structure.

In this study, analytical and numerical liquid models are formulated based on the Navier-Stokes equations with some assumptions for the analytical model. The pressure forces will be calculated and compared to tensile strength for several materials proprieties. The configuration of the free surface of the liquid used in this study is illustrated in Figure 1. The Figure 2 highlights the numerical modeling of the free surface subject to lateral acceleration and longitudinal acceleration respectively.

***Keys words** : Design, vehicle, tank, stability, modeling, materials*

ملخص

تقدم لنا الأدبيات عدة معايير خاصة للتصميم و قواعد الامنية للشاحنات الناقلة للمواد الخطيرة. هذه المعايير تطبق على الخزانات للشاحنات المستخدمة للنقل برا. إن ارتفاع انتشار الغاز دفيئة و زواجر الحكومية تدفع المصممين على الاستعمال مواد أكثر خفة (الألمنيوم، الماغنسيوم، البلاستيك composites إلخ) للتمكن تخفيف ثقل السيارات. هذا ما ينتج عنه نقص استهلاك الوقود و نقص في انتشار CO_2

هذه المعايير تمثل قاعدة بالنسبة للتصميم و البناء و الاختبار و التحقق من الخزانات. إن الأشكال الأكثر استخدما في الشاحنات الصهريج الخزان المستطيل، الاسطواني الأفقي،الكروي الخزانde ذو المسقط العرضي شبه منحرف إلخ. و تعد هذه المعايير صالحة أيضا بالنسبة لتصميم و الحماية من أثار من الحوادث غير أنه لا يوجد معايير حول الثاني السلبي للسائل لأي من القوى في الخزانات المملوءة نسبيا و تأثيرها على ثبات وسلوك الشاحنات الصهريج. تصمم عموما الخزانات بناء على تكاملها البنيوي أكثر منه على ثبات الشاحنة و إن القوى الزمن الناتجة عن التفاعلات بين السائل والخزان في بعض كالكبح الفرملة في منعرج يمكن إن تؤدي إلى تغييرات معتبرة في للسائل و تؤدي إلى ضغوطات محلية عالية و هذا يمكن أن يؤثر على بنية الخزان.

في هذه الدراسة تم طرح نماذج تحليلية و عددية لحركة السائل على أساس معادلة نافيير ستكوك مع بعض التشابهات – التقاربات- بالنسبة للنموذج التحليلي و ثم حساب و مقارنة قوى الضغط بالمقاومة وكالامتداد للعديد من المواد.

***الكلمات المفتاحية:** التصميم، السيارات، الخزان، الثبات، النمذجة، المواد*

The Longitudinal and directional stability limits of partially-filled liquid cargo vehicles are known to be significantly lower than those of the conventional rigid cargo vehicles due to the unique dynamic interactions between the vehicle and the sloshing liquid cargo. The forces and moments arising from this movements yield considerable dynamic load shift. The dynamic load shift affects the stability of the partially filled tank trucks in an adverse manner and can pose unreasonable risk to highway safety and the environment, when dangerous goods are hauled [1, 2, 3]. Movement or sloshing of the liquid in the tank increases significantly when vehicle weights and dimensions increase. The liquid slosh coupled with heavy vehicle dynamics can lead to a significant reduction in longitudinal and lateral stability and controllability [4, 5], as well as to increased stresses on the container structure [6, 7, 8]. The handling and stability limits of tank trucks are thus dependent upon factors other than normal trucking practices. These factors include tank geometry [9] ; height of the centre of gravity (cg); fill level; lateral and longitudinal load shift during typical highway maneuvers such as turning, braking and lane-change; and liquid-structure dynamic interactions [10, 11, 12].

The tank vehicles (cargo tank) commonly used in transport applications consists basically of a few closed shells of simple shape: elliptical, modified oval and cylindrical with hemispherical, ellipsoidal, or flat ends. The shell components are joined together mostly by welding and riveting; sometimes they are bolted together using flanges. Welded joints are not as strong as the parent plate unless welds are thoroughly inspected and, if flawed, repaired during manufacture - all of which is expensive.

This strength reduction is characterised by the weld or joint efficiency (η = joint strength / parent strength) which varies from 100% for a perfect weld (ie. virtually seamless) through 75-85% for a tolerably good weld. Cylinders Cargo tank are usually made from flat plates which are rolled then welded along longitudinal joints [13]. On the other hand, circumferential joints are used to attach end closures (dished ends or heads) to the cylinder, and to weld together rolled plates for a long tank if plate size availability or rolling machine capacity are restricted. Weld types and efficiencies usually differ for longitudinal and circumferential joints, and therefore the joint stresses in a tank must satisfy both the longitudinal and circumferential stress. The shell thickness is designed to keep the maximum stresses below the yield strength of the material.

In the design of tank cargos, a number of criteria should be considered. There are (1) low mass centre height level and good directional stability, (2) selection of the material for construction of the tank based on a working knowledge of the properties of the material, (2) determination of the magnitude of the induced stress in conformity with the requirements of standard codes, and (3) determination of the elastic stability. To simplify the design and keep the cost of fabrication low, the components of a tank should be made in the form of simple geometries shapes, provides minimum surface area per unit volume and requires minimum wall thickness for a given pressure.

Most tanks are designed as cylindrical shells fabricated of rolled sheet metal or as cylindrical shells that are cast. From the point of view of simplified structural design, the stressed state of the material of a thin-wall shell is considered biaxial. This is permissible because the magnitude of the radial stress in such a tank wall is very small. The stressed state of the shell wall is generally the sum of the two basic components: (1) stressed state due to uniformly distributed forces on the surface as a result of fluid pressure, and (2) stressed state due to the action of the forces and moments distributed around the contour. In general, end stresses should be carefully evaluated, and design measures must be taken to keep them within the safe limit. High values of end stresses are to be avoided, especially for brittle materials. The theory of elasticity requires that the strength be determined by the ultimate stress which the part can withstand without rupture, whereas the theory of plasticity suggests that the strength be determined by the ultimate load the part can withstand without residual deformation. The elastic theory is based on the assumption that the material of the component parts of the tank is in an elastic state everywhere and nowhere does the state of stress exceed the yield point. Several loading combinations are often possible. As design pressure, wind loads, thermal loads, impact loads and cyclic load [13]. We should consider them carefully for a particular situation and select the most probable combination of simultaneous loads for an economical and reliable design. Failure of a the tank may be due to improper selection of materials, defects in materials due to inadequate quality control, incorrect design conditions, and computational errors in design, improper or faulty fabrication procedures, or inadequate inspection. Design pressure is the pressure used to determine the minimum thickness of each of the tank components. It is the difference between the internal and external pressures. The design thickness is the minimum required thickness plus an allowance for corrosion. The nominal thickness is the design thickness of the commercially available material actually used in making the tanks cargos. Design temperature is really more a design environmental condition than a load. Thermal loads originate from temperature changes combined with body restraints or existing temperature gradients. Reduction in structural strength due to rising temperature, increase in brittleness with rapidly falling temperature, and the associated changes in the tank dimensions should be taken into careful consideration for the design. Generally, the standard tank design temperature is equal to the maximum operating temperature of the fluid in the vessel plus margin safety considerations. For low-temperature operation, the minimum fluid temperature is used for the design temperature as reported in [15].

When a thin-walled cylinder is subjected to internal pressure, three mutually perpendicular principal stresses will be set up in the cylinder material, namely the circumferential or hoop stress, the radial stress and the longitudinal stress. Provided that the ratio of thickness to inside diameter of the cylinder is less than 1/10, it is reasonably accurate to assume that the hoop and longitudinal stresses are constant across the wall thickness and that the magnitude of the radial stress set up is so small in comparison with the hoop and longitudinal stresses that it can be neglected. This is obviously an approximation since, in practice, it will vary from zero at the outside surface to a value equal to the internal pressure at the inside surface. For the purpose of the initial derivation of stress formulae it is also assumed that the ends of the cylinder and any riveted joints present have no effect on the stresses produced [15]. Consequently, it is important to determine the important condition and select only the important sets of design loads which can most probably occur simultaneously. The tank cargos are exposed to the environment. We considered that the dynamic pressure force is the most several conditions to take in consideration the static force used in the standard constraint for design tank condition. This dynamic forces change with tank vehicles road operational conditions such as Braking, braking in turning and impact accidents comparatively to static conditions.

DESIGN DEVELOPMENT

Figure 1 show the stress which is set up in resisting the bursting effect of the applied pressure and can be most conveniently treated by considering the equilibrium of half of the cylinder [15].

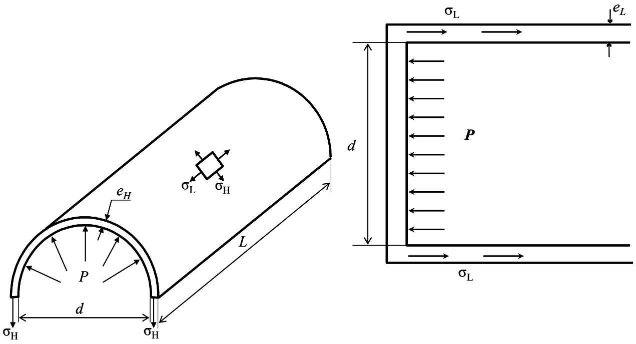


Figure 1 : half of a thin cylinder subjected to internal pressure showing the hoop stress (left) and longitudinal stresses (right) acting on any element in the cylinder surface.

1. Lateral stress or hoop stress:

The total force on half-cylinder owing to internal pressure is formulated as:

$$F_P = p \times A = p \times dL$$

And the total resisting force owing to hoop stress on set up in the cylinder walls is formulated by:

$$F_S = 2L \times e_H \times \sigma_H$$

Then the lateral stress or hoop stress will be calculated by equating forces as:

$$\sigma_H = \frac{pd}{2e_H}$$

If we take the effects of the circumferential joints we add the efficiency factor n_H :

$$\sigma_H = \frac{pd}{2n_H e_H} \qquad (1)$$

Longitudinal stress :

Area of metal resisting stress force is equal to :

$$= \pi\left[\left(\frac{d}{2}+e_L\right)^2 - \left(\frac{d}{2}\right)^2\right] \approx \pi d e_L$$

Then the longitudinal stress force is:

$$F_S = \pi d e_L \times \sigma_L$$

And the total force owing to internal pressure is obtained by:

$$F_P = p \times A = p \times \pi\left(\frac{d}{2}\right)^2$$

The the longitudinal stress will be calculated by equating forces as :

$$\sigma_L = \frac{pd}{4e_L}$$

If we take the effects of the end plates we add the efficiency factor n_L :

$$\sigma_L = \frac{pd}{4n_L e_L} \qquad (2)$$

Allowable working stress-factor of safety :

The most suitable strength or stiffness criterion for any structural element or component is normally some maximum stress or deformation which must not be exceeded. In the case of stresses the value is generally known as the maximum allowable working stress or deign pressure for the tank cargos. Because of uncertainties of loading conditions, design procedures and production methods introduce a factor of safety into their designs, defined as follows:

$$\text{Safety factor} \qquad S_F = \frac{\text{Tensile strength}}{\text{Allowable working stress}}$$

However, in view of the fact that plastic deformations are seldom accepted this definition is sometimes modified to:

$$\text{Safety factor} \qquad S_F = \frac{\text{Yield stress (or proof stress)}}{\text{Allowable working stress}}$$

In the absence of any information as to which definition has been used for any quoted value of safety factor the former definition must be assumed. In this case a factor of safety of 3 implies that the design is capable of carrying three times the maximum stress to which it is expected the structure will be subjected in any normal loading condition. There is seldom any realistic basis for the selection of a particular safety factor and values vary significantly from one branch of engineering to another. Values are normally selected on the basis of a consideration of the social, human safety and economic consequences of failure. Typical values range from 2.5 (for relatively low consequence, static load cases) to 10 (for shock load and high safety risk applications) [**Erreur ! Signet non défini.**] are considered. In this study, we have used a safety factor equal to 3.

When introduce the concept of safety factor in equations 1 and 2. The minimum thickness of the materials for cylindrical shape will be calculated as:

$$e = max\ \ e_H = \frac{pdS_F}{2n_H\sigma_H}, e_L = \frac{pdS_F}{4n_L\sigma_L} \qquad (3)$$

In same manner we can calculate for the elliptical and modified oval cross section by changing only the expression of the area cross section.

Liquid Model :

The following section describes the mathematical equations of Navier-Stokes governing the flow of a liquid. It is a system of nonlinear three-dimensional partial equations. The first equation describes the conservation of mass,

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho V) = 0 \qquad (4)$$

For viscous and Newtonian liquids the momentum equation is

$$\rho\left(\frac{\partial V}{\partial t} + (V.\nabla)V\right) = -\nabla P + (\lambda + \mu)\nabla(\nabla.V) + \mu\nabla^2 V + F \qquad (5)$$

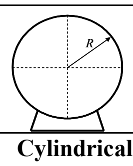
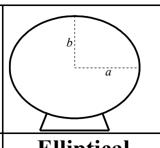
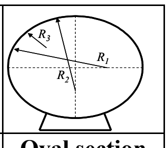
Where: V: Velocity, P: Pressure, ρ: density, λ: Factor of volume compression, μ: Viscosity and F: External force vector.

The numerical resolution of the Navier-Stokes equations was solved by commercial software Fluent [16]. We have used volume of control (VOF) technique to model the free face movement. This technique is robust for multiphase flows implying two fluids or no miscible (liquid, air). This technique has been developed in anterior work [17, 18, 19]. The pressure forces are calculated using the volume integral, such that :

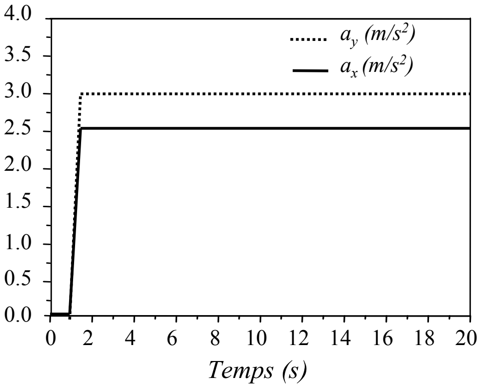
$$F_{px_i} = \sum_{c=1}^{liquid} P_c A_{cx} \quad , \qquad \mathrm{i} \equiv (\mathrm{x,\ y\ and\ z}) \qquad (6)$$

The majority of tank cargos are cylindrical, elliptic or modified cross section. In this study three most used tank section form as cylindrical, elliptical and modified oval section as shown in Table 1. The tank is partially filled (70%) and transport domestic oil (*ρ=966 kg/m ^{3}, ν=0.048 kg/m.s*).

<u>Table 1</u> : tank section form.

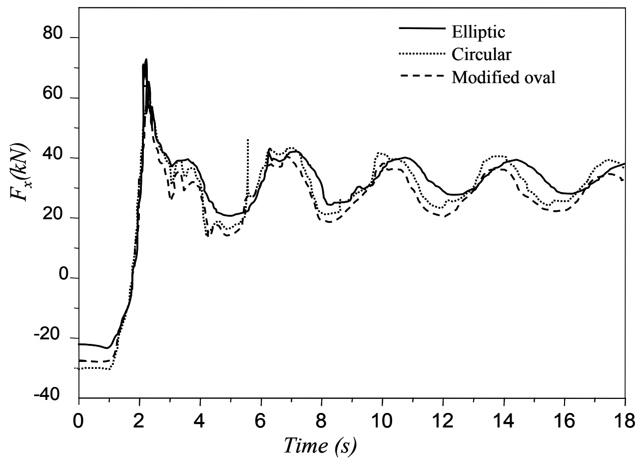
Tank section form			
Characteristics	**Cylindrical** R=1.02 m S= 3.258 m^2 L=7.55 m	**Elliptical** a= 1.08 m , b = 0.96 m S= 3.258 m^2 L=7.55 m	**Oval section** H_1= 2.44 m, H_2= 1.65 m R_1=1.78 m , R_2= 1.78 m , R_3=.39 m S= 3.258 m^2 L=7.55 m

The analyses were performed using modified step functions to realize linearly varying acceleration as shown in figure 2.

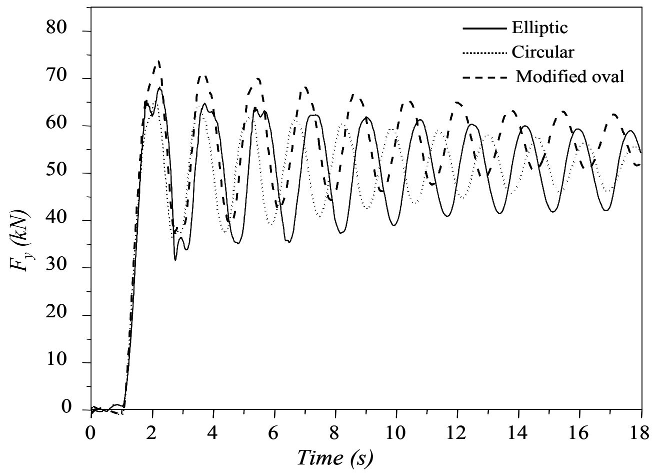


<u>Figure 2</u> : modified step acceleration input

Figures 3 and 4 represents pressure force response for the tank cargo with different tank cross section subjected to acceleration excitation and partially filled (*τ=70%*) condition. From these responses, we get the maximum pressure force. Then, we can calculated the minimum thickness for the tank design cargo.



<u>Figure 3</u> : longitudinal pressure force response for different tank cross section subject to deceleration *(a_x= .25g and τ=70%)*



<u>Figure 4</u> : lateral pressure force response for different tank cross section subject to lateral acceleration *(a_y= .25g and τ=70%)*

The tank material will be chooses following the American Society for Testing and Materials (ASTM) specifications and in accordance with the American Society of Mechanical Engineers (ASME) Code [14, 20]. Aluminum alloys suitable for fusion welding and conforming to the 0, H32 or H34 tempers is shown in Table 2.

Table 2 : material properties

Materiel	Mild steel (Ms)	Stainless steel (Ss)	Aluminum 1 (Al1) 5052-0 5652-0	Aluminum 2 (Al2) 5154-0 5254-0 5454-0	Aluminum 3 (Al3) 5052-H32 5652-H32	Aluminum 4 (Al4) 5154-H32 5254-H32 5454-H32
Tensile Yield Strength (Mpa)	172.36	310.26	89.6	117	193	207
Density g/cm^3	7.85	8	2.68	2.66	2.68	2.66

DESIGN CRITERIA FOR TANK CONSTRUCTION

According to ASME and US Department of transportation [**Erreur ! Signet non défini.**, 21] cargo tanks must be designed under the maximum permissible loads and maximum permissible working pressures, be capable of absorbing the following separately applied static forces: 2g in the direction of travel, 1g horizontally at right angles to the direction of travel, 1g vertically upwards and 2g vertically down wards multiplied by the liquid mass (g =9.81m/s^2 acceleration due to gravity).

As shown in Table 3, thicknesses and tank weight for the operational road conditions (OP) is approximately 90 % less than for design construction conditions (DC). However, in this study we have not take in consideration the effect of baffles. The baffles can reduce more the dynamic forces and increase the difference between these two conditions (OP and DC). These dynamic forces are obtained by modeling the real conditions. In this study we have demonstrated that tank weight can be reduced more by using advanced methods. As shown in Table 3, we have a large margin between OP and DC. This margin can used to decrease tank weight without harm road safety.

CONCLUSION

In this study, we have simulate liquid sloshing in cargo tank by numerical model using real condition, such as exaggerate steady state turning and braking maneuvers. The force results from these maneuvers are used to calculate minimum shell thickness and weight tank for most popular tank shape used in road transportation. This road operational road condition (OP) is compared to the maximum standard design constraint (DC) condition tank construction. The comparison highlights that there is large margin between the two conditions. This margin of 90% approximately can be used to reduce more the tank weight without harming road safety. This results, is important to decrease fuel vehicle consummation or to increase the transported liquid charge.

Table 3 : comparison between operational road and design tank construction

		Maximum Dynamic Pressure Force (N)		Minimum Shell Thickness (mm)			Tank Shell Weight (kg)		
		O.C	D. C	O.C	D. C	Thickness Ratio %	O.C	D. C	Weight Ratio %
Circular	Ms	54772.66	655835.5	0.17	2.09	8.13	66.45	796.75	8.34
	Ss	54772.66	655835.5	0.1	1.16	8.62	37.63	450.87	8.34
	Al 1	54772.66	655835.5	0.33	4.06	8.12	43.66	523.75	8.33
	Al 2	54772.66	655835.5	0.25	3.08	8.11	33.18	397.92	8.33
	Al 3	54772.66	655835.5	0.156	1.87	8.34	20.26	242.9	8.34
	Al 4	54772.66	655835.5	0.145	1.74	8.33	18.75	224.76	8.34
Elliptic	Ms	60777.51	658791.75	0.19	2.1	9.04	73.76	799.52	9.22
	Ss	60777.51	658791.75	0.12	1.17	10.25	41.76	452.65	9.22
	Al 1	60777.51	658791.75	0.37	4.06	9.11	48.44	525.08	9.22
	Al 2	60777.51	658791.75	0.28	3.09	9.06	36.82	399.11	9.22
	Al 3	60777.51	658791.75	0.17	1.87	9.1	22.48	243.76	9.22
	Al 4	60777.51	658791.75	0.16	1.75	9.14	20.81	225.58	9.22
Modified oval	Ms	75304.1	846599.52	0.35	3.94	8.88	91.4	1027.45	8.89
	Ss	75304.1	846599.52	0.2	2.18	9.17	51.74	581.69	8.89
	Al 1	75304.1	846599.52	0.67	7.57	8.85	60.02	674.76	8.89
	Al 2	75304.1	846599.52	0.51	5.8	8.8	45.62	512.88	8.89
	Al 3	75304.1	846599.52	0.31	3.53	8.78	27.86	313.26	8.89
	Al 4	75304.1	846599.52	0.3	3.66	8.2	25.78	289.89	8.89

Note : O.C : Operational road Condition ; D.C : Design construction condition ; MS : Mild Steel ; SS : Stainless Steel ; Al : Aluminium

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