



Modeling Engine Oil Variable Displacement Vane Pumps in 1D to Predict Performance, Pulsations, and Friction

2014-01-1086
Published 04/01/2014

Jonathan Harrison and Rodrigo Aihara

Gamma Technologies Inc.

Mojtaba Eshraghi and Irina Dmitrieva

General Motors Co.

CITATION: Harrison, J., Aihara, R., Eshraghi, M., and Dmitrieva, I., "Modeling Engine Oil Variable Displacement Vane Pumps in 1D to Predict Performance, Pulsations, and Friction," SAE Technical Paper 2014-01-1086, 2014, doi:10.4271/2014-01-1086.

Copyright © 2014 SAE International

Abstract

Variable displacement vane pumps are becoming more popular for engine oil circuits due to their fuel savings over traditional fixed displacement pumps. As a result, engineers need to analyze these pumps to ensure the pump design meets the demands of the oil circuit while having good friction characteristics and avoiding issues like high pressure amplitude and resonance. By employing 1D flow simulation to these pumps, the user can analyze the most important issues surrounding vane pumps at a fraction of the time as 3D CFD.

This paper showcases the prediction of several major performance quantities of a variable displacement vane pump including flow rate, pressure rise, and friction torque vs. engine speed and temperature. The simulation results show good correlation to measurement data. In addition, the pressure pulsation at several locations including in the vane chamber and at the outlet is compared directly with 3D CFD for a different pump. Furthermore, the effect of aerated oil on pump performance is also shown. Finally, a predictive friction model for vane pumps is proposed and shows good agreement with experiment.

Introduction

In recent years, variable displacement vane pumps (VDVP) have become widely used for engine oil circuits due to their fuel savings benefits over traditional fixed displacement pumps. Consequently, these pumps must be analyzed to ensure the pump size is correct for the system installed, produces minimal pressure fluctuations, and has minimized friction. Vane pumps are unique in that the eccentricity, and therefore the pump displacement, is easily regulated by fluid pressure acting on the outer stator, which typically can either pivot or slide relative to the inner spinning rotor. The regulation of the outer stator

movement is important to consider when modeling these pumps, as the displacement is a direct function of outer stator motion.

Vane pumps have been studied with various objectives in the past in both 1D and 3D CFD. For example, the reduced power requirement of a VDVP over traditional fixed displacement and variable flow gerotor pumps is clearly shown by Mancò, et al. [1]. The same authors [2] also described a mathematical approach to analyzing vane pumps in Amesim, a 1D software package. Some co-authors of [1-2] also used 1D to show the influence of the geometry on pump performance including port plate timing, stator ring geometry in both sliding and pivoting configurations, and influence of incomplete filling [3]. Arata et al. [4] also used Amesim 1D software to study the benefits of a two stage VDVP over a single stage pump. Geist and Resh [5] wrote their own 1D model to demonstrate pump influence on a simplified oil circuit model. A dynamic displacement VDVP was studied with Pumplinx, a 3D CFD software, by Wang, et al. [6] where the 3D CFD solution was coupled to a mechanical dynamics model of the outer stator to predict dynamic eccentricity instead of simply imposing the eccentricity as is commonly done in 3D CFD. Furthermore, Pumplinx 3D CFD was used by Sullivan and Sehmbly [7] to analyze internal pressure forces generated by the chambers in the pump, where they showcased the influence of pump speed and eccentricity on the internal forces and moment generated on the stator. Finally, a theoretical friction model for a vane pump was proposed by Inaguma [8], where an empirical correlation of the friction coefficient between the vane tip and outer stator (cam ring) was derived as a function of speed.

Though various 1D and 3D CFD tools have been used in the past to analyze vane pumps, the GT-SUITE software package offers a unique integration of an advanced 1D flow solution and

multi-body dynamics capability that incorporates many of the objectives outlined by this previous research [1, 2, 3, 4, 5, 6, 7, 8]. This paper will show a single tool can be used to analyze performance, pulsations and friction of a VDVP.

Modeling Vane Pumps in 1D

A schematic of a pivoting type VDVP with 5 vanes is shown in [Figure 1](#), along with the equivalent 1D flow network representation. Each chamber has a corresponding volume in the 1D model which changes as the pump rotates. These chambers also communicate with an inlet volume and outlet volume through port area profiles. Up to four ports are commonly seen on the inlet and outlet side due to primary ports on both sides of the vane along with secondary groove ports on both sides. For simplicity though, [Figure 1](#) only shows one port on the inlet and outlet side.

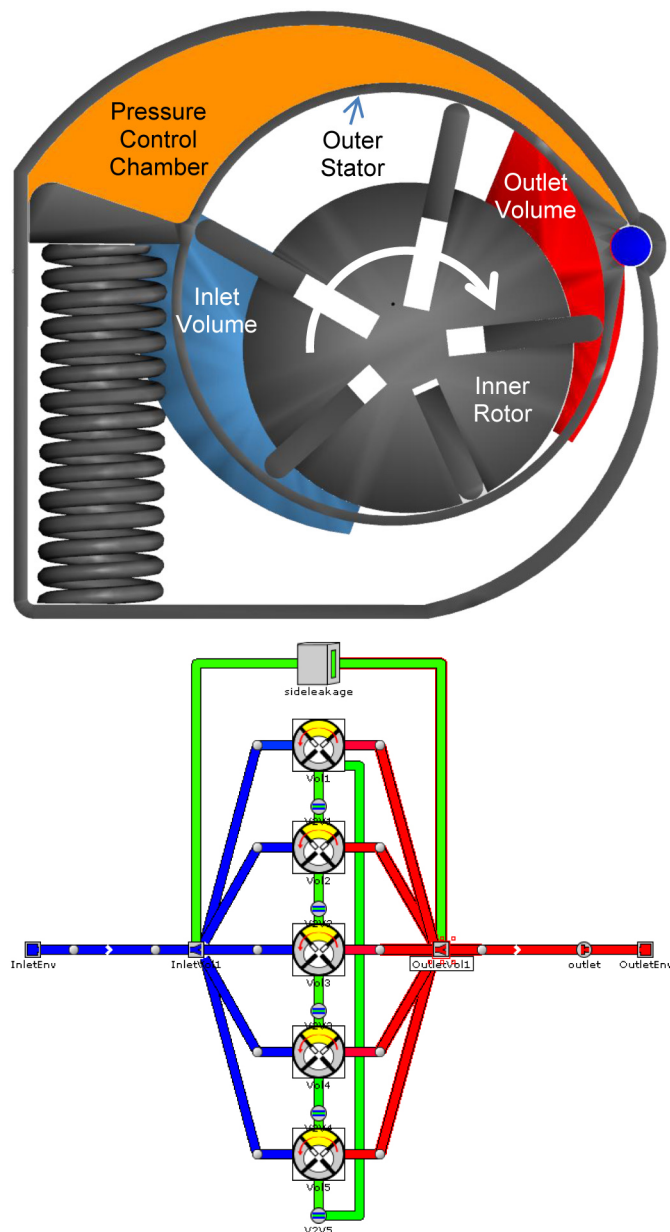


Figure 1. Schematic of a typical pivoting style VDVP and the corresponding 1D flow network representation. Low pressure paths are highlighted in blue, high pressure paths highlighted in red, and leakage paths in green.

In the 1D flow model, the pump flow system is discretized into many volumes comprised of pipes and flowsplits (volumes), and pipe passages are further discretized into subvolumes. The 1D Navier-Stokes equations, namely conservation of mass, momentum and energy are solved every timestep in every subvolume. Conservation of species is also considered to solve for species transport throughout the system. This is very important in a VDVP as pump performance can be significantly affected by aeration content or cavitation. In [Figure 2](#), the 1D staggered grid approach used by GT-SUITE is shown, where scalar quantities like pressure, internal energy, temperature, density, etc. are solved at the center of each subvolume, and scalar quantities like mass flow rate, velocity, species flow rate, etc. are solved at the boundary between subvolumes. Unlike other 1D tools, GT-SUITE makes no assumption about the transient evolution of the species content in a particular subvolume, rather it solves for the transient mass fraction of each species in each subvolume. A fluid mixture is considered homogeneously mixed within a given subvolume, and consequently flow from one subvolume into another will be based on the species mass fractions in the upstream subvolume. In terms of cavitation, if the pressure in any subvolume falls below the vapor pressure of a given liquid at a given temperature, the solver will iterate to calculate the amount of liquid that has cavitated. This cavitating vapor can then be transported throughout the system.

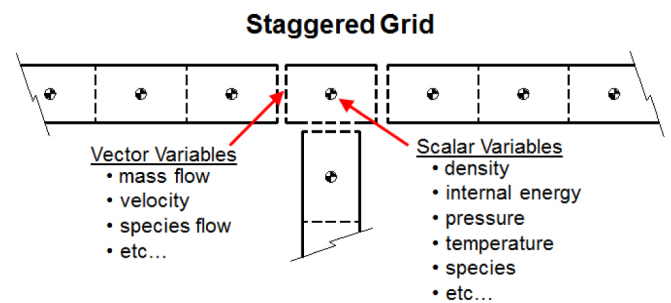


Figure 2. Staggered grid solution utilized in GT-SUITE for solving Navier-Stokes equations and species transport in 1D.

VDVP Model Building Requirements

Building vane pump models can be time consuming, as a complete model requires the following critical information:

1. Discretizing the entire flow path into subvolumes
2. The volume and each port area profile must be defined
3. All dominant leakage paths must be considered
4. A mechanical system must be setup to predict outer stator motion from all pressure, spring, and friction forces to determine eccentricity

The model building process for vane pumps can be made more efficient by using the GEM3D pre-processor in GT-SUITE. The pre-processor allows the user to import a CAD model of the vane pump, point to several shapes representing different components of the pump, then the 1D model will be built automatically for the user at a fraction of the time it would take to build the model by hand.

In [Figure 3](#), an example VDVP CAD model is shown imported in GEM3D, along with the template used to create the VDVP model. In short, the user must point to the CAD shape representing the outer stator, inner rotor, one of the vanes, at least one inlet volume, and at least one outlet volume. Additionally, if the user wishes to predict the eccentricity dynamically, additional information about the regulation spring preload, stiffness, and orientation, as well as control chamber orientation must be defined in the template. Once the template is defined, the user can export the model to the 1D software interface and the model will be built automatically, similar to that shown in [Figure 1](#), where all the volumes will be discretized and linked together, volume and port area profiles will be automatically calculated, and optionally a mechanical system representing the outer stator will be connected to predict the eccentricity dynamically.

To complete the model and run the simulation, the user will need to define boundary conditions in the flow system including initial pressure, temperature, and fluid composition. Other inputs are also required such as pump speed, run time, and plot output.

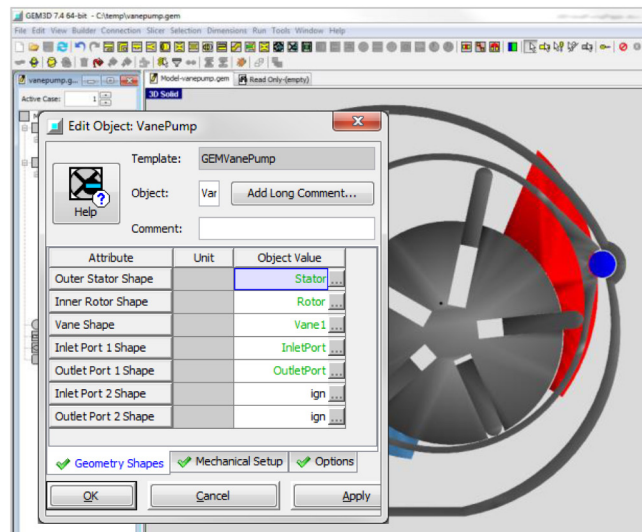


Figure 3. Example pump setup in GEM3D pre-processor of GT-SUITE to create a VDVP model automatically from CAD.

GM VDVP Model Setup

A GM light-duty automotive VDVP 1D model was built to predict pump performance and compared to experiment where the pump was placed on a test rig as shown in [Figure 4](#). An orifice restriction was used in the experiment to represent the pressure drop across the oil filter in the engine, and a smaller restriction was used at the outlet to represent the equivalent restriction of the engine oil circuit. Between these two restrictions an oil passage is connected to pressurize the pressure control chamber. Though the pump has two pressure control chambers which allows for a high pressure and low pressure mode, only the high pressure mode was examined (i.e. the primary pressure control chamber was pressurized, but the secondary control chamber was not). In the experiment, the

speed was ramped from zero to high speed at two different temperatures, and the pressure, flow and torque were sampled every 0.05sec.

The torque was measured at the motor and the rig friction was separated from the pump torque. The flow was measured with a flow meter placed upstream of the filter orifice, and its restriction characteristic was lumped into the oil filter restriction in the model. This was done by applying the Bernoulli flow equation through an orifice plate [9] using the experimental flow rate, Q , and pressure drop across the filter orifice, Δp_{orif} as shown in [Equation \(1\)](#), and solving for the orifice diameter, D_{orif} by setting the discharge coefficient, C_d , to 1.

$$Q = C_d \frac{\pi D_{orif}^2}{4} \sqrt{\frac{2\Delta p_{orif}}{\rho(1-\beta^4)}} \quad (1)$$

The density is represented by ρ , and β represents the ratio of the orifice diameter to the upstream pipe diameter. The pressure difference is the difference between the sensor at location 1 and location 2 in [Figure 4](#).

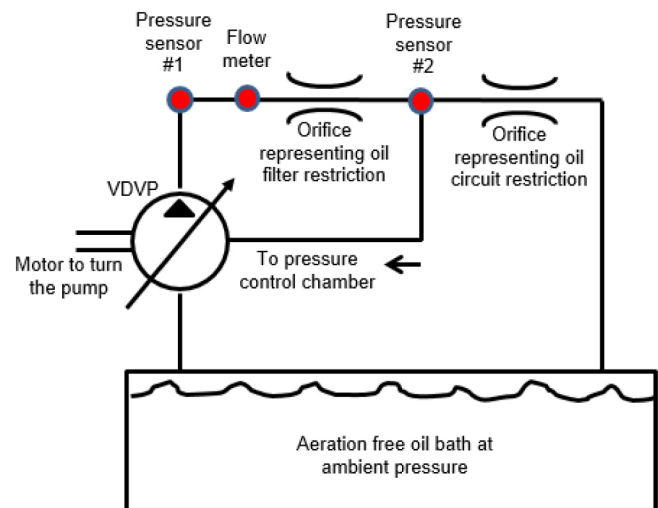


Figure 4. Experimental test rig setup of GM VDVP.

Information about the pump and experimental setup is outlined in [Table 1](#). Since the flow meter does not behave like a perfect orifice plate, but has viscosity dependence due to friction, the diameter was not constant in the model, rather it varied with temperature and speed to mimic the pressure drop across the orifices and flow meter in the experiment. The values quoted in [Table 1](#) for the orifice diameters are the nominal values.

The corresponding 1D model is shown in [Figure 5](#). The pump has 7 vanes and therefore has 7 corresponding chambers with changing volume. There are 4 inlet ports leading to each chamber and 3 outlet ports leaving from each chamber. The low pressure paths are highlighted blue, high pressure paths are red, leakage paths are green, and the path leading to the pressure control chamber is orange. There are 78 flow volumes in the system. All 7 vane chambers are connected to the VaneShaft part to retrieve the instantaneous volume and port area, as the VaneShaft is where the user defines all the

pump geometry required to calculate the volume vs. rotation angle and eccentricity, and is where the user defines the port areas for the inlet and outlet. The instantaneous pressure from each chamber is then sent back to the VaneShaft to calculate the net pressure force acting on the outer stator. These pressure forces are then passed to the mechanical assembly.

Table 1. Specifications of the pump and experimental setup

Pump Maximum Displacement	19 cc/rev
Oil Type	5W20
Speed Range	0-6500 RPM
Temperatures Analyzed	30°C and 140°C
Orifice diameter representing oil filter (also includes flow meter restriction)	5.9 mm
Outlet orifice diameter (representing oil circuit resistance)	3.55 mm
Aeration Content	0 (tester stated experiment was aeration free, but not measured)

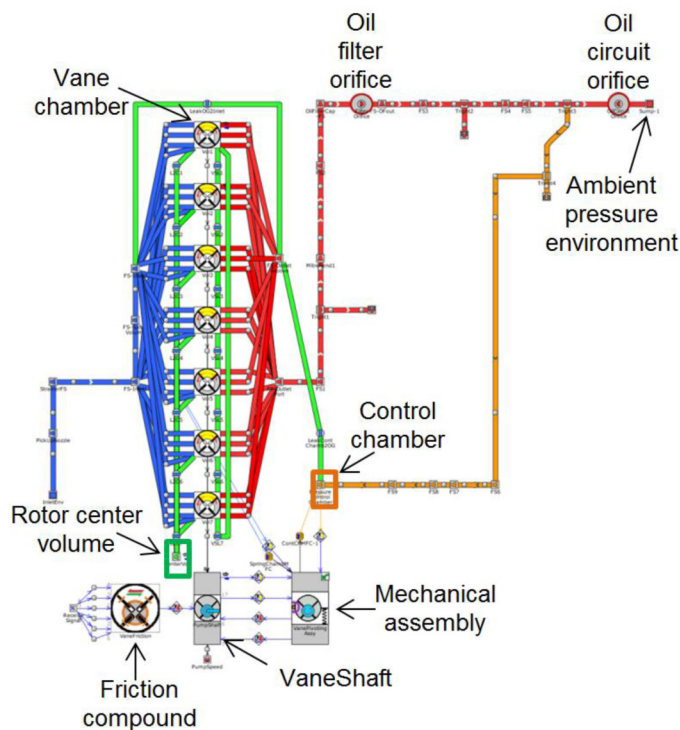


Figure 5. GT-SUITE model setup of GM VDVP in high pressure mode.

The model contains a 2D multi-body dynamics mechanical assembly which calculates the eccentricity dynamically based on the balance of torques acting on the pivoting stator, where the external forces/torques acting on the outer stator are shown in [Figure 6](#), and include:

1. internal pressure forces from each chamber
2. regulation spring force
3. control chamber #1 pressure force
4. control chamber #2 pressure force (~0, not pressurized)
5. spring chamber force, nominally at inlet pressure
6. dry friction force from seals (acts as a damping force)

7. dry friction at pivot (acts as a damping force)
8. viscous friction between outer stator side and housing

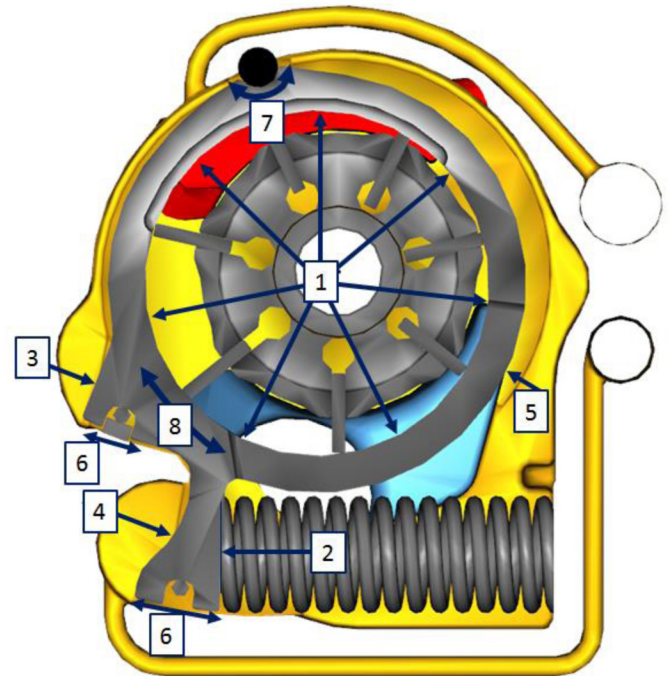


Figure 6. External forces/torques acting on outer stator in the model.

The model also contains a friction template to calculate ten components contributing to total friction, including:

1. dry friction between vane tip and outer stator
2. viscous friction between vane tip and outer stator
3. viscous friction between rotor sides and housing
4. viscous friction between vane sides and housing
5. viscous friction in bearings supporting the shaft of the inner rotor
6. dry friction between vane side and slot of inner rotor
7. viscous friction between vane side and slot of inner rotor
8. viscous friction between outer stator and housing
9. dry friction between any seals and the housing
10. dry friction between outer stator and pivot point

Among these ten components, the most dominant component is due to dry friction between the vane tip and outer stator. Please see the [Appendix](#) for the governing equations used for each friction component.

The normalized volume and area profiles at eccentricity ratio of 1 and 0.25 are shown in [Figure 7](#). The groove ports play a crucial role in reducing the pressure amplitude at lower eccentricity because the minimum volume point advances in the cycle with smaller eccentricity, and since the grooves are cutouts on the moving stator, they open earlier as the eccentricity decreases as well. This effectively reduces pressure spikes that would otherwise occur due to the shift between the advancing volume profile and fixed primary port profiles.

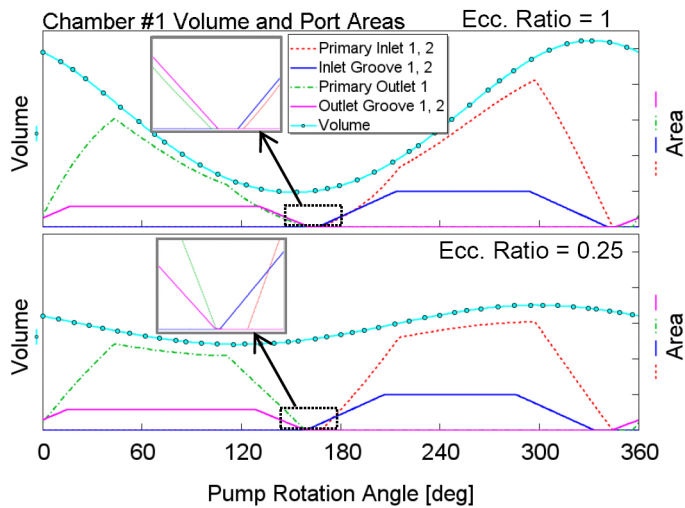


Figure 7. Normalized volume and port areas at maximum and small eccentricity.

Internal leakage in the pump due to gaps between moving parts must be considered when modeling the pump. Figure 8 shows several important leakage paths considered. For example, the gap between the vane side and housing, c_{vane} , will influence the peak pressure seen in the chamber when the chamber is closed to the inlet and outlet, and it will also govern leakage loss in the pump. The gap between the inner rotor and housing, c_{rotor} , will influence leakage between the chambers and the center volume of the rotor, which will also act to pressurize the center volume. The pressure in this center volume will then act as a pressure on the vanes causing them to be pressed against the housing and produce friction. The gap between the outer stator and housing, c_{stator} , will influence how much oil leaks between the outlet grooves and inlet volume since the outlet grooves are cut directly into the outer stator. Finally the gap between the vane tip and outer stator, c_{tip} , is believed to be non-zero but since this film is thin with a high film pressure, leakage is not assumed to flow between vane chambers via this path. All of these gaps will influence viscous friction in the pump, particularly at high speed and low oil temperature. Though it would be good to make the clearances as small as possible to prevent internal leakage, doing so will also increase friction loss.

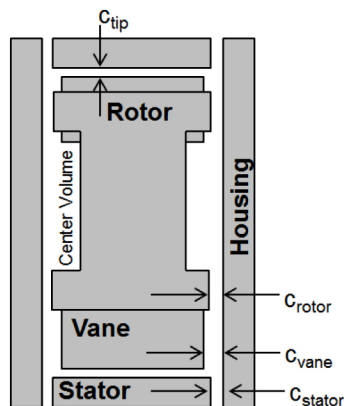


Figure 8. Leakage gaps inside the pump.

In the model these leakage paths assume Poiseuille flow in a small gap between flat plates, where the flow, Q , goes as:

$$Q = \frac{c^3 W \Delta p}{12 \mu L} \quad (2)$$

where c represents the clearance between the flat plates, W represents the width of the passage, Δp represents the pressure drop across the leakage path, μ represents the dynamic viscosity, and L represents the leakage path length.

Results and Comparison

Pump Performance

The VDVP was compared between the model and experiment at two temperatures: 30°C and 140°C. Several output quantities were compared including flow rate, pressure rise and friction torque and this correlation is shown in Figure 9.

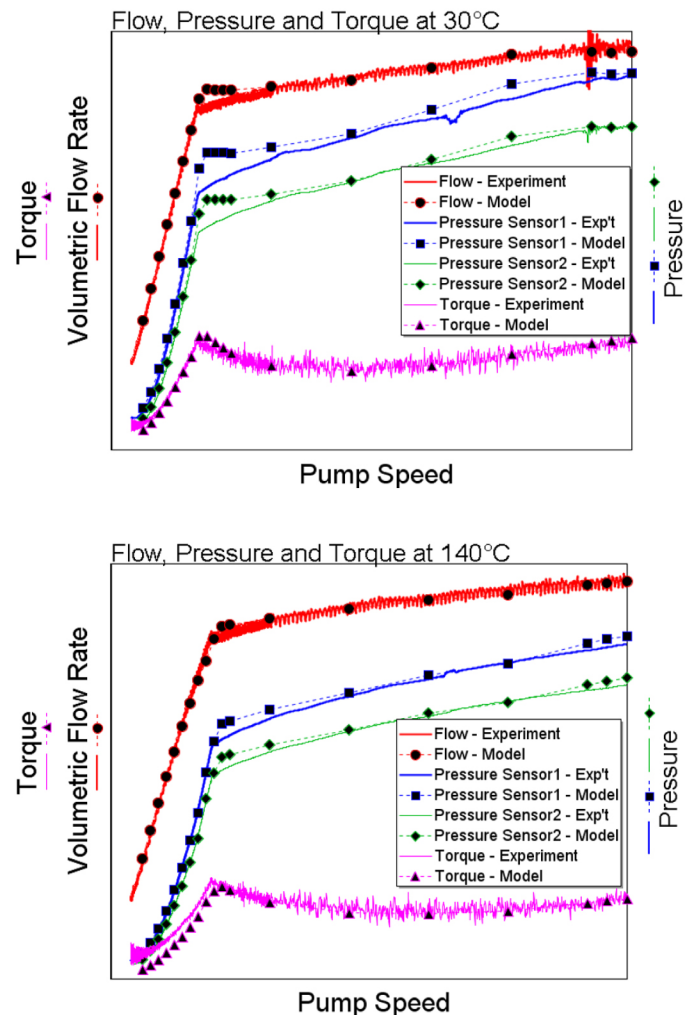


Figure 9. Comparison between model and experiment at 30°C and 140°C for flow rate, outlet pressure at two locations and torque.

All quantities which are subject to tolerances were kept at their nominal values for 30°C and 140°C, including the regulation spring preload and all leakage gaps. Though the pump housing was made of aluminum while the stator, rotor and vane were made of steel, thermal expansion does not appear significant in the leakage gaps and is not taken into account in the model. The pump was operating in high pressure mode, meaning the primary pressure control chamber was pressurized but the secondary control chamber was at ambient pressure. Although the model can account for aeration, the experiment was stated to have no air content in the oil, and thus only pure oil and its cavitating vapor are used in the model to compare to experiment.

In general, the agreement is good throughout the speed range at both temperatures. At 30°C though, the predicted flow and pressure overshoot the experimental values at the breakpoint when the pump eccentricity is reduced from its maximum position. This could be caused by the torque due to internal chamber pressure being over-predicted, causing the pump to stay at its maximum eccentricity position for a higher pressure rise than seen in experiment. At 140°C, the predicted flow and pressure match well throughout the speed range, but the friction is under-predicted at low speed. This could be due to a shortcoming in the friction model, though at intermediate and high speed the friction model correlates quite well with experiment.

Internal Pressure Forces

One of the key factors to matching the slope of flow rate vs. speed in [Figure 9](#) once the outer stator has moved away from the maximum eccentricity position is to have a good correlation for the internal pressure force in the chambers (component 1. in [Figure 6](#)). In the model the internal pressure force from each chamber is summed together at each timestep and imposed on the outer stator. This pressure force acts at the center of the outer stator and therefore exhibits a torque about the pivot point of the stator. The design of the GM pump is such that the internal pressure torque will naturally rotate the stator to its maximum eccentricity position. Thus, including the internal pressure force will slow the rate of the eccentricity decreasing with speed, as is evident in the flow rate curve of [Figure 10](#).

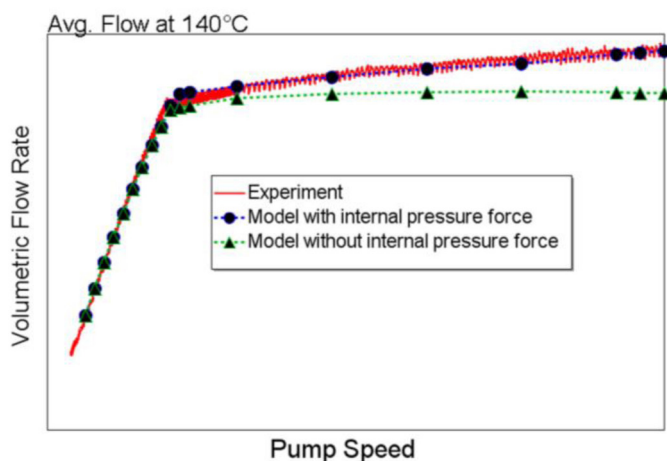


Figure 10. Showcasing the influence of internal pressure force on the flow rate.

The root cause of the internal pressure force/torque is due to pressure in each chamber, and an interesting point is the dependency of the internal pressure force changing with temperature. In [Figure 11](#), the influence of the internal pressure torque is shown at low and high temperature. It is evident the torque increases noticeably at low temperature compared to high temperature. This is primarily due to the pressure spike exhibited in the moments before outlet opening, and the higher pressure in the chamber outlet due to less internal leakage. At high temperature, there is sufficient leakage from chamber to chamber so as not to cause a very high pressure spike. But at low temperature this leakage is much smaller causing a larger spike during the moment when the volume is decreasing but not open to inlet or outlet ports. This is illustrated in [Figure 12](#). The high pressure spike at low temperature could cause vane detachment from the outer stator which would act to dampen the spike, but no detachment is considered in the model.

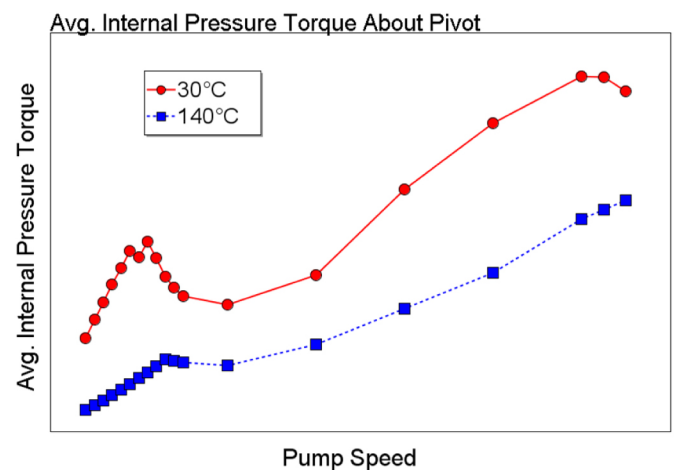


Figure 11. Comparison of internal pressure torque at 30°C and 140°C.

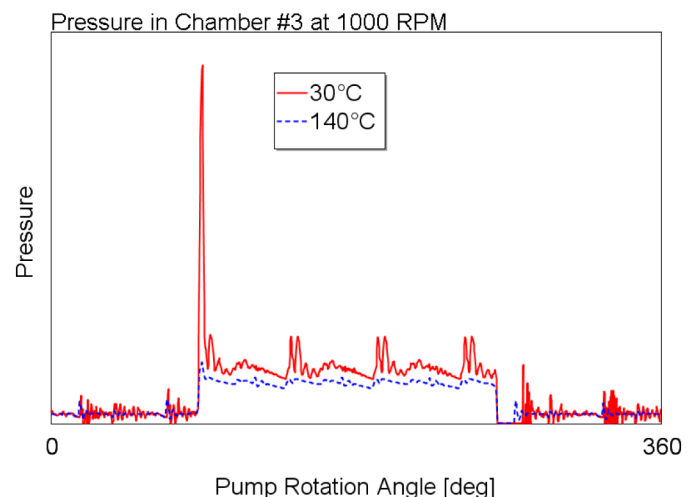


Figure 12. Comparison of internal chamber pressure at 30°C and 140°C and 1400 RPM.

Influence of Aeration and Cavitation

Aeration and cavitation can play a significant role in the performance of the pump, influencing not only pressure pulsations but also average flow rate, as shown in [Figure 13](#). When comparing to experiment, only pure oil and its cavitating vapor were considered in the model. This is based on

information from the tester that the experiment was aeration free, however, the amount of air was not experimentally measured. If 100% dissolved air (0% free air) is added to the oil, the flow rate is reduced, and if 10% free air is added on top of the 100% dissolved air to the inlet side of the pump, the flow is reduced even further. The 100% dissolved air condition is where the air is completely dissolved in the oil to the point where the oil cannot dissolve any more air at the given pressure and temperature. This follows Henry's Law where the amount of gas that can be dissolved in a liquid is directly proportional to pressure. Typically between 8-9% of air by volume can be extracted from oil at STP if the air was completely released from the dissolved state.

The 100% dissolved air case is interesting because this would represent a condition in an engine that has been at rest for quite some time, allowing the air to become dissolved in the oil. The 10% free air case is interesting because a typical engine can have foaming and bubbles in the oil sump, on the order of 10%.

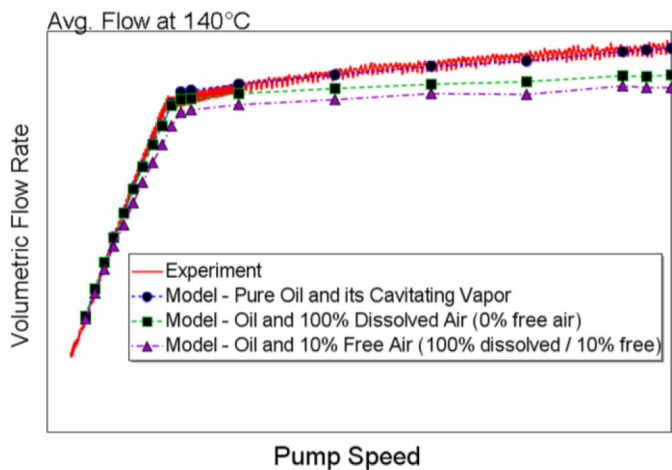


Figure 13. Flow rate comparison at different aeration levels.

The aeration level can significantly change the pressure response in the chamber. Adding air tends to dampen large pressure spikes that occur in the oil only case, but tends to delay pressurization of the chamber upon outlet opening, as exemplified in Figure 14.

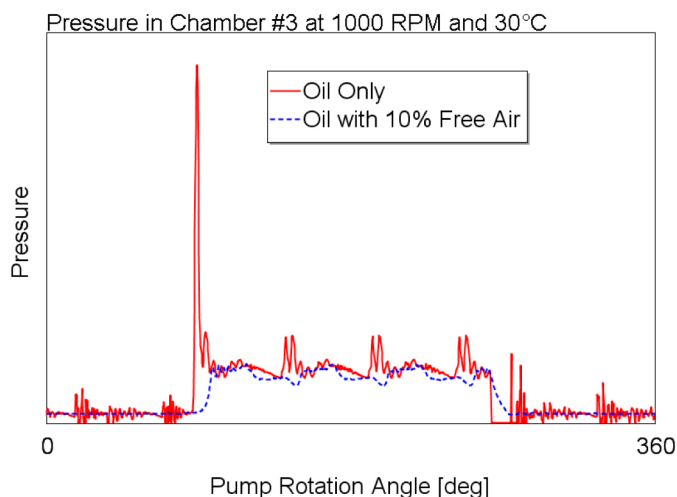


Figure 14. Effect of aeration on pressure in the vane chamber.

The influence aeration has on the pressure in the chamber translates to the internal pressure torque acting on the outer stator. As shown in Figure 15, increasing the aeration can lower or even reverse the sign of the internal pressure torque acting on the stator.

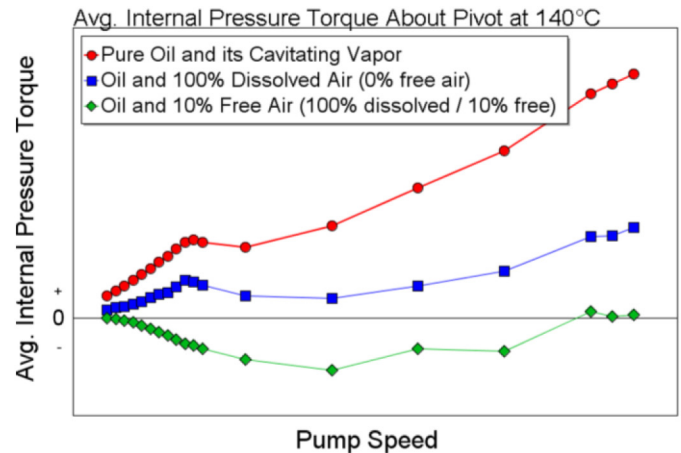


Figure 15. Effect of aeration on internal pressure torque acting about pivot point. A positive torque naturally turns the outer stator to maximum eccentricity, while a negative torque turns the stator to a smaller eccentricity.

Pressure Pulsations

The influence of pressure pulsations in the chamber volumes on pump performance has been showcased above, but pressure pulsations are also important to study to reduce pressure amplitudes, spikes and resonance as much as possible. In order to validate the pressure pulsation prediction of the 1D model, a second 1D model was built for a different GM production VDVP with different geometry than previously discussed. It is compared with PumpLinX, an industry standard 3D CFD tool [6-7], for a fixed eccentricity condition. The goal of this study was to see how well a 1D flow simulation tool could predict pressure pulsations compared to 3D CFD. Some specifications about the pump and the boundary conditions of the study are shown in Table 2.

Table 2. Specifications of the pump and model boundary conditions

Pump Maximum Displacement	33 cc/rev
Oil Type	5W20
Speed	3000 RPM
Temperature	120°C
Fixed Eccentricity Ratio	0.8
Free Air Content	10% at inlet
Outlet Restriction Diameter	7 mm
1D Subvolume Count / 3D Cell Count	49 subvolumes / 50,000 cells
Timestep size in simulation	1° rotation angle in 1D and 3DCFD

The flow domain studied by both GT-SUITE and PumpLinX is shown in Figure 16, and includes the geometry from the pickup tube to the outlet passage with a restriction at the end of the outlet of 7mm diameter which then flows into ambient pressure. The pump has nine vanes, hence there are nine pulsations per pump revolution. There were two sampling locations compared: in the vane pocket and in the outlet exhaust just upstream of the outlet restriction.

The port area profiles for this pump were calculated using a special tool within NX Unigraphics software, which is different than the previous pump discussed above which uses a special template within GT-SUITE to automatically calculate the area.

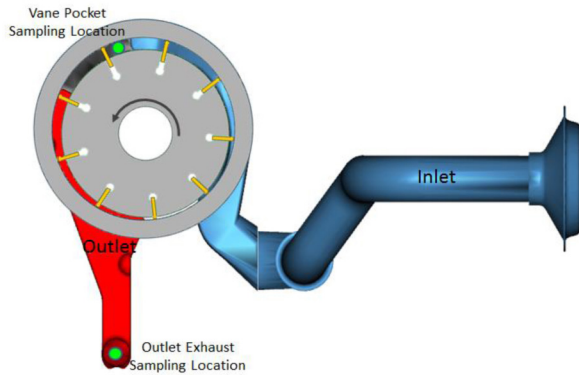


Figure 16. Flow domain used for study to compare 1D to 3D CFD.

The comparison of pressure pulsations at these two locations is shown in Figure 17. As can be seen, both the amplitude and frequency of the results are quite similar between GT-SUITE and PumpLinX. This is an illustration of the potential of using 1D CFD to capture a good estimate of the pressure pulsations predicted by 3D CFD at a fraction of the time. In this case, the 1D simulation took less than 10 minutes to run, while the 3D CFD simulation took around 5 hours to run, making 1D CFD roughly 30 times faster than 3D CFD.

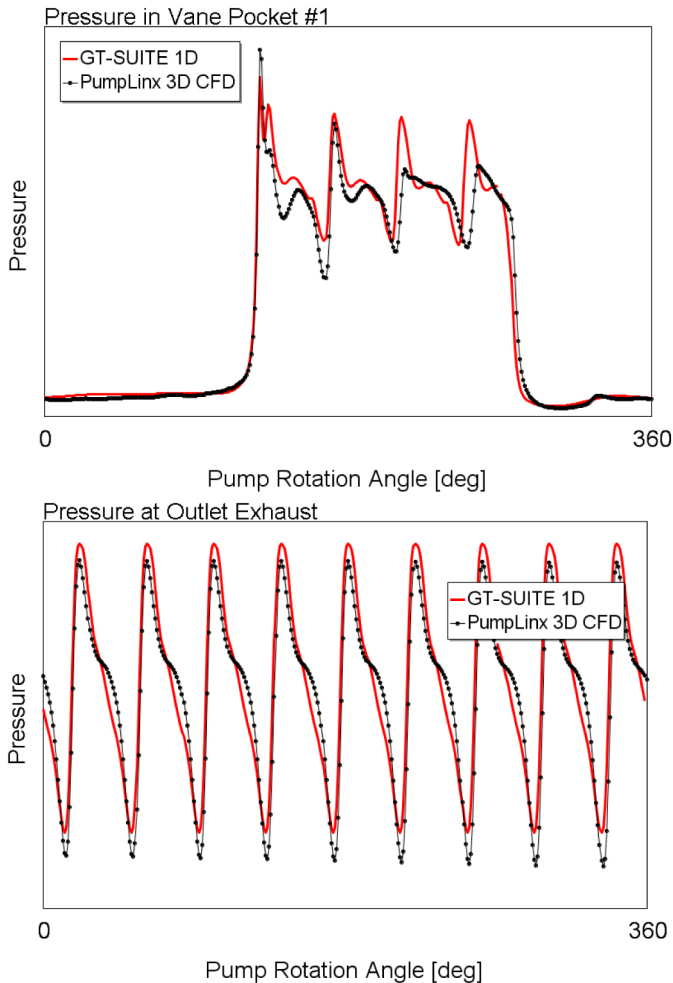


Figure 17. Pressure pulsation comparison between GT-SUITE and 3D CFD inside vane pocket and at outlet exhaust.

Pump Friction

Predicting friction in vane pumps is complex due to the number of interfaces where friction occurs. In the model there are ten components of friction considered as outlined earlier. When all ten components of friction are summed together, the total friction torque can be significant compared to the total torque required to drive the pump. In Figure 18, the breakdown of total, fluid and friction torque is shown, and though friction is a small contributor at low speed, it becomes more dominant than the fluid torque at high speed. By definition the total torque, T_{total} , is the sum of the fluid torque, T_{fluid} , and friction torque, $T_{friction}$, where the average fluid torque is calculated by summing up the total PdV work done by each chamber and integrating over one cycle:

$$T_{fluid} = \frac{\int_0^{360^\circ} \left(\sum_{i=1}^{\#ofChambers} p_i dV_i \right)}{360^\circ} \quad (3)$$

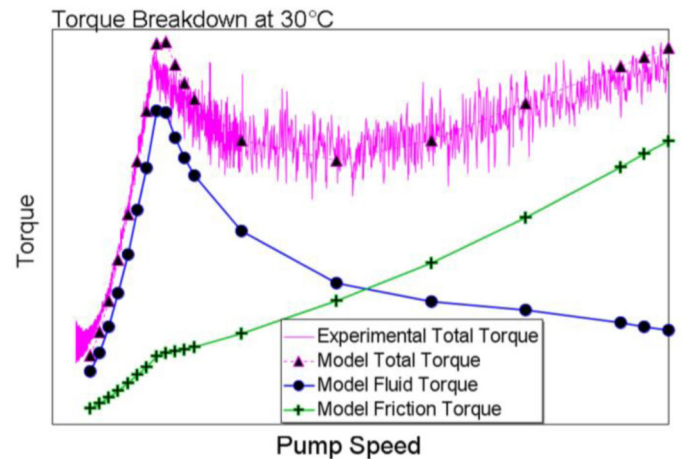


Figure 18. Breakdown of fluid torque, friction torque and total torque at 30°C.

A breakdown of the ten different components of friction power loss at two temperatures is shown in Figure 19. By far the most dominant component is the dry friction between the vane tip and outer stator. At the highest speed, this dry vane tip friction comprises 61% and 96% of total friction loss at 30°C and 140°C, respectively. At low temperature, viscous friction becomes more important, but at high temperature, only the vane tip friction is really important.

Even at low temperature, several of the friction components are nearly negligible including dry friction at the seals, dry friction at the pivot, and viscous friction between the outer stator and housing. The reason these terms are so small is due to the very low angular velocity of the moving stator. The viscous friction between the vane and slot is also negligible due to the low relative velocity between the vane and the slot. A thorough description of each friction component is given in the Appendix.

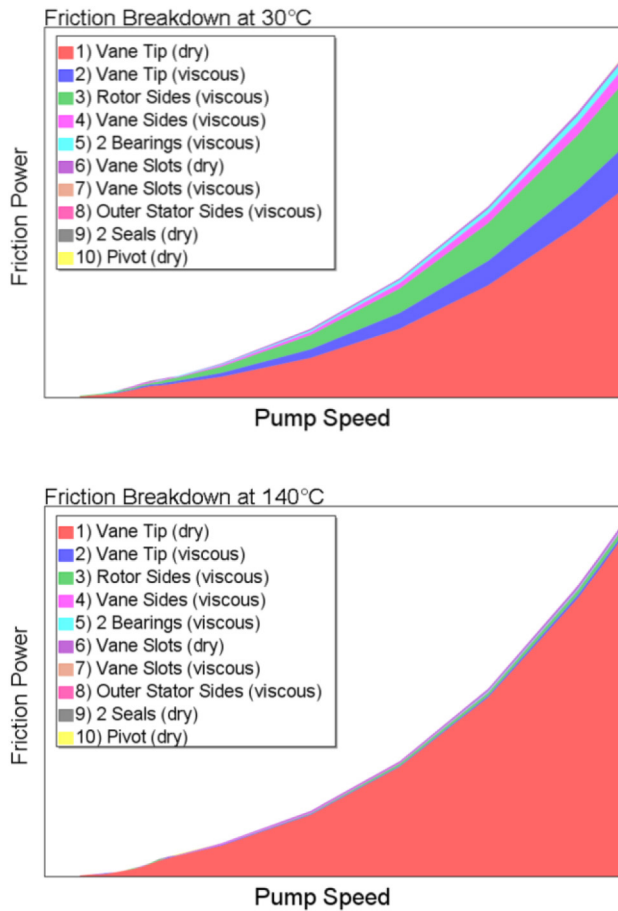


Figure 19. Breakdown of friction power loss by component at 30°C and 140°C. The components are stacked in the order listed in the legend.

The dry friction coefficient assumed at the vane tip is 0.12 across all speeds and temperatures, and coincides nicely with the friction coefficient at low speed obtained from experiments by Inaguma [8].

It should be noted Inaguma only considered the friction coefficient as a function of the pressure acting on the vane, and did not consider the centrifugal force of the vane. The friction coefficient, $C_{f,tip}$, in Inaguma goes as:

$$C_{f,tip} = \frac{T}{N_V R_V W_V T_V \Delta p_{rotor}} \quad (4)$$

where T represents the friction torque due to the vane tips, N_V represents the number of vanes, R_V represents the radius of the vane tip from the rotor center, W_V represents the width of the vane, T_V represents the thickness of the vane, and Δp_{rotor} represents the pressure difference between the pressure acting on the bottom of the vane and the inlet pressure of the pump. The GT-SUITE model for the friction coefficient is both a function of the pressure acting on the vane and the centrifugal force, and is given by:

$$C_{f,tip} = \frac{T}{N_V R_V \left(W_V T_V \Delta p_{rotor} + M_V \omega^2 \left(R_V - \frac{L_V}{2} \right) \right)} \quad (5)$$

where M_V represents the vane mass, ω represents the pump speed, and L_V represents the length of the vane. At zero speed, the GT-SUITE model is equivalent to Inaguma's model. Thus, the friction coefficient of 0.12 obtained by Inaguma at low speed is the appropriate value to use in the GT-SUITE model.

In Inaguma's model, the vane was radially balanced by pressure at the tip and the bottom when the vane was pointed in the direction of the pump outlet. This effectively means there is no friction contribution due to pressure acting on the vane while the vane traverses the pump outlet. Furthermore, in Inaguma's experiment, the pressure on the bottom of the vane was equal to the outlet pressure. The same principle is true in the GT-SUITE model, the only difference is the pressure acting on the bottom of the rotor is lower than the pump outlet pressure, but greater than the pump inlet pressure.

Conclusion

The aim of this paper is to showcase the ability of a 1D flow simulation to predict variable displacement vane pump performance, pulsations and friction. Several major conclusions can be drawn, including:

1. Pump performance comprising of average flow rate, pressure rise and torque can be predicted in an accurate manner using GT-SUITE 1D software.
2. A key component to capturing the correct flow rate behavior of the pump is to ensure the internal pressure forces acting on the stator are accurate.
3. Aeration/cavitation and temperature can significantly affect the internal pressure forces, which affect the outer stator eccentricity position and ultimately the flow rate.
4. Pressure pulsations can be predicted reasonably well in 1D with GT-SUITE compared to 3D CFD with PumpLinx at a fraction of the time (1D roughly 30x faster than 3D CFD).
5. A predictive friction model is proposed containing ten components of friction, and the model compares well to experimental torque of a GM pump.
6. The most dominant friction component by far is the dry friction between the vane tip and outer stator.
7. At low temperature, viscous friction is important, but at high temperature viscous friction is not important due to the low viscosity.

References

1. Mancò, S., Nervegna, N., Rundo, M., and Armenio, G., "Displacement vs Flow Control in IC Engines Lubricating Pumps," SAE Technical Paper [2004-01-1602](#), 2004, doi:[10.4271/2004-01-1602](#).
2. Mancò, S., Nervegna, N., Rundo, M., and Armenio, G., "Modelling and Simulation of Variable Displacement Vane

Pumps for IC Engine Lubrication,” SAE Technical Paper 2004-01-1601, 2004, doi:[10.4271/2004-01-1601](https://doi.org/10.4271/2004-01-1601).

3. Rundo, M. and Nervegna, N., “Geometry Assessment of Variable Displacement Vane Pumps”, ASME Transactions, Vol. 129, pp.446-455, 2007, doi:[10.1115/1.2718245](https://doi.org/10.1115/1.2718245).
4. Arata, T., Novi, N., Ariga, K., Yamashita, A., and Armenio, G., “Development of a Two-Stage Variable Displacement Vane Oil Pump,” SAE Technical Paper 2012-01-0408, doi:[10.4271/2012-01-0408](https://doi.org/10.4271/2012-01-0408).
5. Geist, B. and Resh, W., “Dynamic Modeling of a Variable Displacement Vane Pump Within an Engine Oil Circuit”, ASME Proceedings Paper ICEF2011-60039, 2011, doi:[10.1115/ICEF2011-60039](https://doi.org/10.1115/ICEF2011-60039).
6. Wang, D.M., Ding, H., Jiang, Y., and Xiang, X., “Numerical Modeling of Vane Oil Pump with Variable Displacement,” SAE Technical Paper 2012-01-0637, 2012, doi:[10.4271/2012-01-0637](https://doi.org/10.4271/2012-01-0637).
7. Sullivan, P.E. and Sehmy, M., “Internal Force Analysis of a Variable Displacement Vane Pump,” SAE Technical Paper 2012-01-0409, doi:[10.4271/2012-01-0409](https://doi.org/10.4271/2012-01-0409).
8. Inaguma, Y. “Theoretical Analysis of Mechanical Efficiency in Vane Pump”, JTEKT Engineering Journal English Edition No. 1007 E, p. 28-35, 2010.
9. Munson, B., Young, D., and Okiishi, T., Fundamentals of Fluid Mechanics, John Wiley & Sons, New York, 4th edition, pg. 513, 2002.

Contact Information

Jon Harrison
j.harrison@gtisoft.com

Rodrigo Aihara
r.aihara@gtisoft.com

Moji Eshraghi
mojtaba.eshraghi@gm.com

Irina Dmitrieva
irina.dmitrieva@gm.com

Acknowledgments

The authors would like to thank GM for permission to publish material related to their engine lubrication pump.

Definitions/Abbreviations

VDVP - Variable displacement vane pump

Q - Volume flow rate

Δp - Pressure difference

C_d - Orifice discharge coefficient

D_{orif} - Filter orifice diameter

ρ - Oil density

β - Ratio of upstream pipe diameter to filter orifice diameter

μ - Dynamic viscosity

c - Clearance gap

L - Length of leakage path

W - Width of leakage path

p_i - Static pressure in vane chamber number i

dV_i - Derivative of volume in chamber i

t_{cycle} - Time elapsed during 1 revolution of the pump

T_{fluid} - Fluid torque of the pump

ω - Pump rotational speed

Δp_{rotor} - Pressure difference between the pressure acting on the bottom of the vane and the pump inlet pressure

W_v - Chamber width (vane width)

T_v - Vane thickness

M_v - Vane mass

L_v - Vane length

R_v - Instantaneous distance between vane tip and rotor center

e - Pump eccentricity

D_s - Outer stator diameter

θ_v - Instantaneous vane angle

θ_G - Instantaneous global driver angle

N_v - Number of vanes

θ_e - Pump eccentricity angle

T - Torque

$C_{f,tip}$ - Dry friction coefficient of vane tip to stator contact

c_{tip} - Clearance between vane tip and stator

F_f - Friction force

A - Surface area for friction

u - Velocity of friction surface

c_{rotor} - Clearance between rotor and housing

R_{rotor} - Mean radius of friction surface of rotor with respect to rotor center

$D_{R,O}$ - Rotor outer diameter

$D_{R,I}$ - Rotor inner diameter

c_{vane} - Clearance between vane side and housing

$\Delta R_v / \Delta t$ - Change in vane tip radius with respect to time (axial velocity of vane in rotor slot)

W_R - Rotor inner width

c_{slot} - Clearance between vane side and slot

$C_{f,slot}$ - Dry friction coefficient of vane to slot contact

p1 - Pressure in chamber volume #1

p2 - Pressure in chamber volume #2

$D_{b,i}$ - Diameter of bearing # i

$W_{b,i}$ - Width of bearing # i

$c_{b,i}$ - Radial clearance of bearing # i

R_{stator} - Mean radius for torque between outer stator center of gravity and pivot point

ω_{stator} - Angular velocity of the outer stator about the pivot point

c_{stator} - Clearance between side of outer stator and pump housing

A_{stator} - Surface area of the film between the outer stator and pump housing, (includes both sides of the stator)

$C_{f,seal,i}$ - Dry friction coefficient between seal i and pump

housing

$F_{seal,i}$ - Preload force of seal i

$R_{seal,i}$ - Radius between seal i and pivot point

$C_{f,pivot}$ - Dry friction coefficient between outer stator and pivot

F_{pivot} - Normal load transmitted from outer stator on pivot

R_{pivot} - Radius between pivot point and contact point of outer stator

APPENDIX

Different Components of Friction for Vane Pump

There are ten different components of friction considered for the total friction loss in the vane pump. The governing equations for each component are described below. The schematic in [Figure 20](#) highlights important dimensions to consider for the various friction components.

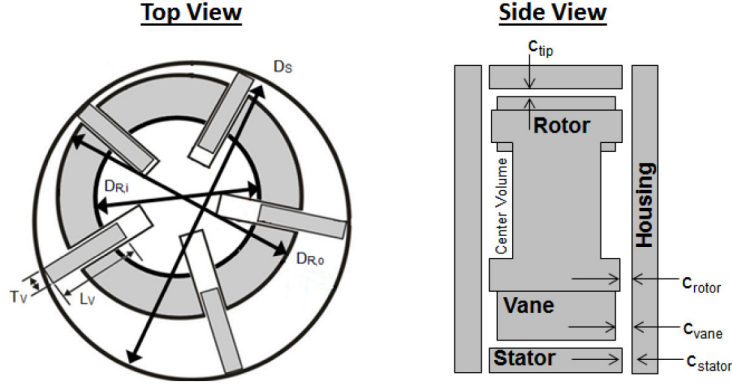


Figure 20. Schematic of important dimensions to consider for the various friction components.

1. Dry Friction between Vane Tip and Stator

The first and most dominant component of friction deals with the dry friction power loss generated between the vane tip and the stator, P_f . The vane tip has two components of force acting on the vane as shown in [Figure 21](#) including the force acting on the vane from the pressure in the volume at the center of the rotor, F_p , as well as the centrifugal force of the vane mass, F_c .

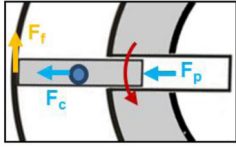


Figure 21. Schematic of forces contributing to dry friction between the vane tip and outer stator.

$$F_p = \Delta p_{rotor} W_V T_V \quad (6)$$

and

$$F_c = M_V \omega^2 \left(R_V - \frac{L_V}{2} \right) \quad (7)$$

$$R_V = -e \cos(\theta_V) + \sqrt{\left(\frac{D_s}{2}\right)^2 + e^2 [\cos(\theta_V)^2 - 1]} \quad (8)$$

and

$$\theta_V = \theta_G - \frac{360}{2N_V} - \theta_e \quad (9)$$

$$P_1 = T\omega = N_V C_{f,tip} (F_p + F_c) R_V \omega \quad (10)$$

2. Viscous Friction between Vane Tip and Stator

The second component of friction deals with the viscous friction power loss generated between the vane tip and the stator, P_2 . This term assumes there is a film across the entire thickness of the vane, as shown in [Figure 22](#).

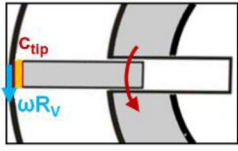


Figure 22. Schematic of thin film contributing to viscous friction between vane tip and outer stator.

The basic governing equation for the viscous friction force, F_f , in a thin film goes as:

$$F_f = \mu A \frac{du}{dy} \quad (11)$$

The corresponding viscous friction power loss between the vane tip and stator follows as:

$$P_2 = T\omega = F_f R_v \omega = \frac{N_v \mu W_v T_v R_v^2 \omega^2}{c_{tip}} \quad (12)$$

3. Viscous Friction between Rotor Sides and Housing

The third component of friction deals with the viscous friction power loss generated between the two sides of the rotor and the pump housing, P_3 , as exhibited by the face highlighted in [Figure 23](#).



Figure 23. Schematic of thin film contributing to viscous friction between rotor sides and housing.

The viscous friction force, F_f , in the gap between the rotor side and housing goes as:

$$F_f = \mu A \frac{du}{dy} = \mu A \frac{\omega R_{rotor}}{c_{rotor}} \quad (13)$$

where

$$A = \frac{\pi(D_{R,o}^2 - D_{R,i}^2)}{4} \quad (14)$$

and

$$R_{rotor} = \frac{(D_{R,o} + D_{R,i})}{4} \quad (15)$$

The corresponding viscous friction power loss between the rotor side and pump housing follows as below. Notice the multiplier of two represents the friction from both rotor sides.

$$P_3 = F_f R_{rotor} \omega = 2\mu \frac{\pi(D_{R,o}^2 - D_{R,i}^2)}{4} \frac{\omega^2 R_{rotor}^2}{c_{rotor}} \quad (16)$$

4. Viscous Friction between Vane Sides and Housing

The fourth component of friction deals with the viscous friction power loss generated between the two sides of the vane and the pump housing, P_4 , as exhibited by the faces highlighted in [Figure 24](#).



Figure 24. Schematic of thin film contributing to viscous friction between vane sides and housing.

The viscous friction force, F_f , in the gap between the vane side and housing goes as:

$$F_f = \mu A \frac{du}{dy} = \mu A \frac{\omega \left(R_V - \frac{L_V}{2} \right)}{c_{vane}} \quad (17)$$

where

$$A = N_V 2 L_V T_V \quad (18)$$

The corresponding viscous friction power loss between the vane side and pump housing follows as below. Notice the multiplier of two represents the friction from both vane sides.

$$P_4 = F_f \left(R_V - \frac{L_V}{2} \right) \omega = \mu N_V 2 L_V T_V \frac{\omega^2 \left(R_V - \frac{L_V}{2} \right)^2}{c_{vane}} \quad (19)$$

5. Viscous Friction in Journal Bearings

The fifth component of friction deals with the viscous shear power loss generated in the two journal bearings supporting the shaft of the pump, P_5 . For simplicity, the bearings are assumed to be lightly loaded and not highly eccentric.

$$P_5 = T \omega = \sum_i^{\#bearings} F_{f,i} \frac{D_{b,i}}{2} \omega = \sum_i^{\#bearings} \mu A_i \left(\frac{du}{dy} \right)_i \frac{D_{b,i}}{2} \omega = \sum_i^{\#bearings} \frac{\mu 2 \pi W_{b,i} \left(\frac{D_{b,i}}{2} \right)^3 \omega^2}{c_{b,i}} \quad (20)$$

6. Dry Friction between Vane Side and Slot

The sixth component of friction deals with the dry friction power loss generated between the two sides of the vane and the slot of the rotor, P_6 , as exhibited by the faces highlighted in [Figure 25](#).

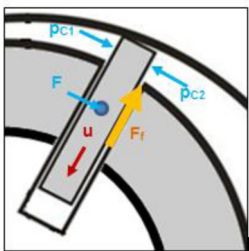


Figure 25. Schematic of dry friction between vane side and slot.

The dry friction force, F_f , in the gap between the vane side and slot of the rotor goes as:

$$F_f = C_{f,slot} \Delta p A = C_{f,slot} |p_1 - p_2| W_V \left(R_V - \frac{D_{R,o}}{2} \right) \quad (21)$$

The corresponding dry friction power loss between the vane side and slot of the rotor follows as below.

$$P_6 = F_f u = F_f \frac{\Delta R_V}{\Delta t} = N_V C_{f,slot} |p_1 - p_2| W_V \left(R_V - \frac{D_{R,o}}{2} \right) \left(\frac{\Delta R_V}{\Delta t} \right) \quad (22)$$

7. Viscous Friction between Vane Side and Slot

The seventh component of friction deals with the viscous friction power loss generated between the two sides of the vane and the slot of the rotor, P_7 , as exhibited by the faces highlighted in [Figure 26](#).

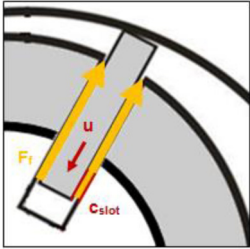


Figure 26. Schematic of viscous friction between vane side and slot.

The viscous friction force, F_f , in the gap between the vane side and slot of the rotor goes as:

$$F_f = \mu A \frac{du}{dy} = \mu A \frac{\Delta R_V}{\Delta t c_{slot}} \quad (23)$$

where:

$$A = N_V 2 W_V \left[\frac{D_{R,o}}{2} - (R_V - L_V) \right] \text{ if } \frac{D_{R,o}}{2} - (R_V - L_V) < \frac{D_{R,o}}{2} - \frac{D_{R,i}}{2}$$

else

$$A = N_V 2 \left\{ W_V \left[\frac{D_{R,o}}{2} - \frac{D_{R,i}}{2} \right] + W_R \left[\frac{D_{R,i}}{2} - (R_V - L_V) \right] \right\} \quad (24)$$

The corresponding viscous friction power loss between the vane side and slot of the rotor follows as below. Notice the multiplier of two represents the friction from both vane sides.

$$P_7 = F_f u = F_f \frac{\Delta R_V}{\Delta t} = \frac{\mu A \left(\frac{\Delta R_V}{\Delta t} \right)^2}{c_{slot}} \quad (25)$$

8. Viscous Friction between Outer Stator and Housing

The eighth component of friction deals with the viscous shear power loss generated between the outer stator side and the pump housing, P_8 .

$$P_8 = T \omega = \frac{\mu A_{stator} R_{stator}^2 \omega_{stator}^2}{c_{stator}} \quad (26)$$

9. Dry Friction between Seals and Housing

The ninth component of friction deals with the dry friction power loss generated between any seals and pump housing, P_9 . Typically there are seals to seal off any leakage between a control chamber and other chambers within the pump. These seals tend to be preloaded like a spring and exhibit a normal force between the stator and housing, which causes dry friction to occur. A preload force, F_{seal} , of 20N was assumed for the two seals in the GM pump, though the preload can be as high as several hundred Newtons. Additionally, the dry friction coefficient, $C_{f,seal}$, was assumed to be 0.1.

$$P_9 = T\omega = \sum_i^{\#seals} C_{f,seal,i} F_{seal,i} R_{seal,i} \omega_{stator} \quad (27)$$

10. Dry Friction between Outer Stator and Pivot

The tenth component of friction deals with the dry friction power loss generated at the contact between the outer stator and pivot, P_{10} . The dry friction coefficient, $C_{f,pivot}$, was assumed to be 0.1. Furthermore, the normal load acting on the pivot, F_{pivot} , is calculated based on the force balance of the outer stator.

$$P_{10} = T\omega = C_{f,pivot} F_{pivot} R_{pivot} \omega_{stator} \quad (28)$$

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. The process requires a minimum of three (3) reviews by industry experts.

All rights reserved. No part of this publication may be reproduced, stored in a retrieval system, or transmitted, in any form or by any means, electronic, mechanical, photocopying, recording, or otherwise, without the prior written permission of SAE International.

Positions and opinions advanced in this paper are those of the author(s) and not necessarily those of SAE International. The author is solely responsible for the content of the paper.

ISSN 0148-7191

<http://papers.sae.org/2014-01-1086>