RELIABILITY ANALYSIS OF HYDRAULIC SYSTEMS EMPLOYED IN PRESSES: APPLICATION TO A DEEP DRAWING PRESS FOR 250 TON

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Abstract

This paper presents a reliability analysis methodology for hydraulic system used in presses aiming to define the critical components and the effects of their failures on the press performance. Reliability is a tool that can be used in the development and analysis of hydraulic systems, from the selection of components to the planning of efficient maintenance routines. In this paper, a reliability analysis methodology is proposed based on the knowledge of the hydraulic systems failure modes. In order to execute the reliability analysis, the presses are divided in three main systems, which are structural, drive and forming, including tools. Using the Failure Modes and Effects Analysis (FMEA), the consequences of the failure modes of each component of the drive system are classified according to their effects on the press performance. Based on the results of FMEA analysis, the wear of the main pump and main cylinder are considered the critical failure modes of the hydraulic drive system, once they can cause the most undesirable press functional failure, which is the absence of motion.
Based on parameter estimates and wear analysis models, the reliability of those components can be evaluated, and using the Fault Tree Analysis, the hydraulic drive system reliability analysis is performed.
The methodology proposed in this paper is applied for a 250 ton deep drawing press, in order to define the operational life of the hydraulic drive system.

Keywords

Reliability, hydraulic, presses.

1. Introduction.

The presses are the oldest industrial machines, which use hydraulics systems. In 1795, hydraulic presses used water as hydraulic fluid, replaced by water solutions and by mineral oil, in 1900, when these machines increased capacity and performance working at highest pressures, by 35 MPa, with fast response of power. Presses are used in mechanical industry to form materials and other operations with high levels of force combined with speed variations.

The technical standard DIN 8550 defines pressing like "a mechanical conformation operation where a shape is plastically modified in three dimensions, with controlled geometry and without mass or material losses" and the technical standard DIN 8584 classifies the deep drawing operation like "conformation under compression and traction conditions".

By these definitions, the press function is to transfer one or more forces or movements to a tool, or a mould, to conform a plate or a piece, (Schuler, 1998). Considering that the main task of a press is form one piece, there are three systems supporting this task and the press reliability: structural, tool and matrix, and hydraulic system. Considering that the press main function, the hydraulic systems, which is responsible for providing movement and force for forming, can be considered as a primary systems as for reliability analysis.

The deep drawing presses can be classified according to tool's movements driven by the hydraulic system such as combined movement presses, with different speeds of approximation and pressing, or single movement, that have only one moving speed. The primary system of a press is the drive system. In hydraulic presses this drive system can be divided in two subsystems: the hydraulic subsystem and the command subsystem with their necessary interfaces to enable the operation. It doesn't mean the subsystems have only one function, because they also have responsibility in the operation and operator safety, considered as secondary functions, according to the definition presented by the technical standard DIN 8550.

Hydraulic deep drawing presses have two typical hydraulic circuits, used by the great number of industries: the dual or high&low circuit and the circuit with auxiliary cylinders with filling valve (free fall). Both the circuits have technical solutions to provide speed variations during the tool's advance, but the circuit analyzed in this study is the dual circuit. This circuit supply two different flows rates to the press drive, with two pumps in parallel disposition. By the flexibility, the dual circuit can be applied in other types of presses, so the results of this reliability analysis can be extended to other machines.

2. Reliability of a Dual Hydraulic Circuit of a Press.

Reliability is associated with successful operation, and with the absence of breakdowns or failures and is defined as the probability that a system will perform its intended function for a specified period of time under a given set of conditions. System is used here in a generic sense so that definition is also applicable to all varieties of products, subsystems, equipment, components and parts (Lewis, 1996). A product or system is said to fail when it ceases to perform its intended function, however there are different failure levels for a same system, considering deterioration or instability of function.

The reliability analysis applied in hydraulic presses intends to evaluate the failure modes – and their occurrence probability – of each subsystem of a press and their consequences to the machine operational condition, mainly the incapability to perform its function in a specified period of time.

The reliability analysis for this industrial equipment, essentially concentrated at its hydraulic system can be used as design parameter and components selection criterion, seeking the determination of operational and maintenance conditions of the hydraulic system at the machine design stages. A complementary analysis based on the reliability evaluation of the other machine subsystems can support the design, production planning and maintenance of this equipment, affecting the design and fabrication costs estimate, usually reducing them.

Reliability uses systems analysis tools to determinate the component failure modes and their effects over the operational capability of the system itself. The most used tools are the reliability block diagrams, the Failure Modes and Effect Analysis (FMEA) and the Fault Trees methods. The method selection considers the advantages and disadvantages of each one in a system analysis. For the deep drawing presses hydraulic system reliability analysis are used the FMEA and Fault Trees methods, with quantitative and qualitative results.

To apply the reliability analysis techniques a specific failure mode should be defined, because hydraulic systems may have different failure modes when the components are separately analyzed. Wear and contamination are the usual failure modes of all hydraulic components. Wear is different for each hydraulic component and it's time progressive, causing cumulated losses until the complete deterioration. The contamination is a wear accelerator, also a random failure factor, depending on quantity or dimensions of contaminants present in the fluid. To estimate the longest operational period of a hydraulic system, the fatal failure occurrence, which means the system inability to transfer to the tool the required power to form one piece, should de analyzed.

3. Failure Modes Analysis of Hydraulic Systems Parts.

Each hydraulic component has a failure mode due to wear, but many components have similar operations and have identical parts of the same material submitted at the same loading, represented by the fluid pressure, such as valves steel springs, or nitrile rubber seals. The most loaded components of the dual hydraulic circuit are the high pressure pump, the relief valve, check valves, directional valve and the hydraulic actuator, which compose the primary system, submitted at the maximum operational pressure.

The hydraulic component that has the critical wear is the pump, due to the continuous work condition and pressure variations. Life determination of pumps almost gives the equipment durability because at normal conditions (pressure, temperature and contamination controlled) the pumps have worse wear conditions than the other hydraulic components in the same time period. The deep drawing presses hydraulic circuits use three kinds of pumps: gear, vanes and pistons. Each one has a different wear condition but the volumetric efficiency may represents the pump wear (Frith, 1996). The volumetric efficiency gives the pump internal leakage as a function of pressure, speed and internal clearances.

A pump at continuous operation has an increase in its clearances due to progressive wear that causes material for material loss and volumetric efficiency loss, so the efficiency drop may be used as pump wear indicator. The critical volumetric efficiency occurs when the pump isn't capable to keep the designed flow rate constant (Frith, 1996), so the pump must be replaced. At this work a vane pump applied in a hydraulic system of a deep drawing press will be analyzed taking in view the wear and the volumetric efficiency drop. A vane pump has a rotor driven by a shaft, with vanes that slide in the rotor fissures, keeping contact with a ring of elliptical shape (balanced pumps). The kit is laterally closed with two fixed plates, called pressure plates.

There are two wear areas in vane pumps: the vanes – ring contact and the plates – rotor contact. A brief analysis of the two wear possibilities shows that the most severe wear area is the vanes – ring contact caused by the intermittent variation of the contact pressure and the contact force magnitude. This is a combined wear, adhesive and abrasive, with an undesirable effect because the ring's material loss allows a flow rate from a high pressure chamber to a low pressure chamber decreasing the volumetric efficiency. Gellrich et al (1994) developed a mathematic model for the vane pumps wear with information about the wear system, which gives important contribution to the pumps wear determination. The model was compared with experiments and the most important conclusion refers to the removed mass calculation, wear progression in time and localized linear wear, as presented in the Figure 1 charts. At the Figure 1, the load zones I, II, III and IV are regions of the ring submitted at different contact forces. The model is based on the shear stress acting on the surface presenting movement, the load zones, where the contact force is constant. The charts of the Figure 1 indicate a comparison between the removed mass calculation in each the load zone and the real mass removed in the experiment at the zone II, where the contact force is maximum.

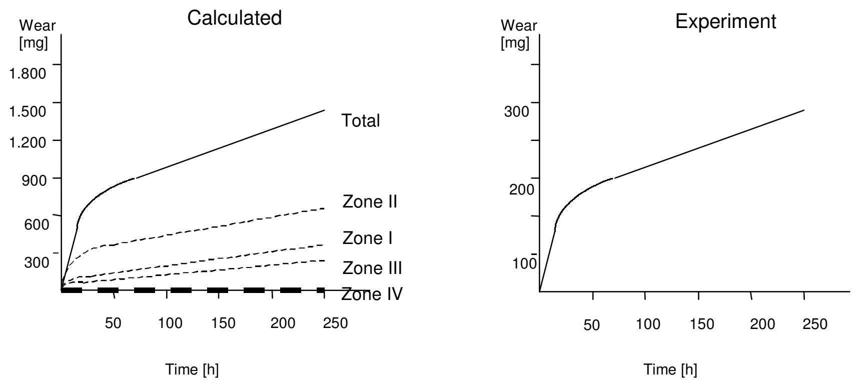

Figure 1. Comparison between Model and Experiment, Gellrich (1994).

The hydraulic cylinder presents another type of wear, the lubricated, in the metal parts in contact in the presence of seals, such as piston and tube, and rod and front bush. At the normal work conditions the cylinder's wear occurs mainly in the tube and rod seal contact.

For the valves applied in the dual hydraulic circuit the durability limit is the life of their springs. The main failure criteria for the springs design are critical frequency and fatigue (Shigley, 1989). In a complete operation cycle of the dual hydraulic circuit the valves have two work conditions: closed, when there isn't pressure acting or opened when they suffer pressure's action. So, the spring's frequencies are low with minimal chances of critical frequency failure. Only the electric directional valve presents another failure mode, characterized by the solenoids burning. The valves aren't analyzed in this paper.

4. Application Example: Reliability Analysis of a Hydraulic System of a Deep Drawing Press for 250 ton.

The press analyzed was designed for powder compaction operation, with the same hydraulic functions of a deep drawing press. This press has pressing pressure of 20 MPa and pressing speed of 2,5 mm/s. The Figure 2 shows the hydraulic circuit of the press and the Table 1 presents the critical hydraulic components as for reliability analysis.

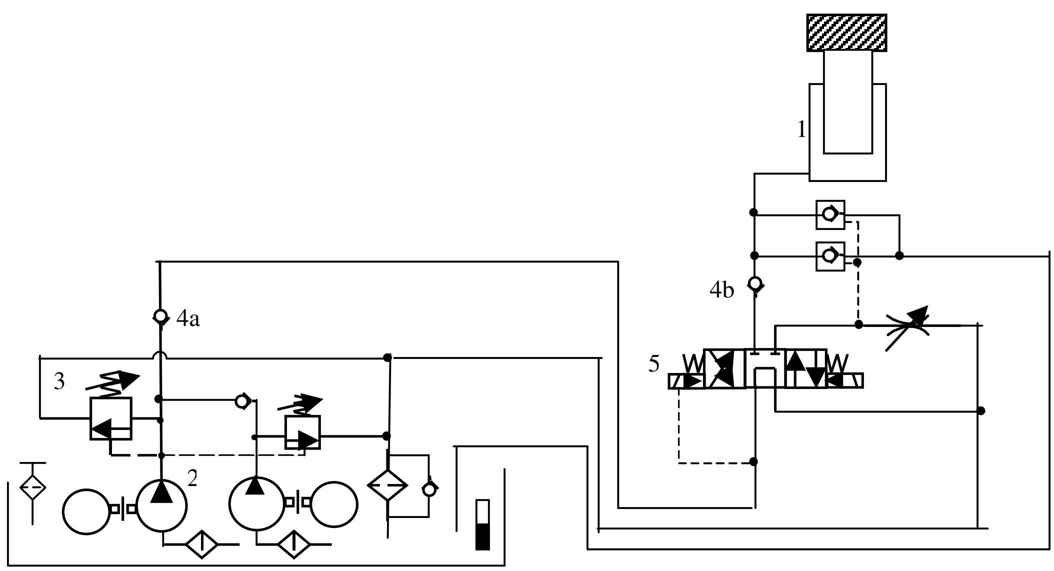

Figure 2. Dual Hydraulic Circuit for a 250 ton Press.

Table 1. Hydraulic Components under Critical Condition.

REF.	DESCRIPTION	QTT.	CHARACTERISTIC
1	Hydraulic cylinder	01	Φ = 400mm; stroke: 450mm; v_{ap} = 25 mm/s; v_{pr} = 2,5 mm/s; F = 250 ton
2	Pump	01	Q = 20 lpm; p = 200 bar
3	Relief valve	01	Q = 20 lpm; p = 200 bar
4a	Check valve	01	Q = 190 lpm
4b	Check valve	01	Q = 190 lpm
5	Directional valve	01	Q = 300 lpm; 24 VCC

N: power; Q: flow rate and p: pressure.

Based on the components failure criticality it's possible to build the FMEA table, presented at the Table 2 (Appendix 1) where it is possible to identify the failure modes associated with each of those critical components. The components failure rates with catastrophic criticality are considered of first priority, once their failure cause the system collapse, characterized by the impossibility of drawing.

The press reliability is related to the failure rates of the critical components. These failure rates must be estimated based on the physical model of the failure process for each component, mainly wear. Those models are presented in the following sections of this paper.

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4.1 Pump Reliability.

For the pump, the failure is represented by the impossibility to reach the minimal volumetric efficiency to keep the pressing speed, because the pump internal leakage doesn't allow the maintenance of the pressing pressure, so the tool's force isn't enough to form one pi ece.

The pump wear occurs in the vane – ring contact area (Gellrich, 1994), represented by deep clearance. The presence of clearances that allow internal leakage results in a critical volumetric efficiency drop. As the ring diameter is larger than vanes width, the rectangular clearance flow rate expression (Linsingen, 2001) may be used to determinate the clearance depth like a vane – ring clearance based upon material removed by the ring's surface wear or the vane top wear.

The rectangular clearance flow rate expression is:

$$Q_f = \frac{bh^3}{12.\mu.l}\left(p_1 - p_2\right) \pm \frac{bh}{2} v \tag{1}$$

where b is the vane width, l is the vane thickness, p_1 is the exit pressure, p_2 is the suction pressure, v is the vane speed, h is the fissure thickness and μ is the dynamic viscosity.

The high-pressure pump of the analyzed circuit has a design flow rate of 20 lpm, at 20 MPa of pressure, at an electric motor speed of 1.750 rpm. So, its calculated volumetric displacement is 11,43 cm^3/rot (Linsingen, 2001). To Vickers vane pumps, series 20 VQ, nominal size 5, there is a real volumetric displacement of 18,03 cm^3/rot with theoretical flow rate of 31,5 lpm, at 1.750 rpm. To this pump, the Vickers Catalogue (1987) points the volumetric efficiency at 20 MPa (1.750 rpm) of 78%, the vane width b is 0,025 m and the vane thickness l is 0,002 m. At the continuous work condition the necessary flow rate for a pressing speed of 2,5 mm/s, at the hydraulic actuator with diameter 400 mm is 18,8 lpm and in Vickers catalogue, at 20 MPa the real flow rate is 24,6 lpm.

Replacing those values in the Equation (1), the depth of the clearance, or the material removed by the ring's surface wear is:

$$h = 3{,}0 \times 10^{-7}\, m$$

According to this result, the volume of removed material will be:

$$V_{ret} = b.2\pi.r.h = 3{,}07 \times 10^{-7}\, m^3$$

and the removed material mass will be, considering the density ρ = 7806 kg/m^3:

$$m_{ret} = \rho.V_{ret} = 2{,}4 \times 10^{-3}\, kg$$

Based on the removed material estimate and using the wear graphic experimentally obtained by Gellrich et al (1994), presented at the Figure 1, and accepting linear wear after 250h of utilization, this mass would be removed in approximated 1050 hours of continuous work. Considering that the press operates 18,5% of the total time at 20 MPa, the pump's life may be calculated as follow:

$$life_{pump} = \frac{1050}{0{,}185} = 5676h.\frac{1day}{8h}.\frac{1month}{25days}.\frac{1year}{12months} = 2{,}4\,years$$

or in 2,4 years, the pump won't be capable of providing the required drawing pressure.

This estimated pump life of 2,4 years represents more than the double of the guarantee period time accepted for the pumps industries, about one year. At this period of time the return of pumps with fabrication failures, misleading design or operation, can be close to 10% of the total sold pumps, according to private information gave by Vickers salesmen.

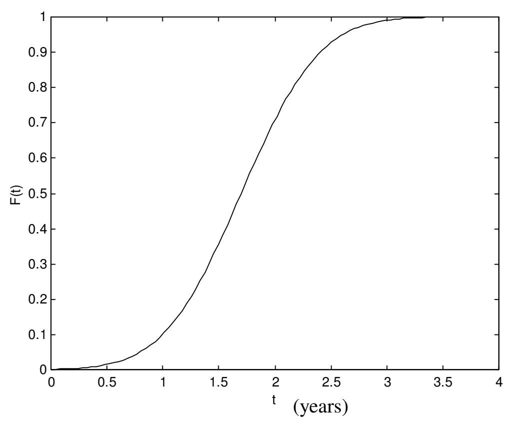
Considering this information and the estimated life, a normal probability distribution may be used to represent the reliability of 250 ton press hydraulic circuit vane pump, with following characteristics:
Failure probability at guarantee period of time: P(life < 1 year) = 10%
Failure probability during the estimated life: P(life < 2,4 years) = 90%

Using the Reduced Normal Distribution the mean (μ) and the standard deviation (σ) are:

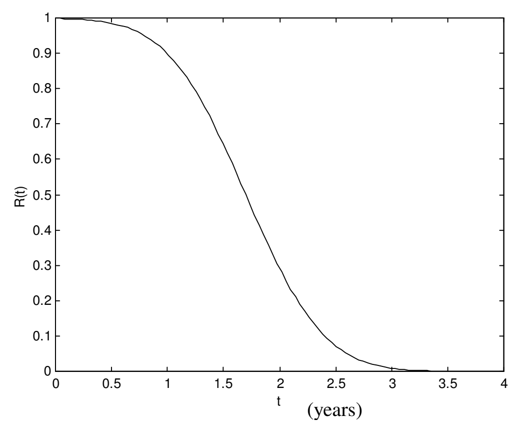
$$\mu = 1{,}70\ years$$

$$\sigma = 0{,}55\ years$$

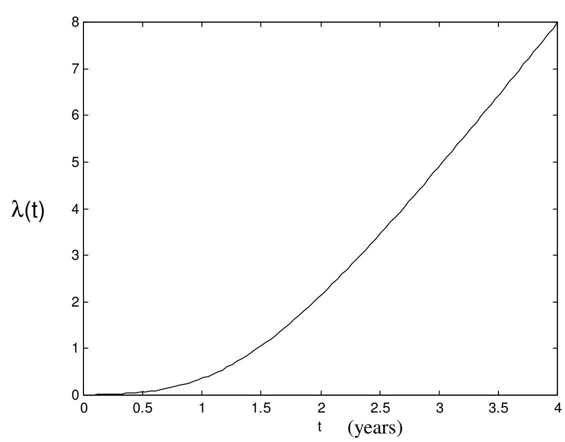
With these results, the Cumulative Distribution Function (CDF) (F(t)), the Reliability (R(t)) and the Failure Rate are represented in the graphics indicated at the Figure 3:



(a)



(b)



(c)

Figure 3. (a) Cumulative Distribution Function, (b) Reliability and (c) Failure Rate for the Vane Pump.

The Failure Rate $\lambda(t)$, which represents the probability that the component will fail in the next period of time, once it has survived until a given time t, increases after one year of operation, indicating the growth of the pump failure probability after the guarantee period.

4.2 Hydraulic Cylinder Reliability.

The hydraulic cylinder's life is related to the performance of rubber seals. There are many failure modes for seals based on wear or material changes, (Bosner, 2002). The common problem for seals at constant deformation by caused pressure is the elastomer stress relaxation, which limits the seal's life due to the loss of their initial stress and their deformation capability. The relaxation is divided in two categories: physical and chemical, which occurs by high temperature and chemical reactions, and is uncommon in power hydraulic circuits. The physical relaxation is associated with reorientation of the molecular network under strain (Gent, 1992). This process is initially fast and slow down with time. The relaxation expression relates stress relaxation in percent of decade of time (A), initial stress (σ_0) and the stress at an instant t (σ_t):

$$\frac{\sigma_0 - \sigma_t}{\sigma_0} = A.\log\left(\frac{t}{t_0}\right) \tag{2}$$

In the expression (2) the stress relaxation (A) is related to the change on the friction force magnitude between the initial instant t_0, when the rubber seals are new with an operating pressure of 20 MPa, and the instant t_1, when the seals are relaxed increasing the operating pressure to 21 Mpa, limited by the electric motor power. This friction can be calculated considering the actuator dynamics (Linsingen, 2001). Once the A coefficient is defined, using the chart presented in Figure 4, the relaxation time for the rubber seal, which is also the cylinder time to failure or operational life, can be calculated.

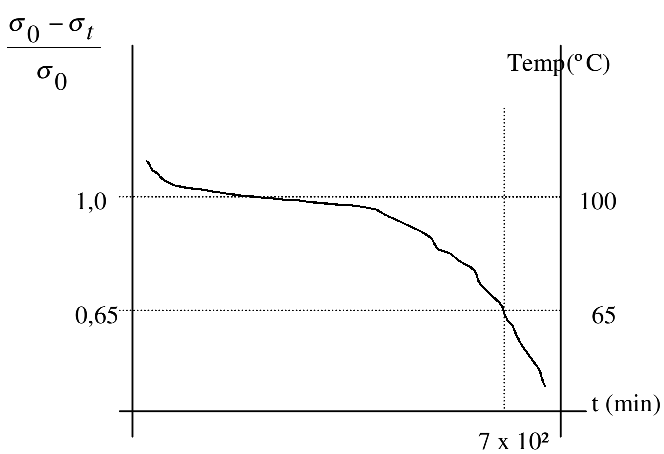
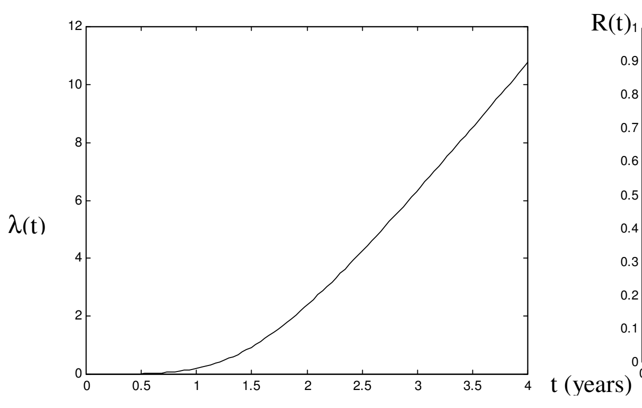


Figure 4. Stress Relaxation by Time and Temperature Chart (Gent, 1992)

For 65°C, operational limit temperature for nitrile rubber, the estimated relaxation time for the actuator gaskets is 2,6 years, similar to the pump operational life. The cylinder reliability can be estimated with the use of a normal probability function, considering the failure probability equal to 10% for one year and 90% for 2,6 years.



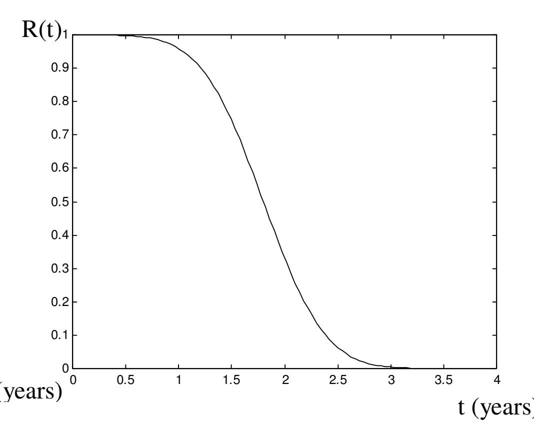


Figure 5. Reliability and Failure Rate for the Hydraulic Actuator.

In the Figure 5 chart, the failure rate is low for 1 year of the presswork, period of guarantee, increasing after this period. At the estimated life, 2,6 years, the failure rate is near 4,0 and presents a linear growth after this point.

4.3. Fault Tree.

Based on the obtained results for component reliability, it's possible to build a Fault Tree, which gives the global failure rate to the press. Figure 6 shows this fault tree, considering only the important system failures, the pump and the actuator failures, and the top event is the loss of drawing ability.

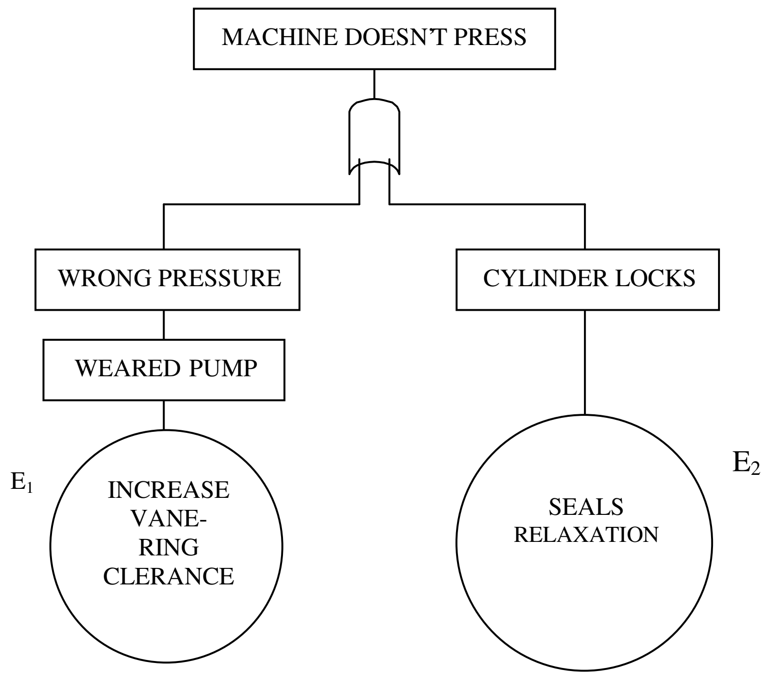


Figure 6. Fault Tree to the Deep Drawing Press Hydraulic System.

The top event occurrence probability is dependent on the pump and cylinder failure. Based on the failure mechanism of each component, the failures can be considered independent, and the top event probability is:

$$P(E_1 \cup E_2) = P(E_1) + P(E_2) - P(E_1 \cap E_2) \tag{3}$$

where:

$$P(E_1 \cap E_2) = P(E_1) \cdot P(E_2) \tag{4}$$

and

$$R(E_1 \cup E_2) = 1 - P(E_1 \cup E_2) \tag{5}$$

The E_1 and E_2 events refer to the two catastrophic failure modes of the press. The equation (3) indicates the calculation of the Global Failure Probability and the Figure 7 shows the Global Reliability Chart at the range from 0 to 4 years of hydraulic circuit operational life, done by the equation (5).

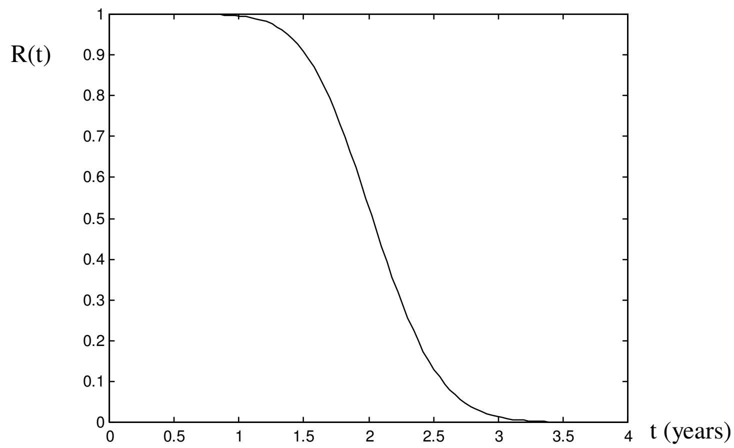


Figure 7. Global Reliability for the Hydraulic Circuit.

The hydraulic circuit reliability reaches 95% in a 1,5 years operational time and 50% in 2,0 years, showing a fast decrease. As the press reliability is highly dependent on the hydraulic circuit, a maintenance program should be developed to check the presence of contaminants in the hydraulic fluid, which would reduce the system reliability. Those results also shows that the pump or the cylinder will probably fail in 2 years.

5. Conclusions and Recommendations.

This work objectives are the application of Reliability Theory with the use of FMEA and Fault Tree methods to evaluate the failure probability of a hydraulic system used in deep drawing presses; defining the main failure modes of the hydraulic components, their causes and effects over the machine performance. Here is proposed a model to analyze the most used hydraulic system in deep drawing presses based on Operational Life concept and on the components failure modes analysis. The model is applied in a 250 ton press analysis, used for powder compact, which operates like deep drawing presses and employ their same hydraulic system.

The development of this study allowed the following conclusions:

1. The hydraulic system reliability used in deep drawing presses must be analyzed considering work conditions of the machine and fluid maintenance or the results may be different than the practical observation.

2. The reliability analysis must have a well-defined criterion chosen by an interpretation of the machine functional characteristics and of the hydraulic system, considering the critical operation conditions; at this work the criterion was the hydraulic components wear and the critical condition was the press operation pressure.

3. The proposed model was applied considering the hydraulic components failure modes obtained at the FMEA analysis and the system failure probability obtained at the fault tree. The equipments wear models are available in the specialized literature avoiding the quantitative analysis.

4. Considering the components operational life, the model indicates that the pump is the most critical component followed by the hydraulic cylinder, and the valves wear isn't significant to the reliability determination of this hydraulic system.

6. The proposed model validation depends on more consistent data given by hydraulic presses and hydraulic components producers.

For future studies the influence of temperature changes and contamination in the real operation conditions of the press, the principal fluid degradation factors, should be considered in the reliability analysis.

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Appendix 1. Failure Mode and Effects Analysis for Hydraulic Systems Components.

Table 2. FMEA table for the Hydraulics Components

FMEA						
Item	Failure Modes	Cause of Failure	Possible Effects	Failure Rate	Criticality	Actions to Reduce Failure Rates or Effects
High pressure pump	Pressure drop; cracking	Excessive wear; contamination; aeration; cavitation; without suction; not aligned; damage coupling	Machine stopped (no pressing pressure)	λ_1	Catastrophic	Change filters; clean suction tube, clean air filter, verify fluid level, verify viscosity, replace coupling; verify pump and motor axis; Remake alignment, verify connectors; bleed system; repair or replace the pump.
High pressure relief valve	Pressure drop; cracking	Excessive wear; spring cracking	Machine stopped (no pressing pressure)	λ_2	Catastrophic	Repair or replace the valve
Check valve for high flow rate pump isolation	Locking; leakage	Excessive wear; spring cracking	No approximation speed; pressing pressure loss	λ_3	High; Critical	Repair or replace the valve
Check valve for pumps isolation	Locking; leakage	Excessive wear; spring cracking	Directional valve not acting; pressing pressure loss; machine stopped	λ_4	Catastrophic	Repair or replace the valve
Directional valve	Not acting; locking	Solenoid without energy; solenoid burning; locking	Machine stopped	λ_5	Catastrophic	Verify electric panel, change solenoids, repair or replace the valve
Cylinder	Locking; Instable speed	Contamination; leakage; cracking	Cylinder doesn't press	λ_6	Catastrophic	Change filters and fluid, verify connectors, bleed system, repair or replace seals.