# Enclosure Design for an Automated Vacuum Application


*Author: Mario R. Flores Milán*

*Advisor: Dr. Julio Noriega Motta*

*Mechanical Engineering*


## Abstract


The plastic injection industries make use of the Deflashing Machine, which requires to recycle the filter that collects the plastic shaving dust, down to 0.2um. A series of automated subsystems, located in portable frame table, was already designed with the purpose to remove the dust from the filter using jet air nozzles and collecting it with a dust collector. The only system that was left to design, was an enclosed system with capacity to isolate the plastic dust from the atmosphere and to hold a given external pressure load of 15 Psi. caused from vacuum pressure. A design concept was made, then optimized with the use of plates theory, comparing stress and deflection results with computer aided engineering and design (CAE/D). The results of the finished design provided an enclosed system with 1.5 safety factor, 88% deflection decrease from the conceptual design and was only consume 19.4% of the general budget ($21,000).


## Problem Statement

The plastic injection molding industry makes use of the Deflashing Machine, which has a primary function of removing the fine dust and plastic shavings of plastic mold parts. The machine collects the dust by using a cylindrical filter, that have the potential to be recycled to minimize operational cost. The main goal is to facilitate a more cost-effective and unharmful way to recycle the filter, by means of cleaning it without exposing the workforce and the atmosphere to harmful dust particles. A mechanical design, comprised of custom sheet metal enclosure, will be needed to allocate the cylindrical filter. The enclosure will be placed in a predetermined station, that has connection ports to work alongside other subsystems. In this case, a provided stationary jet air manifold and dust collector manifold system to remove and collect the dust from the filter, respectively. The problem arises when the suction manifold is subjecting the enclosure to an external pressure of 15 Psi. The cabin will not only trap the particle dust, but also be able to withstand the external pressure to avoid deformation. The enclosure alongside the other existing subsystems will provide a practical approach that results in a cost-efficient and eco-friendly solution.

### Research Objectives

- Develop a manufacturable system, able to maintain below the budget of $4,750 and a factor of safety (FOS) around 1.5.
- Design an enclosed system to protect the workforce and atmosphere from 0.2µm fine dust particles.
- Design a structural reliable enclosure, able to withstand 15 Psi. of external static pressure.

## Background Information

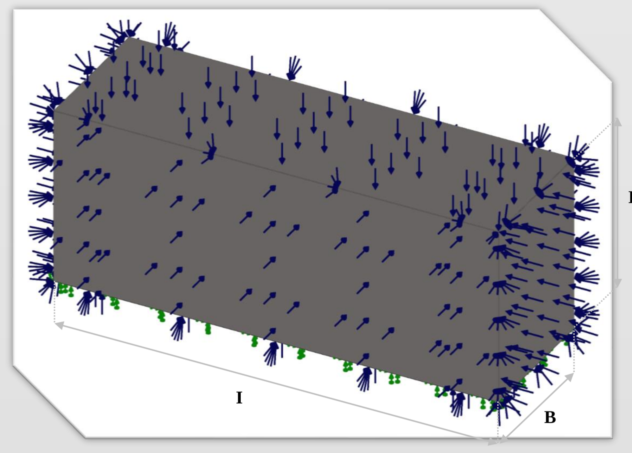
A rectangular box concept dimensions of 18.00”, 56.278” and 22.19” are the initial information of the space constraints. Selecting this type of geometry will cause mechanical stresses in the abrupt changes, such as the edges due to vacuum pressure. In conjunction with the stress, there will be unwanted deflection in the center of the plates. Questions will need to be answered such as:

1. What are the maximum stresses (in Psi.) for each face of the enclosure?
2. Which stress (in Psi.) is critical where is it located?
3. What will be the maximum deflection (in inches) for each face of the enclosure?

## Design Methodology

### Information Breakdown

A simple model of the enclosure system shows the uniform distributed load caused from the external pressure due to the inside vacuum.

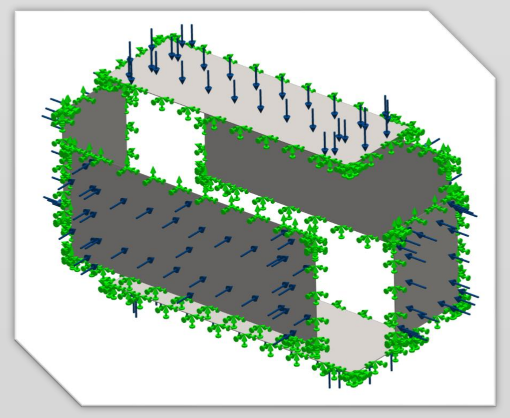


Study Model Concept

1. Shape of the Vacuum Chamber – Rectangular Box
2. Design Pressure (**q**) = 15 Psi. (Uniform Distributed Load)
3. Sheet Metal Thickness (**h**) = 7/64 inches (Gauge12)
4. Material – Stainless Steel 304 (**E** = 29000 Ksi., **µ** = 0.3)
5. Length (**l**) = 56.278 in
6. Width (**B**) = 18.00 in
7. Height (**H**) = 22.19 in

### Solution Strategy

The exploded view helps demonstrate the conditions for each face of the enclosure. The green arrows represent that all edges for each face are fixed. The uniform transverse distributed load (**q**), is defined for each face by blue arrows.



Exploded View Model Concept

| Rectangular Fixed Plates Under Uniform Load | | | | |
|---|---|---|---|---|
| a/b | | 1.5 to ∞ | | |
| Coefficients | | $\frac{w}{h}$ | $\frac{\sigma_o b^2}{Eh^2}$ | $\frac{\sigma_{max} b^2}{Eh^2}$ |
| $\frac{qb^4}{Eh^4}$ | 0 | 0.0000 | 0.0000 | 0.0000 |
| | 12.5 | 0.2800 | 0.2000 | 5.7500 |
| | 25 | 0.5100 | 0.6600 | 11.1200 |
| | 50 | 0.8250 | 1.9000 | 20.3000 |
| | 75 | 1.0700 | 3.2000 | 27.8000 |
| | 100 | 1.2400 | 4.3500 | 35.0000 |
| | 125 | 1.4000 | 5.4000 | 41.0000 |
| | 150 | 1.5000 | 6.5000 | 47.0000 |
| | 175 | 1.6300 | 7.5000 | 52.5000 |
| | 200 | 1.7200 | 8.5000 | 57.6000 |
| | 250 | 1.8600 | 10.3000 | 67.0000 |

The values of span-width ratio will define, for each plate, the numerical values from the Roark’s large deflection table. These preliminary calculations for the coefficients values only work for the case of a fixed configuration and for span-width ratio of 1.5 to ∞. For analytical purposes, the side plates will be analyzed within this range and extrapolation is made for values of pressure ratios.

**For: Top – Bottom and Side Plates**

For $K_1 = \frac{qb^4}{Eh^4} = 379.0654$ and $a/b = 3.127$ (TB) , $a/b = 1.233$ (SP)

$$K_2 = \frac{w_{max}}{h} = 0.0658*(379.065)^{0.625} = 2.691$$

$$K_3 = \frac{\sigma_o b^2}{Eh^2} = 0.0434*(379.065) - 0.1933 = 16.258$$

$$K_4 = \frac{\sigma_{max} b^2}{Eh^2} = 0.7926*(379.065)^{0.8143} = 99.733$$

**For: Front – Back Plates**

For $K_1 = \frac{qb^4}{Eh^4} = 875.494$ and $a/b = 2.53$

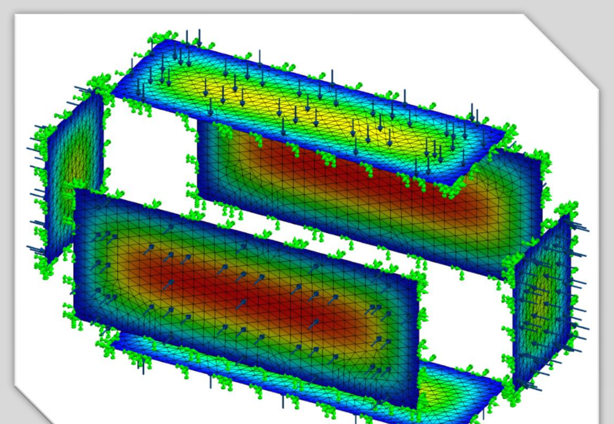
$$K_2 = \frac{w_{max}}{h} = 0.0658*(875.494)^{0.625} = 4.541$$

$$K_3 = \frac{\sigma_o b^2}{Eh^2} = 0.0434*(875.494) - 0.1933 = 37.803$$

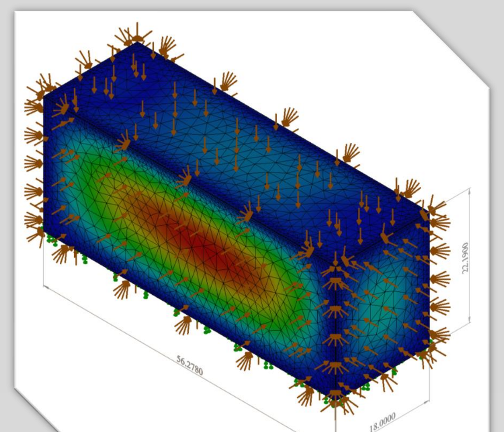
$$K_4 = \frac{\sigma_{max} b^2}{Eh^2} = 0.7926*(875.494)^{0.8143} = 194.358$$

## Results & Optimization

The stresses and deflection for each plate were analytically solved and validated with the finite element analysis tool from Solid Works.



FEA of Insulated plates model



FEA of Uninsulated plates model

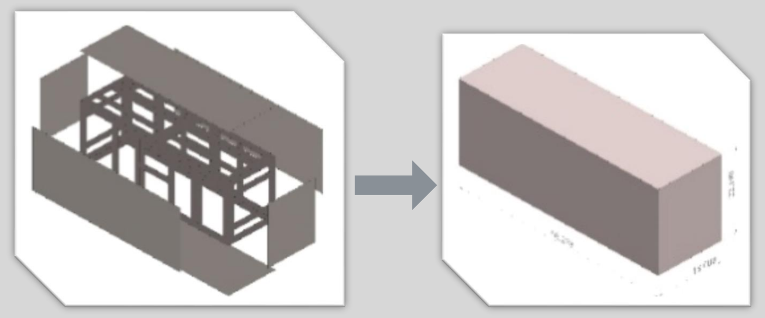
## Results & Optimization

The results that are obtained from each plate is also compared to the uninsulated plate study model. From the results, an optimization for the conceptual design of the enclosure system is made by establishing a factor of safety (FOS) of 1.5, by considering manufacturability and cost-effectiveness.

| Global Results Overview | | | |
|---|---|---|---|
| Stress at Center - $\sigma_o$ (Psi) | | | |
| Method | Analytical Insulated Plates | FEA Insulated Plates | FEA Uninsulated Plates |
| Top and Bottom | 1.742E+04 | 2.525E+04 | 1.843E+04 |
| Sides | 1.742E+04 | 2.296E+04 | 1.843E+04 |
| Front and Back | 2.665E+04 | 3.635E+04 | 3.675E+04 |
| Maximum Stress - $\sigma_{max}$ (Psi) | | | |
| Method | Analytical Insulated Plates | FEA Insulated Plates | FEA Uninsulated Plates |
| Top and Bottom | 1.07E+05 | 6.018E+04 | 7.338E+04 |
| Sides | 1.07E+05 | 5.510E+04 | 5.506E+04 |
| Front and Back | 1.37E+05 | 6.526E+04 | 1.466E+05 |
| Maximum Deflection - $w_{max}$ | | | |
| Method | Analytical Insulated Plates | FEA Insulated Plates | FEA Uninsulated Plates |
| Top and Bottom | 0.295 | 0.254 | 0.118 |
| Sides | 0.295 | 0.233 | 0.157 |
| Front and Back | 0.497 | 0.348 | 0.472 |

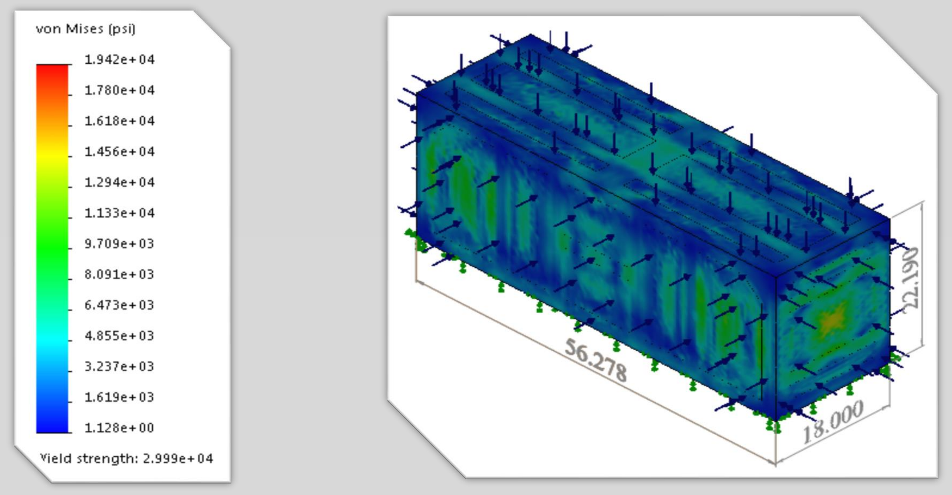
Both studies present a range of stresses that exceed the yield strength of the material (**Y** = 29.99 Ksi.). For example, for both studies, the maximum stress at the edges of the system falls between 6.526E+04 Psi. to 1.466E+05 Psi. Interestingly, it shows comparable results on the maximum deflection for each face of the enclosure, especially for the front and back plates. For instance, the maximum deflection of the uninsulated plate study model is 0.472” and is located at the front and back faces, while the insulated front and back plates model is 0.497”. The uninsulated model alone does not satisfy the objectives, a better reinforcement is required.

The proceeding optimization will consider viable cost-effective design techniques for available manufacture procedures, by maintaining a minimum factor of safety of 1.5. Also, it will take into account that the top plate and the frontal plate needs to be joined perpendicularly, to work as the access door to allow to take the filter in and out of the enclosure.



Reinforced Model

Once the model is defined, a finite element analysis is made, as shown Figure 11, where it shows the stress distribution. The stresses presented in Figure 11 are below the material yield strength (2.999E+04 Psi.), with a maximum stress of 1.942E+04 Psi. This time, the maximum deflection in the system is 0.058”, almost half of the proposed of the material gauge

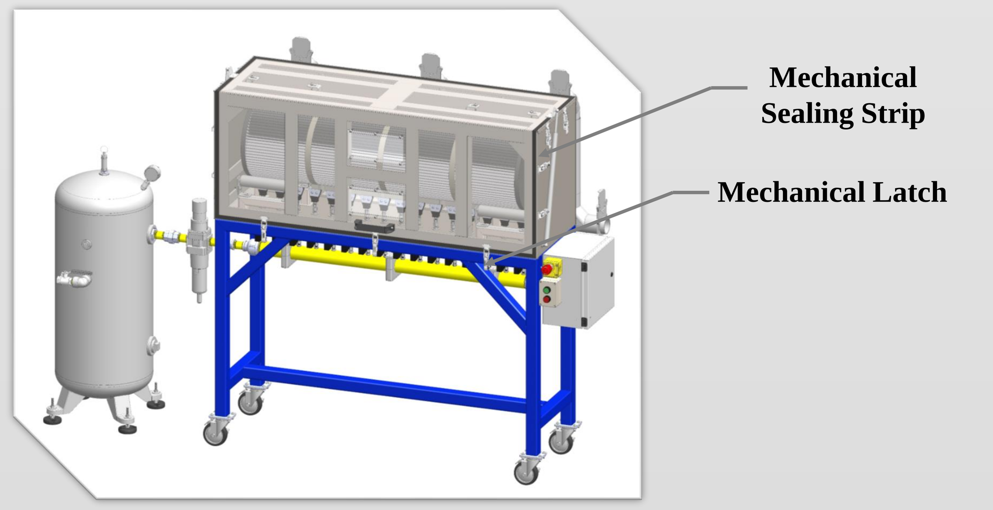


FEA of Reinforced Model

To obtain the exact minimum factor of safety, the yield strength of 2.999E+04 Psi. will be divided by the maximum stress value of 1.942E+04 Psi. which is used as the allowable stress in the proceeding calculation

$$FOS = \frac{Y}{\sigma_{allow}} = \frac{2.999E+04}{1.942E+04} = 1.54$$

The reinforced enclosure system satisfied all the proposed objectives and was successfully integrated with the other defined subsystems. It was also added a 1/16” recommended space between the access door and the main base for an EPDM mechanical black sealing strip. To ensure a proper mechanical sealing, a series of adjustable grip draw latch mechanism is built in, where it provides a 660 Lbs. holding capacity for each latch. The addition of these latching mechanism for the access door, provides more compatibility to the fixed edges plates studies.




Enclosure System Integration

## Conclusions

Both methods provided similar behaviors and served as the backbone of the optimization phase. Initially, the enclosure design without the reinforcement was having a maximum deflection of 0.497” and a maximum stress of 1.466E+05 Psi. However, with the same material (S.S.304), the reinforced enclosure with angular extrusion framing has a maximum deflection of 0.058”, which represent a decrease in deflection by 88.32%. As for the stress, the reinforced system was experiencing a maximum stress value of 1.942E+04 Psi., which is below the yield strength (2.999E+04 Psi.) and provides a 1.54 minimum factor of safety (FOS). Additional to the compliance of the safety factor and the design of an enclosed system, the new enclosure provides viable manufacturing and cost efficiency. Thus, fabrication cost increased from $1,784 to $4,080 but was able to maintain it below of the $4,750 budget. The found relation between the mechanics of pates and a rectangular box, for the given assumptions and conditions, presents potential future research work.

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