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CONTROLLED IMPACT ABSORPTION IN ADAPTIVE PRESSURIZED STRUCTURES.

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Abstract. *The subject of this paper is a new concept of controlling impact absorption by filling a thin-walled structure with compressed air and its controlled release during the impact process. The pressure and its change in time can be adjusted according to the type and direction of loading. The problem was modeled in ANSYS environment were static and dynamic analysis was performed. It was proven that pressure significantly increases load capacity of this structure. The results for different values of pressure are presented. Profit that can be achieved is up to 5 times higher in comparison to the initial conditions. The optimal value of the pressure inside the structure can be found from conditions concerning maximization of dissipating energy or the minimization of stresses. Experimental static test performed on a thin-walled can has been treated as a feasibility study. It shows satisfying accuracy with the numerical results. We also present an idea of dividing the structure into several packages separated by flexible walls to obtain even better results. The pressure in each package can be controlled separately and the flow of the compressed air between cells can be controlled in time. The application of this concept to increase durability of car doors and to optimize impact energy dissipation is described.*

1 INTRODUCTION

Thin-walled structures are the effect of optimization of structural elements. They are commonly used in car and mechanical industry as well as in civil engineering due to its huge durability, stiffness and small weight. However shell thin-walled structures are not adapted to sustain sudden lateral impacts since they have small ability to absorb corresponding energy. Significant increase of durability to dynamic loads can be achieved by applying compressed air and its controlled release during the impact process. As a result we obtain a structure which is possible to adapt to variable load cases or impact types. Further such structures will be called Adaptive Pressurized Structures (APS) (cf. Ref.7).

Filling with gas should take place immediately after the impact or even just before the impact (e.g. applying a radar system monitoring obstacles in traffic movement). The pressure must be adjusted to the velocity, mass and shape of the hitting object. Fast reacting so-called *micro-pyro-systems* can be used for immediate gas pumping. Their functioning is based on the conception of micro explosions similarly as in the air bag in the car. The purpose of this first phase is to increase the load capacity and stiffness of the structure. Pressure of the gas acting outwards prevents huge deformation caused by energy of the hitting object.

During the process of immersing the hitting object into the structure a decrease of the pressure is planned by opening the exit valves (e.g. piezo-valves). The purpose of this is to control the dissipation process and to brake the penetrating object on the appropriate way. The second aim is preventing of too huge accelerations which can be harmful for the structural elements.

It is demonstrated below that APSs behave better than standard structures under static load. Collapse of the thin-walled structures is due to buckling of the walls before the highest admissible stresses are achieved. Loosing of the stability occurs at the areas where compressive forces are high. Additional loading caused by the pressure acting outwards profitably influences the distribution of stresses in the shell, because it reduces compressive forces. The reduction of the bending moment is also possible at the structure with thicker walls. The pressure is adjusted according to the point of acting, direction and intensity of the load, results which we want to reduce. Further improvement of the APS can be achieved by dividing the structure into several packages separated by flexible walls. In a dynamic process the flow of the gas between the packages and outside the structure should be possible.

2 MODELLING OF PRESSURIZED STRUCTURE – STATIC CASE

To confirm predictions given above the Finite Element Method analysis using the ANSYS environment was performed. A thin-walled cylindrical shell resembling a beverage can was modeled and eight-node shell elements were used. The length of the structure was 170mm, radius of the base 33mm and thickness of the wall 0.1mm. It was assumed that the cylinder is made of aluminum having admissible tensile stress of 500 MPa. Young modulus was assumed to be equal to 56 GPa to obtain compatibility with the experiment described later on.

The structure was loaded as a cantilever: one end was clamped and a load perpendicular to the axis of the cylinder directed upwards was applied on the other. The load was distributed along the circumference of the circle. In the static case the results which can be achieved assuming no bending moment theory of the shells are satisfying. Let us calculate stresses at the top and bottom of the cylinder:

1. First load case: force acting at the end of the cantilever F=165 N

$$\sigma_{bottom} = -\sigma_{top} = -\frac{M}{W} = \frac{Fl}{\frac{\pi}{4}(r^4 - (r-t)^4)} = 82{,}4MPa \quad (1)$$

2. Second load case: force at the end of the cantilever F=165N and pressure inside p=0,4 MPa

$$\sigma_{top} = \frac{M}{W} + \frac{pr}{2t} = -82{,}4 + 66 = -16{,}4MPa \quad (2)$$

$$\sigma_{bottom} = -\frac{M}{W} + \frac{pr}{2t} = 82{,}4 + 66 = 148{,}5MPa$$

By applying pressure inside the structure we reduced precarious compressive stresses. The load capacity of the structure is dependent only on admissible tension stresses.

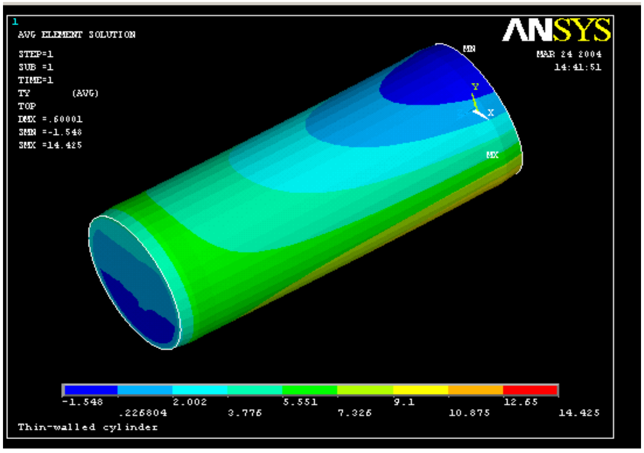

Fig.1 Forces along the cantilever loaded by force and internal pressure

The next step was linear buckling analysis (calculating eigenvalues and eigenvectors of the reduced stiffness matrix). The first shape of deformation is shown on Fig. 2. Longitudinal waves arise on both sides of the cylinder. The fact that there is no deformation in the area where the largest compression occurs (at the top of the shell close to the support) is quite interesting. The value of the critical force was equal to 165N, while exceeding of admissible stresses occurs when the force is above 1000N. The following values of the critical forces are close to each other. In the real structure the form of the stability loss can be a combination of several initial buckling shapes.

This numerical analysis was also used to estimate the value of the Young modulus for aluminum foil. We do not know the exact value of this modulus for aluminum foil since there are serious problems with conducting such an experiment in the laboratory. We obtain good agreement with the experiment when we assume Young modulus to be 20% percent lower than typical value from codes equal to 70 GPa.

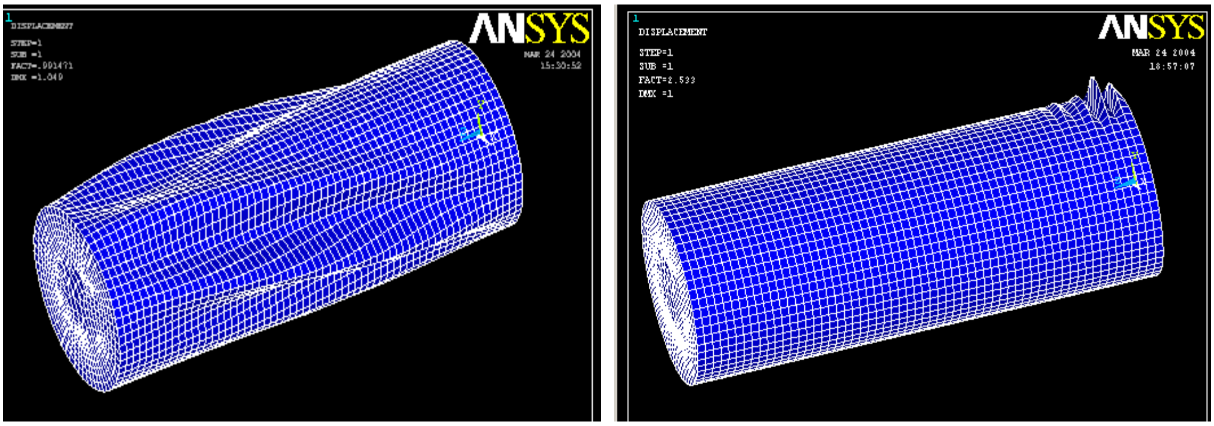

Fig.2 First buckling shape of the cylinder with pressure: a) 0 MPa, b) 0,35 MPa

The cylinder was subjected to additional distributed loading modeling gas pressure acting in the outside direction perpendicular to the inner walls. The values of two forces were observed in terms of increasing pressure. The first was the buckling force and the second was the force causing bursting of the bottom cylinder wall by exceeding maximal tensile stresses. The results of this analysis within the range 0 - 1,3 MPa are presented on the plot (Fig. 3). At the point where curves intersect the gas has optimal pressure equal to 0,975 MPa. Force of the greatest value 678 kN can be applied to the structure and it is 4,1 times larger than the initial force causing collapse. Buckling of the top wall and bursting of the bottom wall of the cylinder occures simultaneously.

When inside pressure was applied the area of buckling was gradually decreasing. Collapse area was moving to the top of the cylinder and in the direction of support (compare Fig. 2b). In this case buckling is not as dangerous as previously. The total collapse of the structure occurs as the result of bursting. According to this numerical analysis the cylinder which is not subjected to vertical load will be destroyed under the pressure of 1,5 MPa because of the high level of stresses in the circumferencial direction.

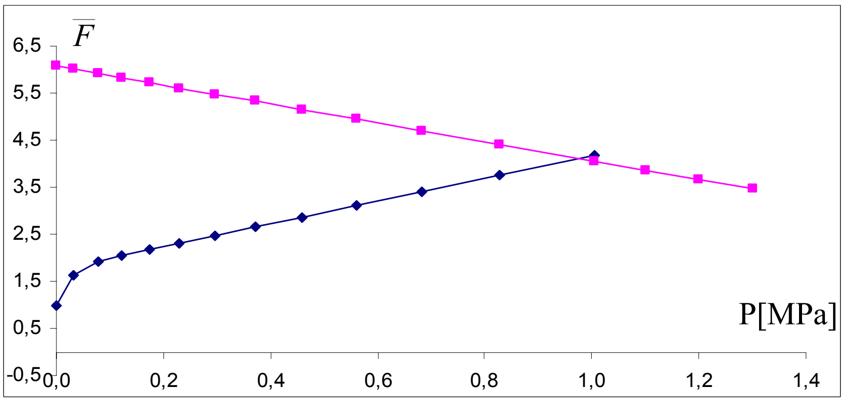

Fig. 3 Plot of first non-dimensional buckling force (increasing function) and non-dimensional bursting force (decreasing function) in terms of pressure of the gas. Both forces are divided by initial buckling force (165N).

Geometrically nonlinear static analysis which was also performed confirmed that our solution is estimated correctly and that we achieved a proper form of stability loss. This type of analysis gives us a chance to model the behavior of cylinder more precisely by taking imperfections into consideration. These imperfections are the differences in the thickness of aluminum foil and small dents. When we do not know the exact form of these imperfections we can assume that they are linear combinations of several initial buckling forms.

3 EXPERIMENTAL VERIFICATION

Numerical analysis was confirmed by experiment. A static test was carried out on a thin-walled beverage can which was subjected to vertical load. The dimensions and material properties of the can had the same values as in the model from numerical analysis. Static and kinematic boundary conditions were also the same as previously. At the beginning the empty can was loaded only by the force at the end. The collapse of the structure was due to the buckling of the sidewalls of the cylinder as it was predicted. The value of the critical force was equal to 155 kN so it was nearly the same as from the numerical considerations. The buckling occurred in the large area between the middle of the can and the support at the top and was symmetrical on both sides (compare Fig. 4). The deformation is a little different than the theoretical buckling shape from FEM, however the direction of the weaving remained. The reason for this fact could be the omitting of imperfections and plasticity in the numerical analysis.

Fig. 4 Deformation of the beverage can loaded by pressure: a) 0 MPa, b) 0,35 MPa

Then the cylinder was sealed and filled with compressed gas from the compressor. The pressure was equal to 0.2 MPa, 0.4 MPa, 0.6 MPa and 0.8 MPa. Such loading acts against buckling because it reduces compressive stresses and does not allow the walls of the cylinder to deform inwards. While the cylinder was filled with pressure the area of buckling was decreased and moved into the support. This can be observed on Fig. 4b.

Load capacity of the structure increased gradually and this favorable change was even faster than in the numerical calculations. When pressure had maximal value of 0.8 MPa it was possible for our structure to carry the load of 700N. In each case the change of force in terms of displacement was observed. The area under the plot (equal to work done by the force on displacement) increased when the pressure was higher. This confirms the property that APSs can absorb larger amounts of energy than classical structures.

Total destruction of the can filled with maximal pressure was caused by a sudden burst since the admissible tensile stress was exceeded. This is one of the most important problems connected with applying APSs. Unexpected and sudden destruction (explosion) can be very dangerous for the other parts of the structures or for the people finding themselves nearby. The best method of preventing the effect of bursting is letting the air out from structure when the high pressure is no longer necessary.

4 TESTING DYNAMIC RESPONSES

After verification using the static analysis we begin modeling a more complicated dynamic case. Four types of dynamic analysis were performed:

1. Modal analysis – calculating natural frequencies and modal shapes of the vibrations
2. Transient analysis – applying load for short period of time
3. Slow transient analysis – applying different masses to the end of the structure with constant relatively small velocity
4. Fast transient analysis – applying constant mass to the end of the structure with different velocities

The first step of the dynamic analysis was calculating several initial natural frequencies of the cylinder without filling it with gas. When structure is subjected to pressure then tension or compression internal forces arise. These forces enters geometrical stiffness matrix which is included in the differential equation describing vibrations. In this manner tension forces increase frequencies of vibrations and compressive forces decreases them. This effect is significant even when the gas inside the cylinder is under low pressure (exact results are shown in table 1). The change of the modal shape also occurs. One of such modes is shown on Fig.5. Taking this fact into account the APS is a structure in which active and fast controlling of dynamical properties can be applied.

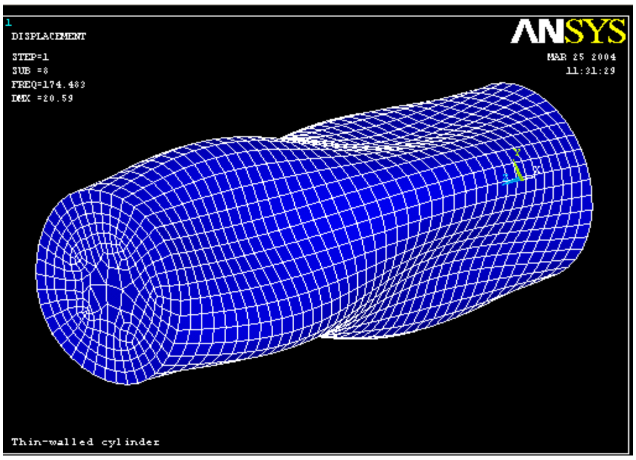

Fig. 5 Fifth modal shape of cylinder filled with gas under pressure p=0.9 MPa

	p=0	p=0.3	p=0.6	p=0.9
I	22.12	54.05	54.90	55.73
II	25.29	82.78	92.83	101.86
III	25.53	85.62	114.71	137.75
IV	32.67	111.96	139.04	142.33

Table.1 Four initial natural frequencies of vibrations [Hz] corresponding to different pressures [MPa]

In the first type of transient analysis the force equal to 700N (arbitrary chosen value) was applied to the end of the empty cylinder at a time of 0.001s. The change of displacements in a period of time equal to 0.1s is shown in Fig 6a. The response of the structure in the case when it is filled with gas under pressure 1 MPa (that is the biggest value that does not cause destruction of the cylinder) seemed to be quite similar. If applying higher pressure were possible, we would be able to control the frequency of vibrations and values of the amplitude (displacements). As an example we show analysis performed in the case of load intensity equal to 7 MPa.

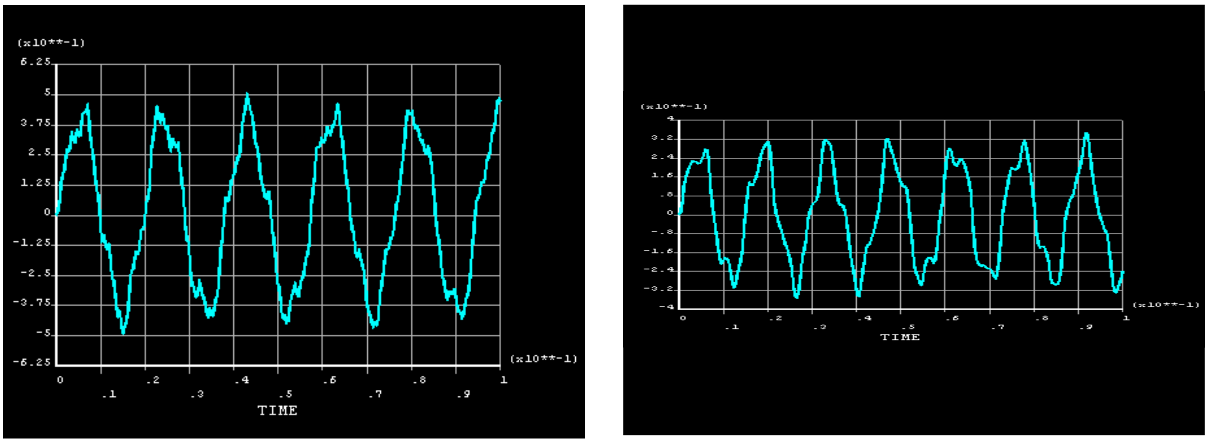

Fig. 6 Response of the cylinder for short force load at the end. Pressures inside structure equal to 0 and 7 MPa.

The response of the structure in case of dynamic load caused by mass hitting with initial velocity is similar to shown above. Applying initial pressure to the cylinder does not cause remarkable changes in values of displacements and frequencies.

5 FUTURE STEPS

At this point we will present further concepts of improving properties of the structure (increasing load capacity under static and dynamic loading) and their application to the construction of the car door. Additional advantages can be achieved by the appropriate choice of pressure in different parts of the structure because they work in a different state of stresses. The structure should be divided into several packages and elastic thin membrane divisions should be used for this purpose. Piezoelectric valves may be placed in every membrane to allow the flow controlling.

The application of this concept will be shown on an example of a plane frame loaded in the middle of the span modeling the door of the car in a simplified manner. Pressure of the air will be used to reduce values of bending moments on the top and bottom beam. We will use the horizontal division as the most effective in this case. In the initial situation shown on Fig.7 bending moments are equal to M_1=18,54 kNm on the top and M_2=4,90 kNm on the bottom. When using constant pressure of 138,6 kN/m in the whole structure we can make moments on top and bottom equal to each other. This way we can increase load capacity by 58,4%.

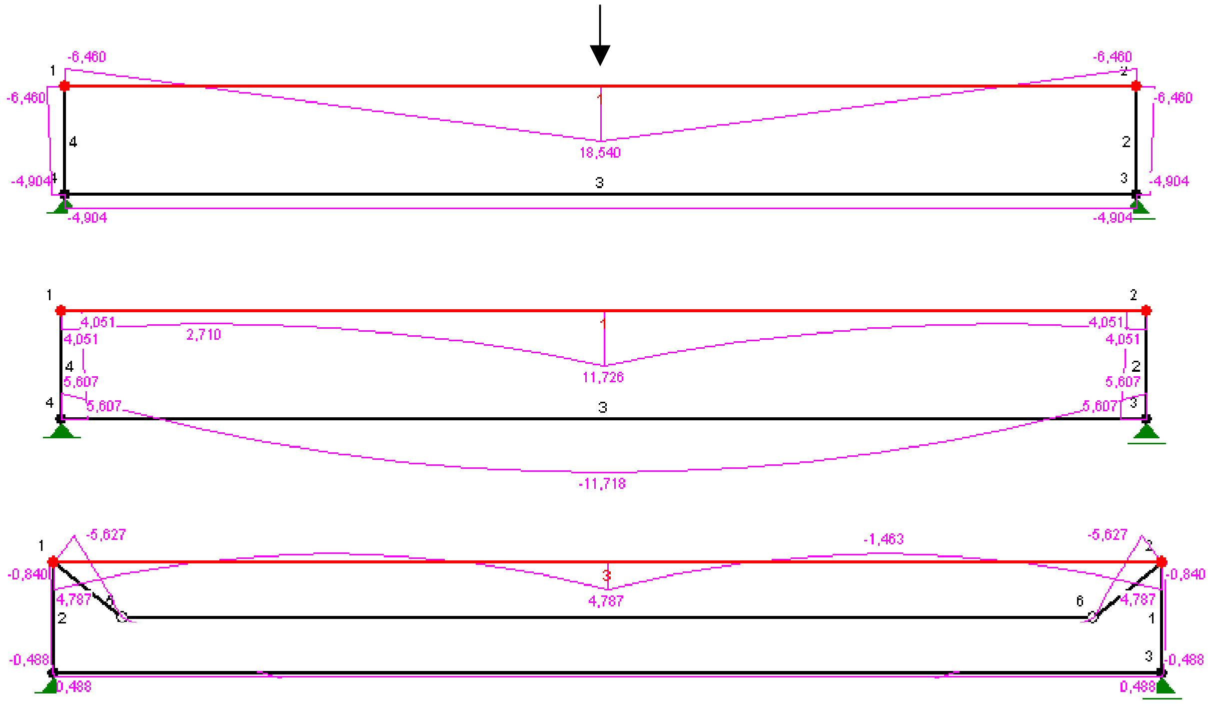

Fig 7. Reduction of bending moment in the frame. Load capacity factor: a)1 b)1,58 c)3,87

Dividing the frame by horizontal cables does not give a good effect while the difference between bending moments at the top and bottom beam is not significant. We can not reduce the moment on the top cause we will exceed the maximal value on the bottom.

The concept to divide structure horizontally and to put pressure in the top package is much more efficient. We assume that the division is made by cables which do not have stiffness on bending so we can omit the value of the moment at the internal rod. We assume that the maximal moments which occur at the top beam in the middle of the frame and near the internal columns must be equal. The increase of the load capacity depends on the length of the pressurized package. Load capacity factor is the highest (equal to 3,86) when the package is supported as on picture 7c.

Following the concept described above, it is planned to control changes of pressure (during dynamic process) in each package of APS separately (cf.Fig.8). The flow of the gas should be possible between the cells and also outside the structure. In the general problem we have to find the optimal distribution of the packages and functions $p_n(t)$ where n is the number of pressurized packages. The objective function in the dynamic case could be defined as the maximization of dissipating energy, minimization of displacements or minimization of accelerations acting on the structure. Numerical results will be presented at the conference.

The pressurized structure formulated above could serve as a basis for a car door design. Applying APS will be especially useful during the side impact with another car, rigid obstacle or pedestrian. Every mentioned objective function is significant for the safety of passengers. In the case of an accident involving a pedestrian it is necessary to ensure low stiffness to decrease the force acting on such a person. Concept of applying Adaptive Pressurized Structure to controlled impact absorption is patented (Ref. 7).

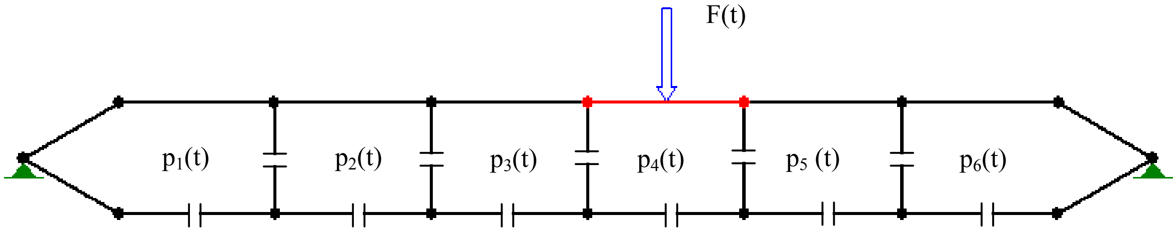

Fig. 8. Concept of dividing the structure into several separately controlled pressurized packages

6 CONCLUSIONS

- The paper presents the concept of new Adaptive Pressurized Structures and shows possibilities and methods of controlling it
- A numerical solution of the static problem of thin-walled shell structure shows the beneficial influence of the pressure on the distribution of the internal forces and the increasing the value of the buckling force
- Numerical results are confirmed by experiment with good accuracy
- The solution of the dynamic problem shows that gas under pressure can be used to control properties and response of the structure
- The concept of dividing the structure into several pressurized packages is presented together with its application to decrease the values of bending moments in the frame
- This paper can be treated as a preliminary study to show numerical tools that can be used to model and control impact absorption in APS
- Possibilities of practical applications are also shown in the paper

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