Experimental Procedures and Analysis
4.1 OVERVIEW

The details of various mechanical characterization, buckling and vibration tests performed on composite samples under study are described in this chapter. The analysis of composite samples by finite element method are also presented in this part of thesis.

4.2 MECHANICAL CHARACTERISATION

4.2.1 Density

The theoretical density ($\rho_{ct}$) of composite materials in terms of weight fraction of different constituents can easily be obtained as for the following equation given by Agarwal and Broutman [12].

$$\rho_{ct} = \frac{1}{(W_f/\rho_f)+(W_m/\rho_m)} \quad (4.1)$$

where, W and $\rho$ represent the weight fraction and density respectively. The suffixes f and m stand for the fiber and matrix respectively. Since the composites under this investigation consist of three components namely matrix, fiber and particulate filler, the expression for the density has been modified as

$$\rho_{ct} = \frac{1}{(W_f/\rho_f)+(W_m/\rho_m)+(W_p/\rho_p)} \quad (4.2)$$

However the actual density of composite can be found experimentally by simple water immersion technique or by density meter as shown in Fig. 4.1.
4.2.2 Tensile Strength

The tensile test is generally performed on flat dumbbell shaped specimen with dimensions specified in annexure as per ASTM D638. Present work involves use of universal testing machine at crosshead speed of 5mm/min to calculate tensile strength of composite plates. Five specimens from a given plate were taken and mean value was considered as tensile strength of that particular composite plate. The universal testing machine and loading arrangement are as shown in Fig. 4.2 a and Fig. 4.2 b.
4.2.3 Flexural Strength

Maximum tensile stress that a composite can resist during bending before reaching breaking point is called its flexural strength. Three point bend test was carried out on all composite plates to determine its flexural properties as per ASTM D790.

\[
\text{Flexural Strength} = \frac{6PL}{2bt^2} \quad (4.3)
\]

Where, 
- \(L\) is the span length of the sample (mm)
- \(P\) is maximum load (N)
- \(b\) is the width of specimen (mm)
- \(t\) is the thickness of specimen (mm)

Five specimens from a given plate were taken and mean value was considered as flexural strength of that particular composite plate. The universal testing machine and loading arrangement are as shown in Fig. 4.3 a and Fig. 4.3 b

![Fig. 4.3 (a) Universal testing machine (b) Loading arrangement for flexural test](image_url)
4.2.4 Compressive Strength

Compressive properties of composite plates are determined using Aimil (AIM 302, India) compressive testing machine of 100 tonne capacity as per ASTM D695. The compression testing machine and loading conditions are as shown in Fig. 4.4 (a) and Fig. 4.4 (b).

Fig. 4.4 (a) Compression testing machine (b) Loading arrangement for compressive test

4.2.5 Micro-hardness

Micro-hardness of composite plates was measured with Barcol micro-hardness tester as per ASTM D2583. Firstly the barcol impressor and material to be tested were kept on hard and flat surface. The point sleeve of impressor was set on the surface to be tested. Subsequently a uniform downward force was applied quickly by hand till dial indication reached the maximum. This maximum value on dial indicates micro-hardness of composite plates in barcol hardness units. The barcol tester is as shown in Fig. 4.5.
4.3 BUCKLING ANALYSIS

In plane compressive loads, when high enough, cause out-of-plane deflections that may be excessive and lead to failure. This is called buckling and the load at which excessive out-of-plane deflection occurs is called buckling load. Buckling analysis of the coir polyester composite plates was done using finite element analysis package ANSYS12.1 to find critical buckling load. The results of theoretical analysis were than verified by doing experimental buckling analysis on all composite plates.

4.3.1 Governing Equation For Buckling

Assume that plate is compressed in its middle plane by forces uniformly distributed along the sides x=0 and x=a. Let the magnitude of this compressive force per unit length of the edge be denoted by Nx. By gradually increasing Nx we arrive at the condition where the flat form of equilibrium of the compressed plate becomes instable and buckling occurs. The corresponding critical value of the compressive force can be found in this case by integration of equation,
\[
\frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} = \frac{1}{D} \left[ N_x \frac{\partial^2 w}{\partial x^2} + N_y \frac{\partial^2 w}{\partial y^2} + 2 N_{xy} \frac{\partial^2 w}{\partial x \partial y} \right]
\]

The same result is obtained also from a consideration of the system. The deflection surface of the buckled plate can be represented, in case of simply supported edges by the double series,

\[
w = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} a_{mn} \sin \left( \frac{m \pi x}{a} \right) \sin \left( \frac{n \pi y}{b} \right)
\]

The strain energy of bending, in this case,

\[
\Delta U = \frac{\pi^4 ab}{8} D \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} a_{mn}^2 \left( \frac{m^2}{a^2} + \frac{n^2}{b^2} \right)^2
\]

The work done by compressive forces during buckling of plate,

\[
\frac{1}{2} N_x \int_0^a \int_0^b \left( \frac{\partial \omega}{\partial x} \right)^2 dx dy = \frac{\pi^2 b}{8a} N_x \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} m^2 a_{mn}^2
\]

Thus, for determining the value of the compressive forces, becomes

\[
\frac{b \pi^2}{8a} N_x \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} m^2 a_{mn}^2 = \frac{ab \pi^2}{8} D \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} a_{mn}^2 \left( \frac{m^2}{a^2} + \frac{n^2}{b^2} \right)^2
\]

From which,
\[ N_x = \frac{\pi^2 a^2 D \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} a_{mn}^2 \left( \frac{m^2}{a^2} + \frac{n^2}{b^2} \right)^2}{\sum_{m=1}^{\infty} \sum_{n=1}^{\infty} m^2 a_{mn}^2} \]

By utilizing this equation, we can derive the formula for critical buckling load but it requires lots of calculation [142]. So present work involves use of finite element analysis to find critical buckling load of composite plates.

4.3.2 Finite Element Analysis

ANSYS 12.1 is used to analyse the critical buckling load of composite plates of same sizes and different material composition. The dimensions of the specimen were 50×250×10 mm in length, width and thickness. The plate is modelled using 20 nodes hexahedral element using coarse, medium and fine grid. Hex 20 is the most common element used for simple flat geometries. Fig. 4.6 shows a typical meshing. The applied boundary condition and load for linear buckling analysis is as shown in Fig. 4.7.

Fig. 4.6 Meshing of the composite plate
4.3.3 Experimental Buckling Analysis

Eight specimens of different compositions having dimension of 50×250×10 mm were used to determine the buckling loads of the specimens. The specimens were loaded in axial compression using Aimil (AIM 302, India) compressive testing machine of 100 tonne capacity as shown in Fig. 4.8. A dial gauge was mounted at the centre of the specimen to observe the lateral deflection. For axial loading, the test specimens were placed between the two extremely stiff machine heads, of which the lower one was fixed during the test, whereas the upper head was moved downwards by servo hydraulic cylinder. All plates were loaded at constant cross-head speed of 1 mm/min. As the load was increased the dial gauge needle started moving, and at the onset of buckling there was a sudden large movement of the needle. The load corresponding to this point is taken as the buckling load of the specimen. Test specimen after buckling is shown in Fig. 4.9. However, Load V/s Deflection graph has to be used to find experimental bucking load as shown by Yang et al. [143] & Mohtaram et al. [144].
4.4 VIBRATION ANALYSIS

Maximum damage to any structural component which is subjected to dynamic loading is due to resonant vibrations. Intensity of stress and its frequency are the deciding parameter which determines probability of material failure due to vibrations. So it becomes very much important to limit maximum amplitude of vibration for safety of structure. Hence vibration analysis which helps to predict the response of structure and controlling structural vibrations is becoming very popular nowadays.

Present investigation involves studying the vibrational behavior of coir polyester composite plates with and without red mud filler using ANSYS 12.1. The results obtained are validated using experimental method and a MATLAB program developed by Dr. Sulaymon L. Eshkabilov. Natural frequency of coir polyester composites are carried out by considering the plates as a cantilever beam. Boundary condition for the cantilever beam is one
end fix and another thee ends are free. Main reason for considering cantilever beam is length of the plate is limited so proper oscillation is not possible in other boundary condition and another thing is structural application of cantilever beam is also very important.

4.4.1 Free Vibrations

By free vibration we mean the motion of a structure without any dynamic equation external forces or support motion. The motion of the linear SDF systems without damping specializes to

$$m \frac{d^2 u}{dt^2} + ku = 0$$

Free vibration is initiated by disturbing the system from its static equilibrium position. By imparting the mass some displacement $u(0)$ and velocity $\dot{u}(0)$ at time zero, defined as the instant the motion is initiated:

$$u = u(0), \dot{u} = \ddot{u}(0)$$

So, solution to the equation is obtained by standard methods:

$$u(t) = u(0) \cos \omega_n + \frac{\ddot{u}(0)}{\omega_n} \sin \omega_n t$$

Where natural circular frequency of vibration in unit radians per second $= \omega_n = \sqrt{k/m}$

The time required for the un-damped system to complete one cycle of free vibration is the Natural period of vibration of the system.

$$T_n = \frac{2\pi}{\omega_n}$$

Natural cyclic frequency of vibration is denoted by $f_n = \frac{1}{T_n}$, unit in Hz (cycles per second).
4.4.2 Finite Element Analysis

Commercial finite element code ANSYS12.1 was used to study the vibration behavior of eight cantilever composite plates of 185×55×10 mm size but with different material composition. 20 node hexahedral elements with coarse grid were used for modeling the plate. Hex 20 is the most common element used for simple flat geometries. Fig. 4.10 shows typical meshing of plate for analysis. Fig. 4.11 shows applied boundary conditions and load for vibration analysis.

Fig. 4.10 Meshing of composite plate
Fig. 4.11 Loading and boundary conditions

The equations of equilibrium of a discretised elastic structure undergoing small deformations can be expressed as

\[
[M][\ddot{u}] + [c][\dot{u}] + [k][u] = \{F(t)\}
\]

For free un-damped vibration, the equation reduces to

\[
[M][\ddot{u}] + [k][u] = 0
\]

If modal co-ordinates are employed the equation becomes

\[
[[K] - \omega^2[M]]\{\phi_m\} = \{0\}
\]

There are various methods of finding the natural frequencies \(\omega_n\) and modal vectors \(\{\phi_n\}\) if once the system mass \([M]\) and stiffness matrices \([K]\) are formulated [145,146]. These matrices can be solved with the help of MATLAB. Programme for finding out theoretical natural frequency in MATLAB is given in Annexure 1. Available properties of coir fiber laminates like young’s modulus density and geometric properties are used for solving this programme.


4.4.3 Experimental Vibration Analysis

Experimental modal analysis has become important tool for researcher due to latest instrumentation and computer aided data acquisition system. In present work vibration Test was carried out in Multi-recorder frequency processing library TMR-211-01. The TMR-211 is the main unit for multi-recorder small multi-channel data acquisition system and controls various input/output units, supplies power and collects data. This unit enables measurement of 10 various input/output units up to 80 channels. In addition, by installing the frequency analysis library software TMR-211-01 option, real time frequency analyzing process is possible. The connections of Data acquisition system, computer, and accelerometer, modal hammer and cables to the system were carried out. The coir polyester composite plates were clamped at one end on table by using C clamp and free at other end in cantilever pattern. The dimensions of the specimens were 185×55×10 mm in length, width and thickness respectively. The accelerometer was mounted on free end of plate with the help of bees wax and it was connected to data acquisition system TMR 211 by Tokyo Sokki Kenkyujo Co. Ltd. Japan. Complete vibration test set up along with TMR 211 Analyzer is shown in Fig. 4.12 and Fig. 4.13 respectively.

![Fig. 4.12 Experimental vibration test setup](image)
For free vibrations response of the plates, the free end of plate was hit for a second with modal hammer. Signals from the accelerometer attached to the plates were received by the TMR-211 analyzer which in turn was fed to computer. When free oscillation is given to the plate the accelerometer attached to it converts into electric signal and gives it to multi recorder TMR211. Multi recorder gives output in form of frequency versus time. Graphical outputs of all eight coir polyester composite plates with and without red mud were obtained. The output of Multirecorder was than processed in MATLAB program to give magnitude v/s frequency diagram for various composite plates.