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COMPARING ANALYSIS OF A SPECIFIED PRESSURE VESSEL'S DESIGN METHODS

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Abstract: The utilization range of pressure vessels is really wide, especially in chemical, oil industry and energetic. These instruments store mostly on high pressure and temperature and often hazardous materials. Originally we made the calculation for educational purpose, than we thought it is worthy to release it. In our work we analyzed the shell weakening effect of saddle in a horizontal retention tank. The calculations involve the whole strength control of the tank but considering the maximized length of the work we introduce only the supporting saddle's stress increasing effect. We compared standardized calculations (EN 13445:2009 Unfired pressure vessels- Part 3: Design [1]/ ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 2007 Edition [2]) to Autodesk Inventor software's stress analysis. The two standards and the Inventor's approximate calculation using Finite Element Method brought similar results.
Keywords: pressure, vessel, support, standard, strength control, saddle

1. INTRODUCTION

The utilization range of pressure vessels is really wide, especially in chemical, oil industry and energetic. These instruments store mostly on high pressure and temperature and often hazardous materials. For that very reason the design of these equipments is regulated by strict specifications. To these specifications belong the EN13445-1-5, and ASME SEC VIII DIV 1-2 standards too. The standardized calculations of pressure vessels found on the one part on the membrane stress of shell of revolution and on the other part on the stresses coming from bending the shell. While the previous one has analytical solution for almost every type of shell the latter has solution only with significant simplifications.

2. STRENGTH CONTROL OF HORIZONTAL RETENTION TANK SUPPORTS (SADDLE)

Geometric dimensions from Figure 1 and Figure 2.

Design data:

- Design Internal Pressure: p_d= 0,52 MPa
- Static Head of the fluid fill: p_{st}= 0,024 MPa
- Outside diameter of the cylinder: D_e= 2440 mm
- Tank estimated volume: V= 14,25 m^3
- Shell thickness: e= 8 mm
- Shell Material: P295GH (EN 10028:2009 [3])

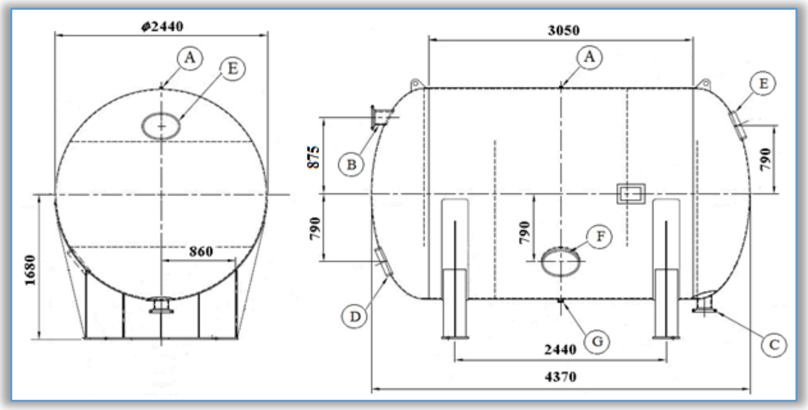

Figure 1. Dimensions of Horizontal retention tank and its supports [1].

L= 3130 mm; H_i= 617 mm; a_1= 343 mm; b_1= 305 mm; b_2= 0 mm (vessels have no reinforcing plate); e_2= 0 mm; a_2= 0

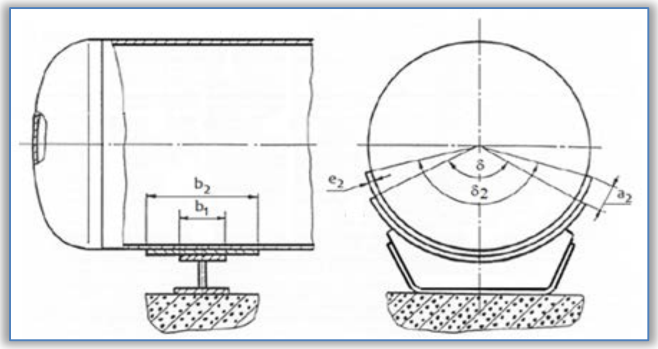

Figure 2. Dimensions of supports [1].

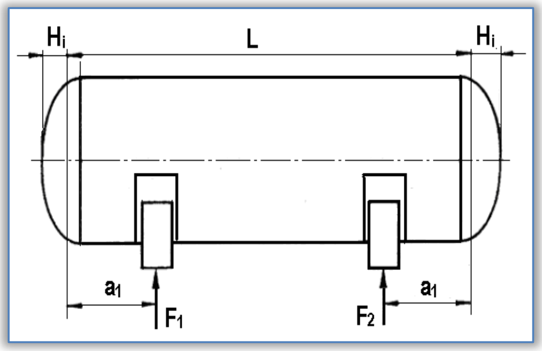

Figure 3. Saddle-reactions [1]

2.1. Control for excess stress caused by supports according to EN 13445:2009 standard [1]

The data of thank are coming from previous calculations. The loads of support are the saddle-reactions shown on Figure 3. The highest load occurs when the tank is full of water (e.g. at test pressure).
Determination of reaction forces:

- W_{RT} = 25.000 N (Retention Tank Weight)
- W_F = 142.500 N (Weight of Fluid)
- W = W_{RT}+W_F = 167.500 N (Total Flooded Shell Weight)
- $F_1 = F_2 = W/2$ = 167.500/2 N = 83.750 N

Maximum value of the reaction at the saddle support from weight and other loads as applicable.
For vessels with two saddles of type A (Figure 3), the calculation is not required when the following conditions are met (the existing values in brackets):

a) there is no external pressure on the tank
b) density of fluid ≤ 1000 kg/m³(density of the stored medium < 1000 kg/m³, density of the testing medium (water) = 1000 kg/m³)
c) permissible stress in shell material ≥ 130 MPa (f_d= 197 MPa > 130 MPa)
d) welding factor of support steel structure ≥ 0.8 (z= 1,0 > 0,8)
e) $a_1 \leq 0.5 * D_i$(a_1= 343 mm < 0.5*2424 mm= 1212 mm)
f) $L \leq {}_{max}$, where L_{max} derived from Figure 4.

- continuous lines: vessels without reinforcing plate)
- dotted lines: vessels with reinforcing plate)

L= 3.13 m <L_{max}= 12 m

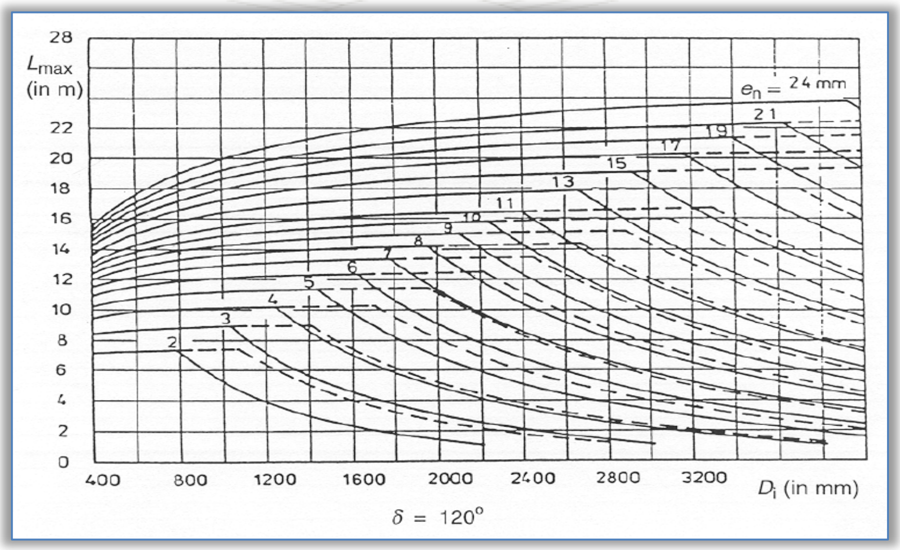

Figure 4. Length (L) – diameter (D_i) – shell thickness (e_n) relationship [1]

g) $b_1 \geq 1.1 * \sqrt{D_i * e_n}$

$$b_1 = 305 \text{ mm} > 1.1 * \sqrt{D_i * e_n} = 1.1 * \sqrt{2424 * 8}\text{mm} = 153{,}2 \text{ mm}$$

where e_n= 8 mm (nominal thickness)
As the circumstances exist, the support is not necessary to be controlled for stress according to EN13445:2009 standard.

2.2. Determination of stresses derived from excess loads according to ASMEVIII-2 [2]

Application of Rules control:

Geometry (the existing values in brackets):

» $\delta \geq 120°$ ($\delta = 120°$)

» $a_1 \leq 0.25*L$ (a1= 343 mm < 0.25*L= 0.25*3130 mm= 782.5 mm)

Reinforcing Plates: vessels have no reinforcing plate

Stiffening Rings: vessels have no stiffening rings

❖ **Moment and Shear Force**

» the moment at the saddle:

$$M_1 = -F_1 * a_1 * \left(1 - \frac{1 - \frac{a_1}{L} + \frac{R_m^2 - H_i^2}{2*a_1*L}}{1 + \frac{4*H_i}{3*L}}\right)$$

where $R_m = \frac{D_e - e}{2} = \frac{2440\ mm - 8\ mm}{2} = 1216\ mm$ mean radius of the cylindrical shell

$$M_1 = -83750\ N * 343\ mm * \left(1 - \frac{1 - \frac{343\ mm}{3130\ mm} + \frac{1216^2 mm^2 - 617^2 mm^2}{2*343*3130\ mm^2}}{1 + \frac{4*617\ mm}{3*3130\ mm}}\right)$$

$$M_1 = -2{,}011 * 10^7\ Nmm$$

» the moment at the center of the vessel:

$$M_2 = \frac{F_1 * L}{4} * \left(1 + \frac{\frac{2*(R_m^2 - H_i^2)}{L^2}}{1 + \frac{4*H_i}{3*L}} - \frac{4 * a_1}{L}\right)$$

$$M_2 = \frac{83750\ N * 3130\ mm}{4} * \left(1 + \frac{\frac{2*(1216^2 - 617^2)mm^2}{3130^2 mm^2}}{1 + \frac{4*617\ mm}{3*3130\ mm}} - \frac{4 * 343\ mm}{3130\ mm}\right)$$

$$M_2 = 4{,}843 * 10^7\ Nmm$$

» the shear force at the saddle:

$$T = \frac{F_1 * (L - 2 * a_1)}{L + \frac{4*H_i}{3}} = \frac{83750\ N * (3130\ mm - 2 * 343\ mm)}{3130\ mm + \frac{4*617\ mm}{3}} = 51784\ N$$

❖ **Longitudinal Stress**

≡ The longitudinal membrane plus bending stresses in the cylindrical shell between the supports:

» top of shell:

$$\sigma_1 = \frac{p * R_m}{2 * e} - \frac{M_2}{\pi * R_m^2 * e} = \frac{0.544\ MPa * 1216\ mm}{2 * 8\ mm} - \frac{4.843 * 10^7}{\pi * 1216^2 mm^2 * 8\ mm} = 40\ MPa$$

» bottom of shell:

$$\sigma_2 = \frac{p * R_m}{2 * e} + \frac{M_2}{\pi * R_m^2 * e} \sigma_2 = \frac{0.544\ MPa * 1216\ mm}{2 * 8\ mm} + \frac{4.843 * 10^7}{\pi * 1216^2 mm^2 * 8\ mm} = 42.6\ MPa$$

≡ The longitudinal membrane plus bending stresses in the cylindrical shell at the support location:

» at points A and B in Figure 5. a): $\sigma_3 = \frac{p*R_m}{2*e} - \frac{M_1}{K_1*\pi*R_m^2*e}$

» bottom of shell: $\sigma_4 = \frac{p*R_m}{2*e} - \frac{M_1}{K_1^**\pi*R_m^2*e}$

where: $K_1 = \frac{\Delta + \sin\Delta*\cos\Delta - \frac{2*\sin^2\Delta}{\Delta}}{\pi*\left(\frac{\sin\Delta}{\Delta} - \cos\Delta\right)} K_1^* = \frac{\Delta + \sin\Delta*\cos\Delta - \frac{2*\sin^2\Delta}{\Delta}}{\pi*\left(1 - \frac{\sin\Delta}{\Delta}\right)}$

$$\Delta = \frac{\pi}{6} + \frac{5 * \delta}{16} = \frac{\pi}{6} + \frac{5 * 2 * \pi}{16 * 3} = 1.178$$

$$K_1 = \frac{1.178 + \sin 1.178 * \cos 1.178 - \frac{2*\sin^2 1{,}178}{1{,}178}}{\pi * \left(\frac{\sin 1{,}178}{1{,}178} - \cos 1{,}178\right)} = \frac{0{,}08258}{1{,}2613} = 0.0655$$

$$K_1^* = \frac{1{,}178 + \sin 1{,}178 * \cos 1{,}178 - \frac{2*\sin^2 1{,}178}{1{,}178}}{\pi * \left(1 - \frac{\sin 1{,}178}{1{,}178}\right)} = \frac{0{,}08258}{0{,}6778} = 0.1218$$

$$\sigma_3 = \frac{0.544\ \text{MPa} * 1216\ \text{mm}}{2 * 8\ \text{mm}} + \frac{2.011 * 10^7}{0.0655 * \pi * 1216^2\text{mm}^2 * 8\ \text{mm}} = 49.6\ \text{MPa}$$
$$\sigma_4 = \frac{0.544\ \text{MPa} * 1216\ \text{mm}}{2 * 8\ \text{mm}} - \frac{2,011 * 10^7}{0.1218 * \pi * 1216^2\text{mm}^2 * 8\ \text{mm}} = 36.9\ \text{MPa}$$

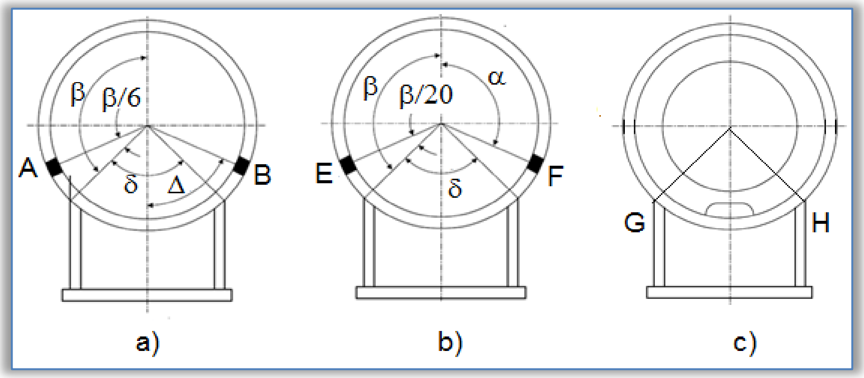

Figure 5. Shear stress maximum values in the cylindrical shell

≡ Acceptance Criteria

The absolute value of σ_1, σ_2, and σ_3, σ_4, as applicable shall not exceed f_d.

$$\sigma_1 = 40\ \text{MPa} < f_d = 197\ \text{MPa},$$
$$\sigma_2 = 42{,}6\ \text{MPa} < f_d = 197\ \text{MPa},$$
$$\sigma_3 = 49{,}6\ \text{MPa} < f_d = 197\ \text{MPa},$$
$$\sigma_4 = 36{,}9\ \text{MPa} < f_d = 197\ \text{MPa}.$$

❖ **Shear Stresses**

As the shell is without stiffening rings and a_1=313 mm < 0.5*R_m = 0.5*1216 mm = 608 mm, so:

≡ The shear stress in the cylindrical shell is a maximum at Points E and F of Figure 5. b)

$$\tau = \frac{K_3 * F_1}{R_m * e}$$
$$K_3 = \frac{\sin\alpha}{\pi} * \frac{\alpha - \sin\alpha * \cos\alpha}{\pi - \alpha + \sin\alpha * \cos\alpha}$$
$$\alpha = 0{,}95 * \left(\pi - \frac{\delta}{2}\right) = 0{,}95 * \left(\pi - \frac{2 * \pi}{2 * 3}\right) = 1{,}99$$
$$K_3 = \frac{\sin 1{,}99}{\pi} * \frac{1{,}99 - \sin 1{,}99 * \cos 1{,}99}{\pi - 1{,}99 + \sin 1{,}99 * \cos 1{,}99} = 0{,}881$$
$$\tau = \frac{0{,}881 * 83750\ \text{N}}{1216\ \text{mm} * 8\ \text{mm}} = 7{,}58\ \text{MPa}$$

≡ The shear stress in the formed head is a maximum at Points E and F of Figure 5. b)

$$\tau^* = \frac{K_3 * F_1}{R_m * e_h} = \frac{0{,}881 * 83750\ \text{N}}{1216\ \text{mm} * 8\ \text{mm}} = 7{,}58\ \text{MPa}$$

e_h: thickness of formed heads

≡ Acceptance Criteria

The absolute value of τ, τ^*, as applicable, shall not exceed 0,8*f_d

$$\tau = \tau^* = 7.58\ \text{MPa} < 0.8 * 197\ \text{MPa} = 157.6\ \text{MPa}$$

❖ **Circumferential Stress**

≡ The maximum compressive circumferential membrane stress in the cylindrical shell at the base of the saddle support:

$$\sigma_6 = \frac{-K_5 * F_1 * k}{e * (b_1 + x_1 + x_2)}$$

where

$$K_5 = \frac{1 + \cos\alpha}{\pi - \alpha + \sin\alpha * \cos\alpha} = \frac{1 + \cos 1.99}{\pi - 1.99 + \sin 1.99 * \cos 1.99} = 0.76$$

k: factor to account for the vessel support condition; k=0,1 is the vessel is welded to the support

x: width of cylindrical shell used in the circumferential normal stress strength calculation

$$x_1 = x_2 = 0$$
$$\sigma_6 = \frac{-0.76 * 83750\ \text{N} * 0{,}1}{8\ \text{mm} * 305\ \text{mm}} = -2.53\ \text{MPa}$$

≡ The circumferential compressive membrane plus bending stress at Points G and H of Figure 5:

$$\text{If} L = 3130\ \text{mm} < 8 * R_m = 8 * 1216\ \text{mm} = 9728\ \text{mm}$$

then

$$\sigma_7 = \frac{-F_1}{4 * e * (b_1 + x_1 + x_2)} - \frac{12 * K_7 * F_1 * R_m}{L * e^2}$$

where $K_7 = \frac{K_6}{4}$, since $\frac{a_1}{R_m} = \frac{343\ \text{mm}}{1216\ \text{mm}} = 0{,}282 < 0{,}5$

$$K_6 = \frac{\frac{3 * \cos\beta}{4} * \left(\frac{\sin\beta}{\beta}\right)^2 - \frac{5 * \sin\beta\cos^2\beta}{4 * \beta} + \frac{\cos^3\beta}{2} - \frac{\sin\beta}{4 * \beta} + \frac{\cos\beta}{4} - \beta * \sin\beta\left[\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin 2 * \beta}{4 * \beta}\right]}{2 * \pi\left[\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin 2 * \beta}{4 * \beta}\right]}$$

$$\beta = \pi - \frac{\delta}{2} = \pi - \frac{2 * \pi}{2 * 3} = 2{,}094$$

$$\frac{3 * \cos\beta}{4} * \left(\frac{\sin\beta}{\beta}\right)^2 = \frac{3 * \cos 2{,}094}{4} * \left(\frac{\sin 2{,}094}{2{,}094}\right)^2 = -0.0641$$

$$-\frac{5 * \sin\beta\cos^2\beta}{4 * \beta} + \frac{\cos^3\beta}{2} = -\frac{5 * \sin 2.094 * \cos^2 2.094}{4 * 2{,}094} + \frac{\cos^3 2.094}{2} = -0.1915$$

$$-\frac{\sin\beta}{4 * \beta} + \frac{\cos\beta}{4} = -\frac{\sin 2.094}{4 * 2{,}094} + \frac{\cos 2.094}{4} = -0{,}2283$$

$$-\beta * \sin\beta\left[\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin 2 * \beta}{4 * \beta}\right] = -2{,}094 * \sin 2{,}094\left[\left(\frac{\sin 2{,}094}{2{,}094}\right)^2 - \frac{1}{2} - \frac{\sin 2 * 2{,}094}{4 * 2{,}094}\right] = 0.4091$$

$$2 * \pi\left[\left(\frac{\sin\beta}{\beta}\right)^2 - \frac{1}{2} - \frac{\sin 2 * \beta}{4 * \beta}\right] = 2 * \pi\left[\left(\frac{\sin 2.094}{2.094}\right)^2 - \frac{1}{2} - \frac{\sin 2 * 2.094}{4 * 2.094}\right] = -1.417$$

$$K_6 = \frac{-0.0641 - 0.1915 - 0.2283 + 0.4091}{-1.417} = 0.0528$$

$$K_7 = \frac{0.0528}{4} = 0.132$$

$$\sigma_7 = \frac{-83750\ \text{N}}{4 * 8\ \text{mm} * 305\ \text{mm}} - \frac{12 * 0.0132 * 83750\ \text{N} * 1216\ \text{mm}}{3130\ \text{mm} * 8^2\text{mm}^2} = -89.11\ \text{MPa}$$

≡ Acceptance Criteria

» The absolute value of σ_6, as applicable, shall not exceed f_d

$$\sigma_6 = 2.53\ \text{MPa} < f_d = 197\ \text{MPa}$$

» The absolute value of σ_7, as applicable, shall not exceed $1.25 * f_d$

$$\sigma_7 = -89.11\ \text{MPa} < 1.25 * f_d = 1.25 * 197\ \text{MPa} = 246.25\ \text{MPa}$$

2.3. Stress analysis withAutodesk Inventor Professional 2012 software

With the given data we made a 3D model of the tank and tested it with water or pressure (Figure 6). For the educational purpose we had simulations for the shell and the saddle as well, but here we introduce only the saddle part. It is obvious that the shell has higher stress at the touching areas of saddle (Figure 6. left). But the light blue colour does not sign extreme stresses, only tolerable stress-rising comes into existence. The peak stress occurs in the saddle (yellow area), but it can be solved by the construction of the saddle.

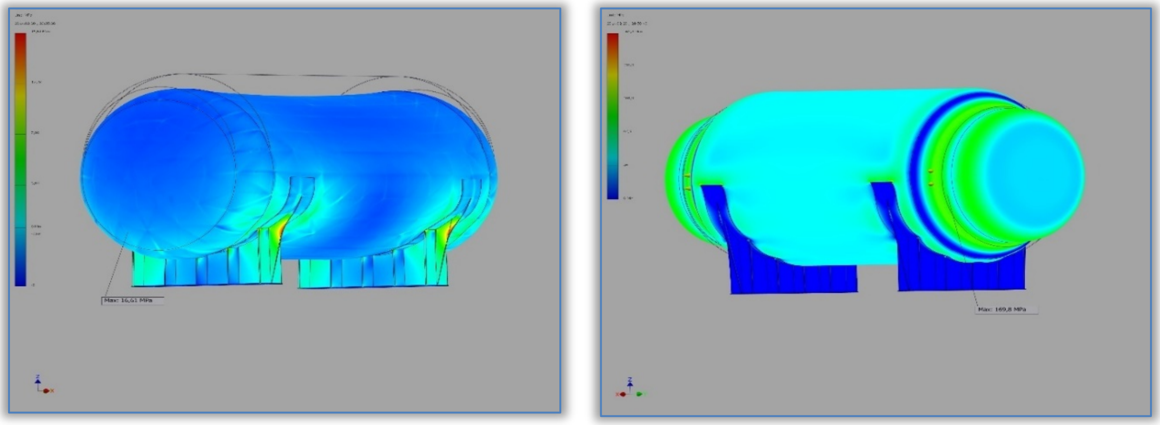

Figure 6.a, b. Stress analysis with water without pressure (a. left) and pressure without gravity (b. right)

3. CONCLUSION

In the analysis of subsidiary bending stresses of the support of horizontal retentional tanks we applied the calculation processes of the EN 13445:2009, and the ASME VIII-2 standard for a specific example. With this example our educational purpose was to introduce standard design methods to our students. The other purpose of this example is to familiarize our students the final element method. The final element calculations are meticulous, responsible tasks. The introduced models are only approaching to be able to qualify the tank's stress conditions. But even by these models can be shown the shell stresses coming from inner overpressure (6.b.) and the subsidiary stress distribution (6.a.). Our further developing goals are to purchase and enter higher performance software and hardware into education.

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