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A New Technique For Improving The Performance Of A Dumping External Gear Pump

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Abstract. A new modification technique to control the liquid leakage due to internal wearing of the gear pump casing at the suction side is investigated. The basis of amendment depends on loss compensation by applying a pressure force on the gear bearings towards the wear location. Volumetric efficiency, mechanical efficiency and overall efficiency were calculated versus flow rate, pressure and the rotational speed. The rotational speed was ranging between 1500-0 rpm, the output pressure was between 5-60 bar and the corresponding flow rate was ranging between 4-12 L/min. The results showed a very good improvement of the volumetric efficiency. Volumetric efficiency was increased from 54% to 80% at 1500 rpm.

Nomenclature:

Q : Flow rate (lit/min)
n : Speed (rpm)
V_g : Geometric Stroke volume (cm³)
Δp : Pressure drop (bar)
T : Drive torque (N.m)
P : Output power (kw)
η_m : Mechanical efficiency, and subscript a,b referring to after and before modification
η_v : Volumetric efficiency, and subscript a,b referring to after and before modification
η_{over} : Overall efficiency, and subscript a,b referring to after and before modification
Q_{act} : Actual flow rate (lit/min)
T_{act} : Actual Torque (N.m)
n_{act} : Actual speed (rpm)
T_{th} : Theoretical Torque (N.m)
Δp_{act} : Actual pressure drop (bar)

1. Introduction

The volumetric efficiency of the gear pump is essentially determined by its frontal and circumferential clearances which is usually not constant along the whole circumference since the gear wheels often shift within bearing slackness towards the suction space.

Most of the leaks are used to cool and lubricate the bearings. After they pass through the bearings the leaks are directed, via special grooves, to the suction chamber but some of the volumetric losses pass directly through the gap to the suction space.

The volumetric efficiency is a curtail factor to increase the overall gear pump efficiency. An attention to the influence of the radial clearance between pump components should be deeply studied according to Emiliano Mucchi 2011.

The leakage through radial clearance was studied by David Del Campo 2012 to give a solution for the cavitation problem associated with this kind of damage in the gear pump. But he did not give a solution for the leakage itself.

Some researches were conducted to suggest an engineering development to overcome the leakage due to radial clearance. Some of them were experimental such as that introduced by Pitor Osinski 2015, and others were numerically such as that by Adam Deptula 2013, who used the logic decision tree method.

Piotr Osiniski suggested a mechanical compensation by internal tightness pressing on the loading bearing and directed to the suction opening in order to decrease the radial clearance and the leakage.

Both Adam Deptula and Piotr Osiniski provided positive results with the same range of rotational speed 500 – 2000 rpm to increase the volumetric efficiency to exceed 90%

2. Test rig

An experimental study was developed to improve the volumetric efficiency.

The static characteristics of the external gear pump and the power unit were specified in Fig (1). The tested pump was Turolla (Aluminum gear pump) 14.64 lit/min and Vcc = 3.66 cm³/rev. It was driven by siemens motor; 0.75KW, 50HZ and max speed n = 1500, with differential voltage ranging 380 – 420 volts.

The number of revolutions (n)of the pump was controlled on the driving motor shaft bySiemensInverter.Tested pump is loaded throughThrottle valve and the actual delivery was measured by analogflow meter with a measuring range of 0 – 120 LPMflow and 0 – 400 bar pressure. Output torque of the pump was measured by photo/contact tachometer.

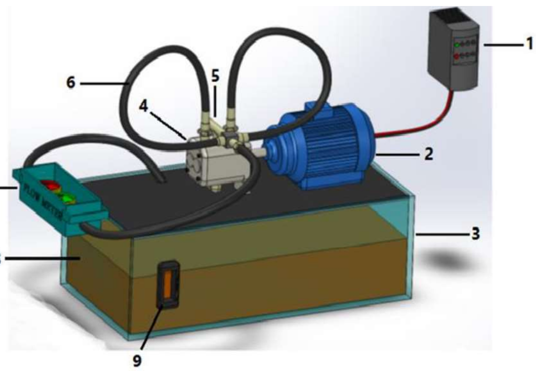


Figure (1): Schematic diagram for hydraulic power unit for the external gear pump

The test components:
1-Inverter
2- Driven motor
3- Tank
4- Gear pump
5- Mutual
6- Hydraulic hoses
7- Flow meter
8- Hydraulic oil
9- Oil obvious

All experiments were conducted at Training Sector - MTI- in the Technical Institute for Developed Industries (TIDI) at the hydraulic department sector.

The test on the mentioned pump was carried out before and after amendment. The amendment was carried out by applying a pressure on the bearing compensating discs towards the suction side of the gear pump (see Fig (2)). Two pressurized stainless-steel connections were fixed on small ports as bypass for flowing pressurized oil through a hydraulic hosehandled from the delivery line over the centroid of the radial direction compensators which substitutes the wear on the suction area.
The Pump loading was affected by a range of pressures and its related rotational speed according to the following table:

Pressure range (bar)	Rotational speed (rpm)
20 - 60	1500
5 - 25	1000
5 - 15	500

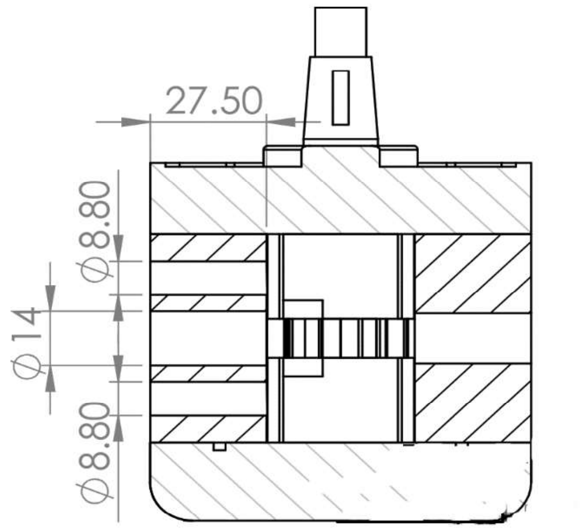


Fig (2): Diagram shows the sections of the pump with the modification made

3. Results

The experiments were conducted on the gear pump to investigate the effect of implementing pressurized oil to the bearing discs. The exit pressure was adjusted using a throttle valve, and consequently the volume flow rate Q (L/min) and the rotational speed n (rpm) were changed as both were related.

$$Q = \frac{Vg * n * \eta v}{1000}$$

Hence, the outlet pressure p (bar) was changed through the range of speeds as mentioned in table (1).
These ranges were determined practically by using a controlling inverter to achieve the maximum pressure limit in each speed range. The torque T (N.m), power P (kw) and the actual torque T_{act} (N.m) can be estimated by using the following equations:

$$T_{th} = \frac{Vg * \Delta p}{20 \pi * \eta m}$$

$$P = \frac{Q * \Delta p}{600 * \eta v * \eta m}$$

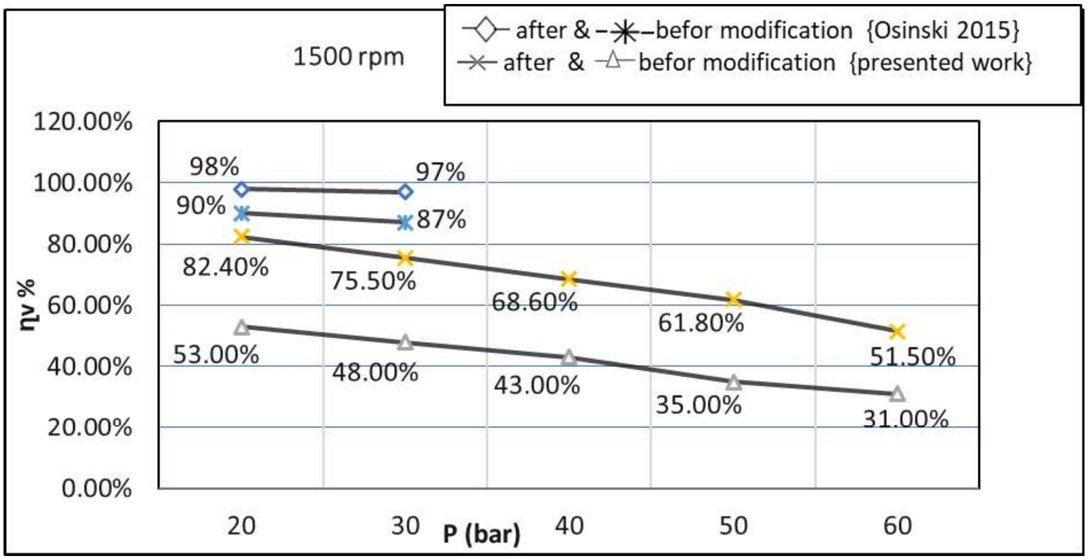
$$T_{act} = \frac{Qact * \Delta pact}{10 * \eta v * \eta m * 2 \pi * nact}$$

The exit pressure, volume flowrate and rotational speed were recorded; however, the volumetric efficiency η_v, mechanical efficiency η_m and overall efficiency η_{over} were calculated.

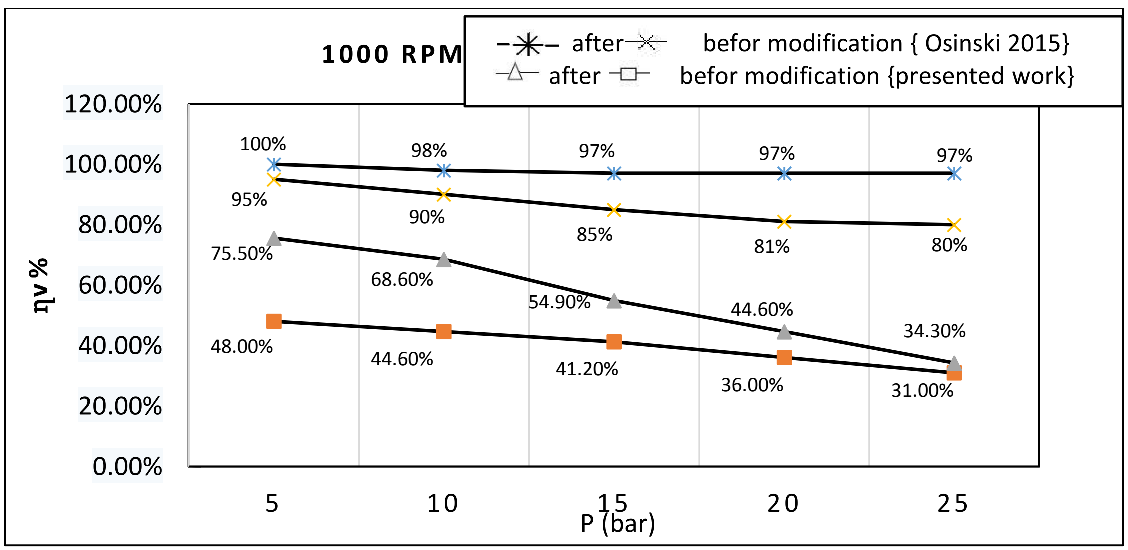
$$\eta_v = \frac{Qact}{Qth}$$

$$\eta_m = \frac{Tth}{Tact}$$

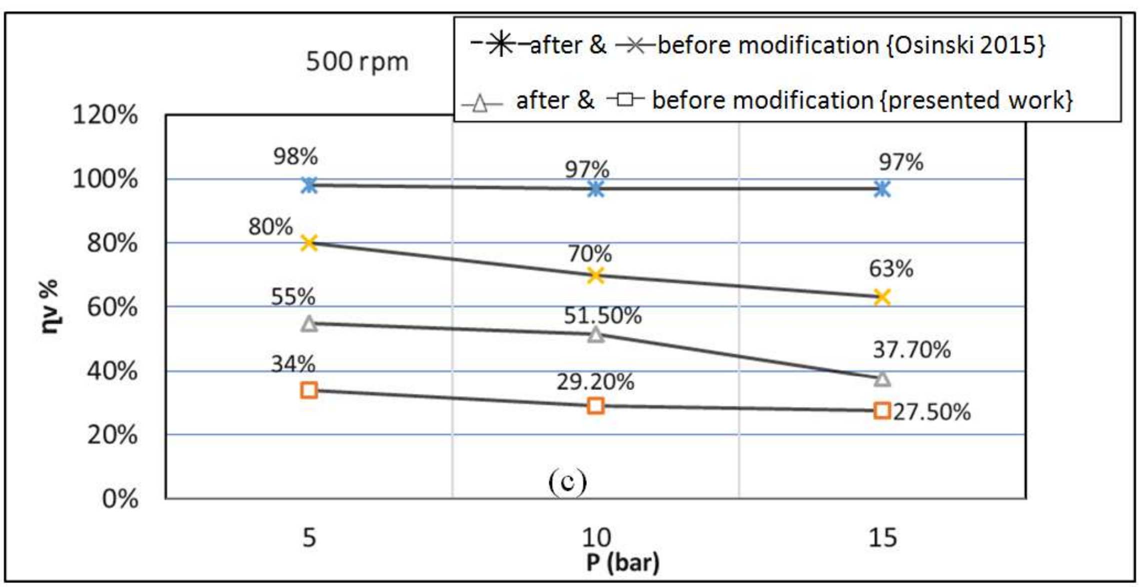
$$\eta_{over} = \eta_v \times \eta_m$$



(a)



(b)



(c)

Fig(3): Relationships (a), (b) and (c) between volumetric efficiency after and before the modification and exit pressure at different speeds for Pitor Osinski 2015 and the presented work

Figure (3) shows three curves for three speeds each curve is denoted by the abbriviations showen at each curve which refers to the results before and after modifications Osinski 2015 and the presented work.

It can be seen that increasing the exit pressure p (using throttle valve) leads to decreasing the rotational speed and consequently decreasing the volumetric efficiency η_v, for all speeds.

For the speed range 1500 rpm, it can be seen that the volumetric efficiency is increased by an approximatlly same value as the volumetric efficiency curves for both cases (befor and after) modification – Osinski and presented work - are approximately parallel. It increased by 29 % at 20 bar and increased by 21 % at 60 bar at the same speed.

For 1000 rpm and 500 rpm speeds, the volumetric efficiency enhancement is valiable at the minimum pressure (5 bar), while it tends to decrease with increasing the exit pressure for both speed ranges. At 1000 rpm the value of the volumetric efficiency increased by 21% at 5 bar pressure, while it increased just by 3% at 25 bar as the performance of the driving motor affected by the lowest speed reduction.

At the lowest speed 500 rpm, the volumetric efficiency is increased about 14% at 5 bar, while increased by 9% at 15 bar.

A comparison between the results of the volumetric effeiciency in this work and those obtained by Piotr Osinski 2015 is conducted, to validate the results with different modification technique. Piotr's results illustrates the exit pressure limits (0 – 30) Mpa at rotational speed ranges from 500, 1000 and 1500 rpm. It is clear that volumetric efficiency decreases as the exit pressure increasing for all speed ranges, which is making good agreement with the results in this this work. On the other hand, Piotr found that the volumetric efficiency enhancement at low pressure (5 Mpa) is very little comparing to the enhancement percentage at the maximum pressure (30 Mpa).

That is contradicts with the results in this work which shows a maximum enhacement in volumetric efficiency at minimum pressure and vice versa. This may be attributed to that the volumetric efficiency is strongly related to the volume flow rate, which is turned to be very small amount when raising the pressure by closing the throttle valve.

The mechanical and overall efficiencies η_m, η_{over} as a function of exit pressure p, are showenin Fig (4) and Fig (5).

There are three pairs of curves for three speed ranges with same notations as mentioned before.

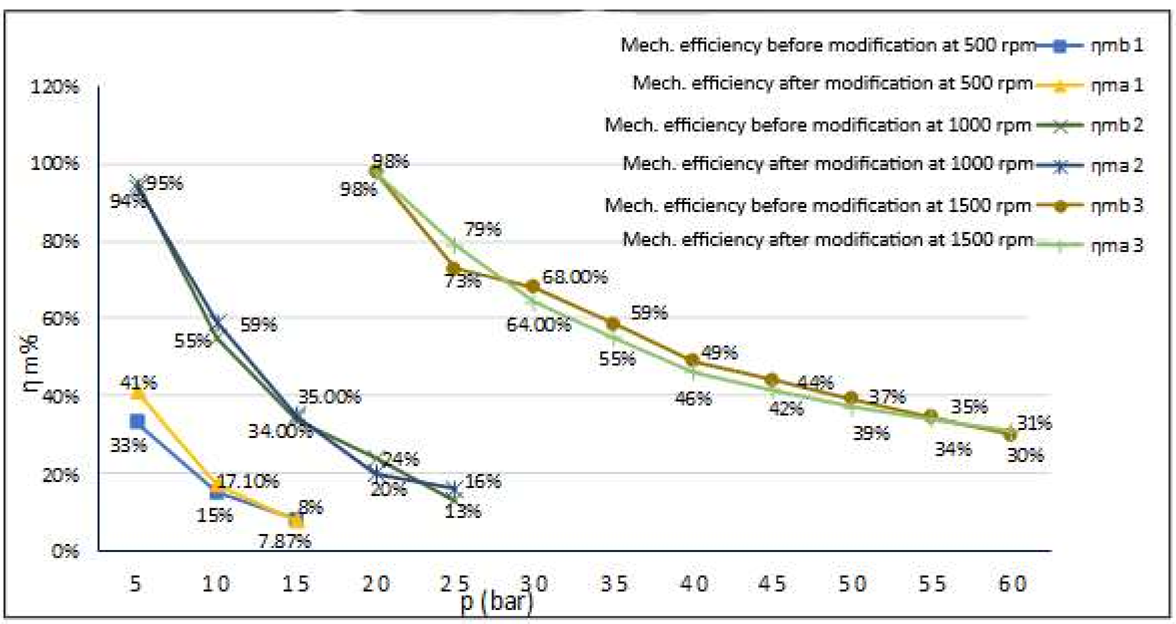


Fig (4): Relationship between mechanical efficiency after and before the modification and exit pressure at different speeds

Figure (4) shows no effictive difference on the mechanical efficiency ηm after and before the modification at all speed ranges. As the mechanical efficiency is an expression of the driving motor performance, so it seemed to be relatively equal at the same ranges of speed.

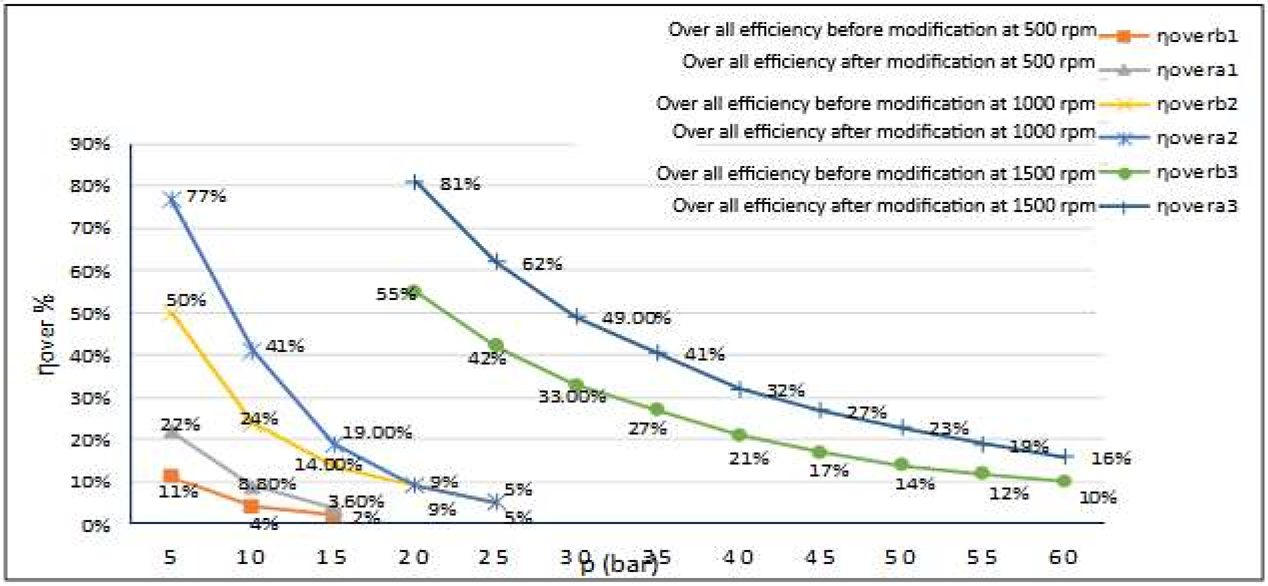


Fig (5): Relationship between overall efficiency after and before the modification and exit pressure at different speeds

According to the volumetric efficiency enhncement, a remarkable increase in overall efficiency η_{over} is indicated specially at the minimum exit pressure for all speed ranges.The overall efficiency η_{over} is increased by 26 % at 20 bar , 27 % at 5 bar and by 11 % at 5 bar, for the speed ranges 1500 rpm

Increasing the exit pressure leads to decrease the enhncement in η_{over}gradually so that η_{over} reach a value approximatelly equal to that before the modification at the allowable maximum pressure for each speed range.

The measurement devices are calibrated and the error was estimated for each measurment device:

The photo/contact Tachometer with model number Handy – 1 is calibrated according to calibration procedure number ML12 and environmental conditions and the calibration process expanded uncertainty $U_{expa}$for the equipment used equal to (± 0.14 %), the Analoge German Flowmeter with model number (WIKA – CL1.6) has a ±0.5 percent of reading. This percentage for a fully developed flow profile which usually means that sufficient straight – run pipe up and down stream and the maximum relative error in the flow rate difference is about 10.2% and the pressure gauge is calibrated according to calibration procedure number ML7 and the calibration process expanded uncertainty U_{expa} for the equipment used equal to (± 0.59 Bar) and the maximum relative error in the pressure difference is about 6.24%.

4. Conclusion

A new challengable development of leak compensation is introduced by applying bypass pressure line from the discharge line, leads to improve the overall efficiency in dumping pump by 26 %

The volumetric efficiency for the rated operating parameters could be increased to over 80% in comparison of its values before the modification. This proved that the flow rate approximately reach the delivery side completely.

Using this new technique in overcoming the increasing wear inside the pump casing, shows that the feeding pressure is proportional to the clearance size. Therefore, a high performance of the gear pump is achived with relatively no added cost.

The test results indicated that, in the nearest future, the development of gear pumps will be oriented towards feeding the pump itself by its normal pressure to get a higher pressure also better performance and higher efficiencies. All results show a good agreement with those obtained by Piotr Osinski2015.

5. References

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