Development of a Gas Dynamic and Thermodynamic Simulation Model of the Lontra Blade Compressor™

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Abstract. The Lontra Blade Compressor™ is a patented double acting, internally compressing, positive displacement rotary compressor of innovative design. The Blade Compressor is in production for waste-water treatment, and will soon be launched for a range of applications at higher pressure ratios.

In order to aid the design and development process, a thermodynamic and gas dynamic simulation program has been written in house. The software has been successfully used to optimise geometries and running conditions of current designs, and is also being used to evaluate future designs for different applications and markets.

The simulation code has three main elements. A positive displacement chamber model, a leakage model and a gas dynamic model to simulate gas flow through ports and to track pressure waves in the inlet and outlet pipes. All three of these models are interlinked in order to track mass and energy flows within the system.

A correlation study has been carried out to verify the software. The main correlation markers used were mass flow, chamber pressure, pressure wave tracking in the outlet pipe, and volumetric efficiency. It will be shown that excellent correlation has been achieved between measured and simulated data. Mass flow predictions were to within 2% of measured data, and the timings and magnitudes of all major gas dynamic effects were well replicated.

The simulation will be further developed in the near future to help with the optimisation of exhaust and inlet silencers.

1. Introduction

After the filing of an initial patent, Lontra was founded in 2004 by Steve Lindsey, the inventor of the blade compressor. After favourable concept and market studies carried out at Southampton University, Imperial College, and Cosworth Technology, it became apparent that in order to evaluate the design further, it would be necessary to develop a parametric model of the compressor. At first this took the form of a simple geometry model driven by a small number of basic dimensions, which quickly enabled

the calculation of chamber volume with shaft angle. This was useful for estimating the necessary speed and size of machine for a given application. The geometry model was a springboard for the development of a basic time marching thermodynamic model using simple port geometry with isentropic port flow, and calculation of chamber properties by using conservation of mass and energy. This gave useful estimates of flow, work and temperature that could be modified by imposing assumed values of volumetric and isentropic efficiency.

It became obvious that the next refinement should be the inclusion of a leakage model. To date 21 separately modelled leak paths have been identified. The leakage calculations are based on a method for calculating viscid flow of air in a narrow slot set out by Shires (1950). This method, which calculates the pressure drop down a slot due to viscous drag, is combined with an isentropic nozzle calculation which finds the pressure drop needed to accelerate the fluid to the speed at the entry of the leak path. The two are solved simultaneously to find the pressure at the start of the slot.

Lontra is constantly improving its testing and instrumentation capabilities. Pipe pressure measurements using high speed transducers have shown that there were measurable pulsations in the ducting of the compressor, and that the chamber pressure was being influenced by wave effects in the outlet pipe. These waves if timed correctly, can be used to increase efficiency, but if timed badly, can have a detrimental effect. This effect was simulated using a one dimensional unsteady gas dynamic simulation, a technique which is used extensively in the automotive industry to tune engine manifolds. The method used by Lontra is that set out by Blair (1999), but also uses some elements of the non homentropic method of characteristics described by Benson (1982). Comparison of measured and predicted pressure traces have shown that the timing and magnitude of the major pressure fluctuations are predicted well. It is planned that this technique will be used in the future to help with the design of silencers.

The simulation has been used successfully to optimize the geometry and port timing of a 1.5 bar compressor with a displacement of 11 litres running at 3000 rpm. This machine has performed very successfully in site trials at Severn Trent's Worcester sewage works where it is used to aerate sewage sludge. It has shown to be up to 20% more efficient than the existing compressors in this application.

2. Core design

The basic design uses a ring shaped chamber with a continually open inlet port and an outlet port valved by the action of the rotor. The principal components are a bladed rotor which rotates within a housing, and a rotating disc. The blade is shaped so that it can pass through a slot in the disc (fig.1a). The machine is double acting, drawing in gas behind the blade and compressing it ahead of the blade. The compression chamber is formed as the blade, rotating within the housing, works against the disc (fig 1b). The compressed gas exits though a port in the housing (fig 1c). This simple, elegant design creates a basic compressor. An animation of the design is available at www.lontra.co.uk

2.1 Geometry

The geometry of the machine is derived from 5 key parameters. In order to quickly evaluate different sizes and machine proportions, volumes and surface areas, including port areas are generally calculated analytically using double and triple integrals within the simulation code. This information is stored in arrays, and during the running of the simulation, angle based geometry data is found using interpolation of the array data. Unusual or complicated geometries can also be catered for by entering CAD derived angle based data as an input to the program.

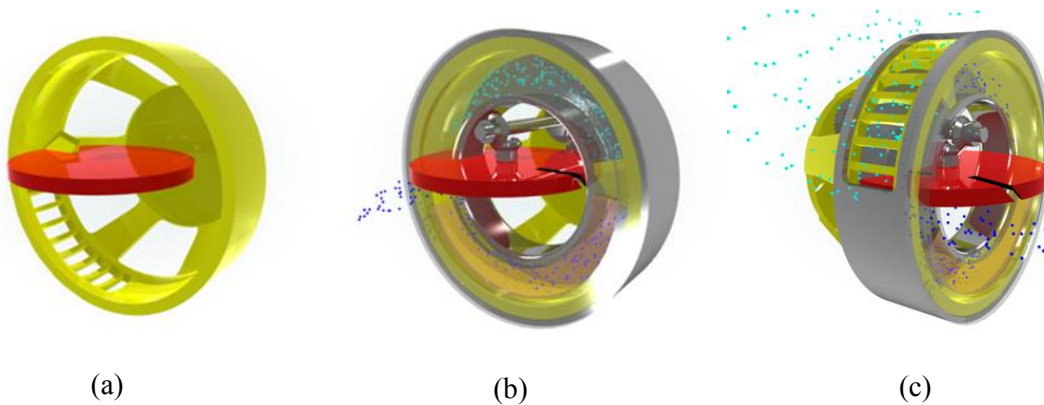| (a) | (b) | (c) |

Fig. 1 (a) the blade can pass through a slot in the disk, (b) anticlockwise rotation sucks in air behind the blade and compresses it in front of the blade, and (c) air exits through a port in the rotor.

3. Model of the working chamber

The working chamber is modelled using principles of conservation of mass and conservation of energy. The assumption is made that the chamber properties only vary with time and do not vary with position. This is commonly known as a zero dimensional model.

Conservation of mass over a time step dt

$$m_{t+dt} = m_t + dm_{in} - dm_{out} \qquad (1)$$

Conservation of energy over a time step dt

$$U_{t+\mathrm{dt}} = U_t + dH_{in} - dH_{out} + Q - W \qquad (2)$$

where

$$W = \frac{(p_{t+dt} + p_t)}{2}(v_{t+dt} - v_t) \qquad (3)$$

The method of calculating chamber properties at the end of a time step is to calculate the specific internal energy from knowledge of mass and internal energy at the end of a time step. Temperature can then be found directly from internal energy, and the pressure calculated using the ideal gas equation. It can be seen that the work term uses values of pressure from the beginning and end of a time step. This means that the calculation of properties at the end of a time step needs to be found iteratively, by repeating the calculation procedure until values of P and T are found to within acceptable limits.

In equations 1 and 2 mass and enthalpy flows are due to leakage flows which will be described in section 2, and port flows, which will be described in section 3.

Heat transfer within the chamber is accounted for using the Dittus Boelter relationship, where

$$Nu = 0.023 Re^{0.8} Pr^{0.4} \tag{4}$$

This formula is meant for fully developed turbulent flow in pipes, but it is the closest that the author has found to the situation within the blade compressor. For calculation of the Reynolds number, the gas is assumed to be flowing at the average speed of the blade.

4. Leakage calculations

Like many other oil free compressors, the Lontra blade compressor relies on close running non contacting surfaces for sealing. The mass flow through a narrow slot increases with width and clearance and decreases with length. In most designs, the clearance seals are formed by non conforming surfaces, which results in a short effective length of clearance seal (Fig. 2a). However the major clearance seals in the blade compressor are made of conforming surfaces that allows for seals that are much longer and offer increased resistance to flow (Fig. 2b).

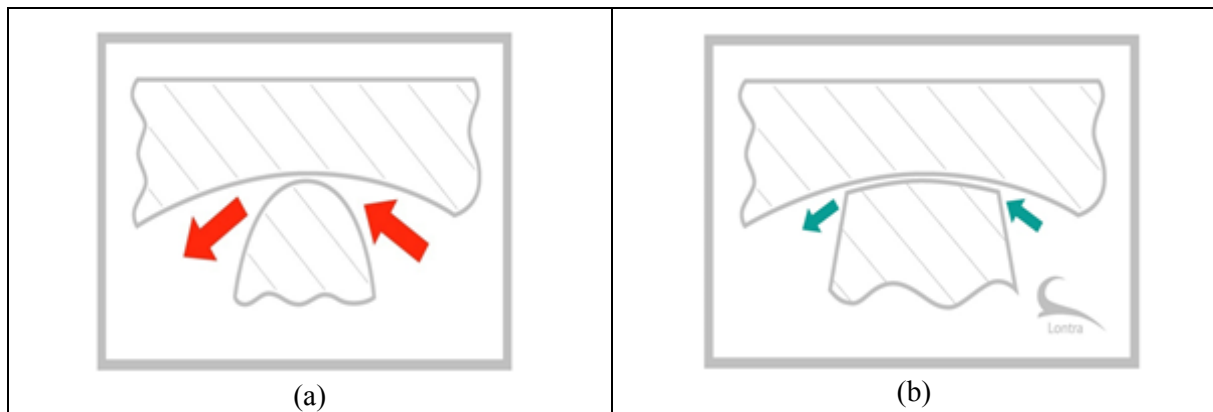|	
(a)	(b)

Fig 2 (a) Non conforming seals have a short effective seal length, (b) conforming surfaces offer a greater seal length.

Looking at figure 3, for steady flow, with force on the element acting to the right, the balance of forces on a fluid element is described by equation 5.

$$-\frac{\partial p}{\partial x} = -\frac{\partial s}{\partial y} + \frac{\partial(\rho u^2)}{\partial x} \tag{5}$$

Or by using the definition of viscosity $s = \mu \frac{\partial u}{\partial y}$

$$-\frac{\partial p}{\partial x} = -\mu \frac{\partial^2 u}{\partial y^2} + \frac{\partial(\rho u^2)}{\partial x} \tag{6}$$

$$s + \frac{\partial s}{\partial y}\delta y$$

$$p + \frac{\partial p}{\partial x}\delta x$$

$$u + \frac{\partial u}{\partial x}\delta x$$

$$\rho + \frac{\partial \rho}{\partial x}\delta$$

p

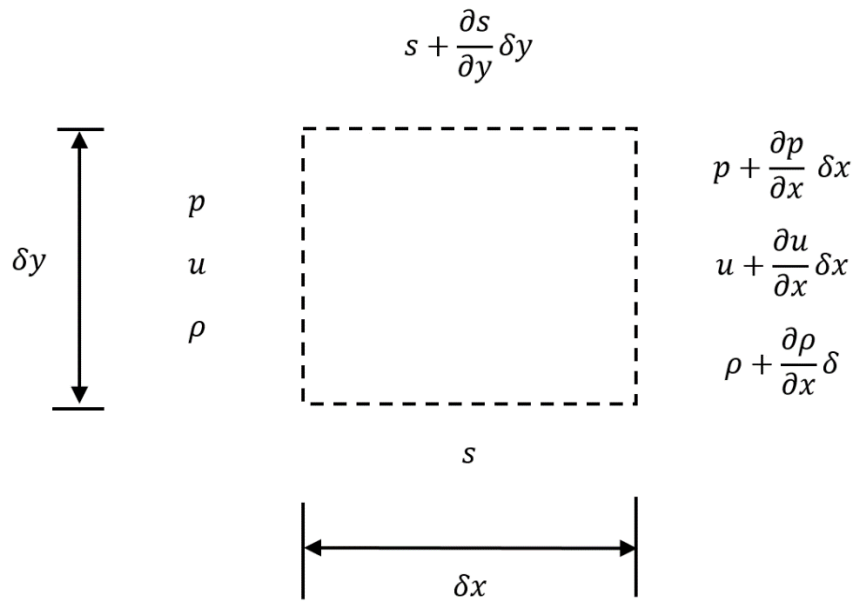δy u

ρ

s

δx

Fig 3. Volume of fluid

The term on the left hand side of equation 6 is the force due to the pressure gradient, the first term on the right is the force due to viscous shear, and the second term is the change in momentum.   The methodology used to calculate mass flow through the clearance seals is based on an approach proposed by Shires (1950) which has become a standard approach in the design of air bearings. It assumes that for laminar flow in a narrow slot, the viscous forces are far higher than the inertia forces. Therefore the momentum term can be neglected and the equation simplifies to equation 7.

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial y^2} \qquad (7)$$

Integrating between limits and rearranging yields

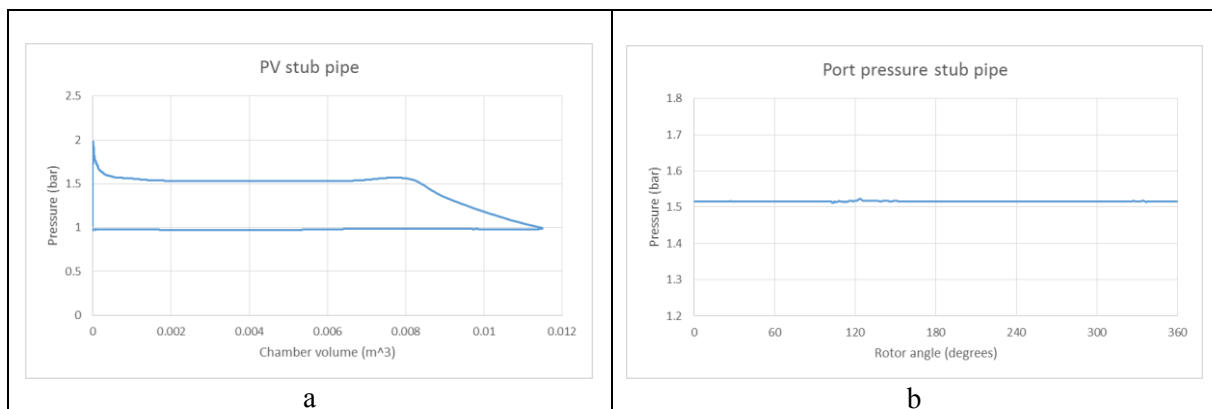$$p_1{}^2 - p_2{}^2 = \frac{24\mu RT\dot{m}}{bc^3}(x_2 - x_1) \qquad (8)$$

An extension to this theory, suggested by Shires, that uses the concept of a resistance coefficient to extend the useful range of application of this approach to include turbulent flow has been included. The unknowns are p_1, the pressure at the entry of the slot, and mass flow. The length of the slot is ($x_2 - x_1$), and p_2 is the pressure at the exit of the slot which is assumed to be at the pressure of the reservoir that gas is exiting into. Equation 8 has been combined with equation 9 for an isentropic nozzle, which is used to model the pressure drop needed to accelerate the fluid from rest in the upstream reservoir (subscript 0) to the speed of the fluid at the start of the slot. The two are solved simultaneously to find a value of p1 for which equations 8 and 9 give the same mass flow rate. The speed of the gas at the slot exit is monitored, and if this exceeds the speed of sound, then the routine is modified so that the speed of the gas at the slot exit is limited to the speed of sound.

$$\dot{m} = A\rho_0 \sqrt{\left\{2\left(\frac{\gamma}{\gamma-1}\right)\frac{p_0}{\rho_0}\left(\frac{p_1}{p_0}\right)^{2/\gamma}\left(1-\left(\frac{p_1}{p_0}\right)^{((\gamma-1)/\gamma)}\right)\right\}} \tag{9}$$

5. Gas dynamic calculations

Computer simulation of pressure wave propagation in ducts is a science that is over 40 years old. The technique is used extensively in the automotive industry to simulate unsteady flow in internal combustion engine ducting to help realise the benefits of pressure wave tuning. There are numerous books on the subject, the most comprehensive of which are by Benson (1982), Blair (1999), and Pearson and Winterbone (2000). There have also been a small number of researchers who have simulated unsteady gas dynamics in the ducts of positive displacement compressors for example (Stosic N. and Hanjalic K. (1978). In order to achieve accurate correlation, and also to investigate pressure wave tuning, it was decided that simulation of unsteady gas dynamics in the ducting of the blade compressor was important to Lontra

The routine used at Lontra is based on the method set out by Blair with elements of Benson's approach, and includes, plenums, junctions, tapered pipes, non isentropic inflow and outflow through ports, friction and heat transfer. The port flow coefficients chosen are those published by Blair for two stroke ports, as these are considered the most appropriate for the ports in the blade compressor. The model is object oriented, meaning that it is simple to join together any number of elements quickly and easily to produce the desired system. The details of the wave tracking method will not be given in this paper, as the three previous references give a very thorough explanation of the various techniques. Figure 4 shows simulated port and chamber pressure, with and without pressure wave tracking in the pipe system that is currently used on the test bed. It shows that gas dynamics accounts for significant pressure fluctuations in the outlet port and the working chamber, and that wave tracking is necessary for accurate correlation and simulation.

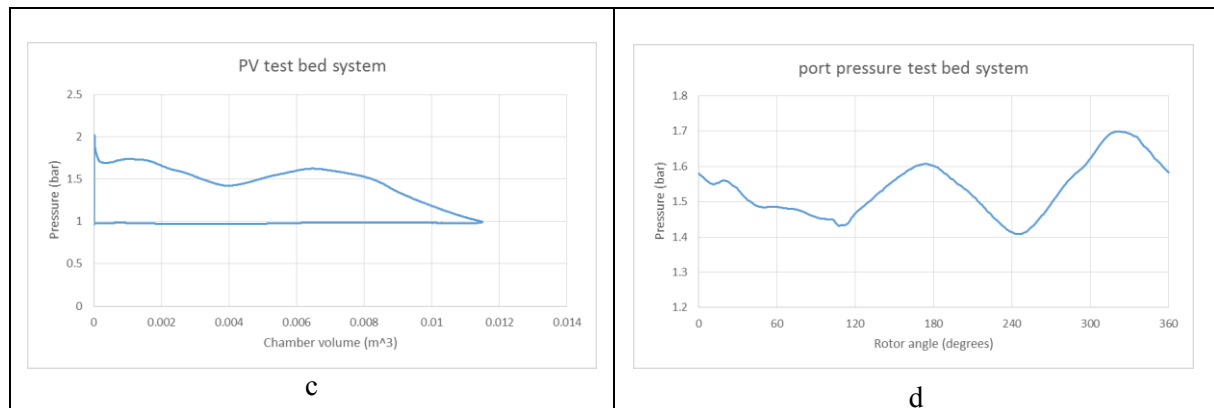|	
PV stub pipe	Port pressure stub pipe
a	b

Fig. 4. Simulated outputs at 0.5 bar outlet pressure, (a) PV diagram with no pipe, (b) port pressure with no pipe, (c) PV diagram with the test bed outlet system, (d) port pressure with the test bed outlet system

6. Test data and correlation.

6.1 The test rig

An instrumented test rig has been built to evaluate and develop the blade compressor design (figure. 5). The working chamber is instrumented with 10 Kulite high speed pressure transducers. Inlet and outlet pipe pressures are also measured with Kulite high speed pressure transducers of the appropriate pressure range, and temperatures are measured with PT100 class 1 PRTs. Torque is measured with an HBM T40 torque transducer. Angular position is determined using two encoder wheels in an offset configuration. Flow rate is measured using an orifice plate flow meter designed in house to ISO 5167. All data is acquired at 25 kHz, using an NI 9237data acquisition card and the test bed is controlled using bespoke software programmed in Labview. Knowledge of volume change with angle allows indicated work and power to be calculated directly from the pressure-volume history. Software has been written in house to regress the test data.
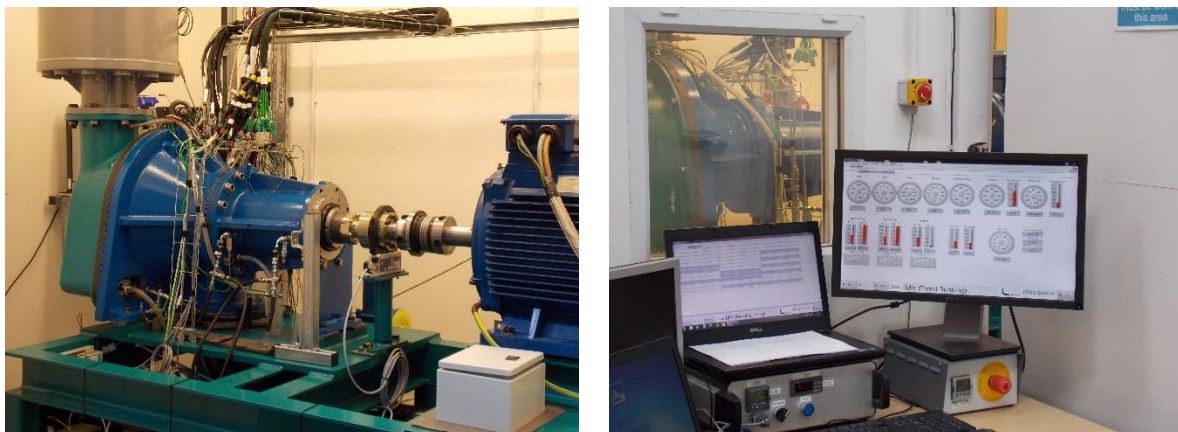

Figure 5. The Lontra test bed and user interface.

6.2 Correlation

In order to correlate the model over a wide range of operating conditions, measured data was acquired at 25 test points. The outlet pressure range chosen was from 0.2 to 0.6 bar in 0.1 bar increments and the speed range was from 1000rpm to 3000rpm in 500rpm increments. Geometric inputs into the simulation have been found either from physical measurements, or from CAD models. Calculated data such as volumes and surface areas have also been verified by CAD measurements. The dimensions that are most difficult to measure are the seal clearances. 21 separate clearance seals have been identified, some of which can be measured as the machine is being assembled, but others can only be estimated from machining tolerances. Comparisons of measured and simulated mass flow vs speed for the lowest and highest outlet pressures using the best estimates of clearances are shown in figure 6 and 7.
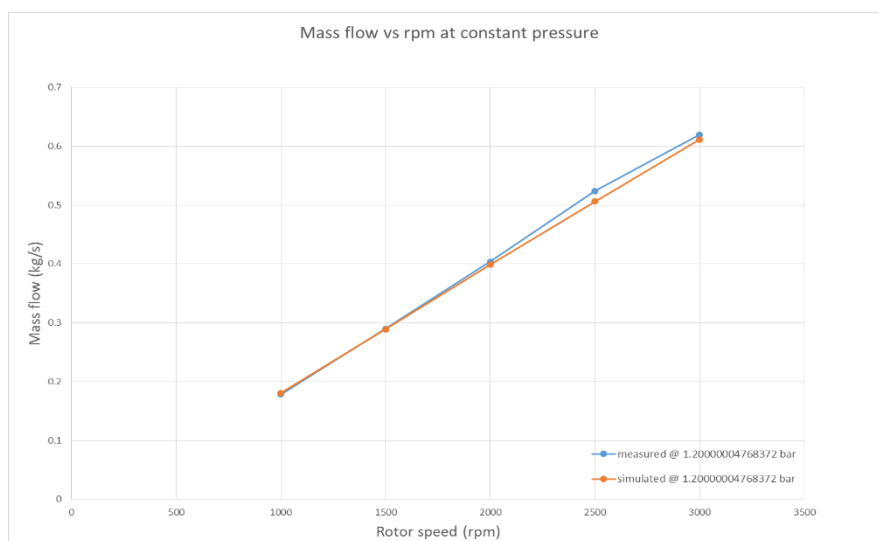

Figure 6. Predicted and simulated mass flow vs speed with a 0.2 bar outlet.
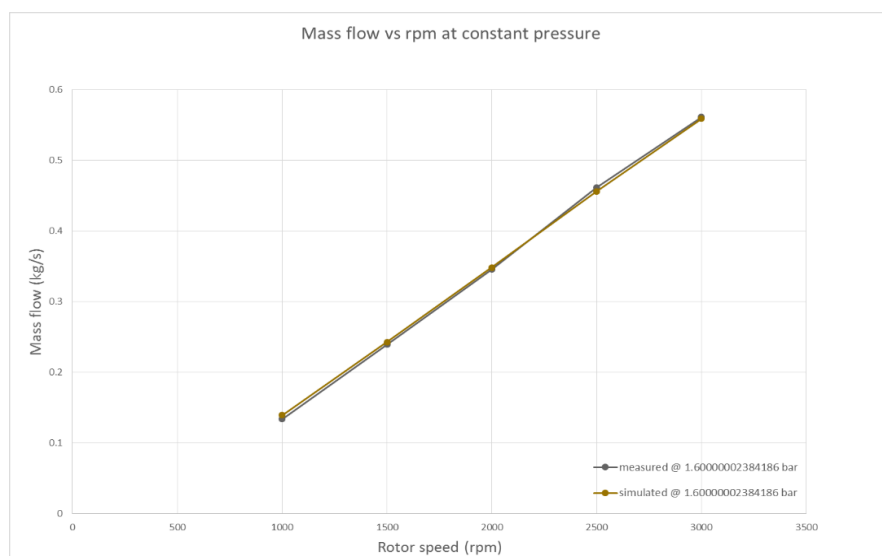

Figure 7. Predicted and simulated mass flow vs speed with a 0.6 bar outlet

Graphs at the intermediate pressure points show similar agreement between measured and predicted flow rates. The maximum error in mass flow prediction across all 25 test points was 4.2%, and the average mass flow error was 1.53%.

Graphs of predicted vs measured chamber pressure and outlet pipe pressures are shown in figures 8 and 9.

(a) 1000rpm   (b) 3000rpm
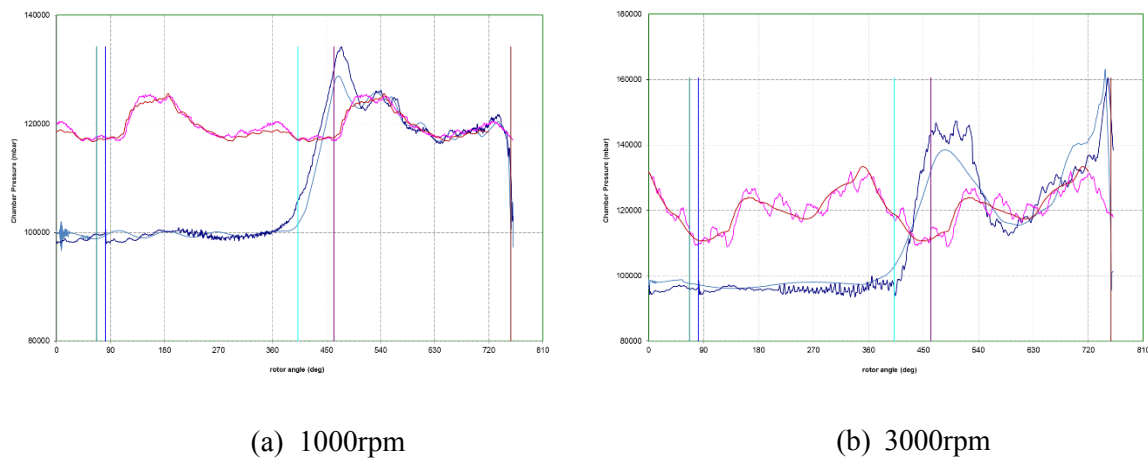
Figure 8. Predicted and measured pipe pressure (pink) and chamber pressure (blue) at 0.2 bar back pressure. Smoother line in both cases is the simulated result.

(a) 1000rpm   (b) 3000rpm
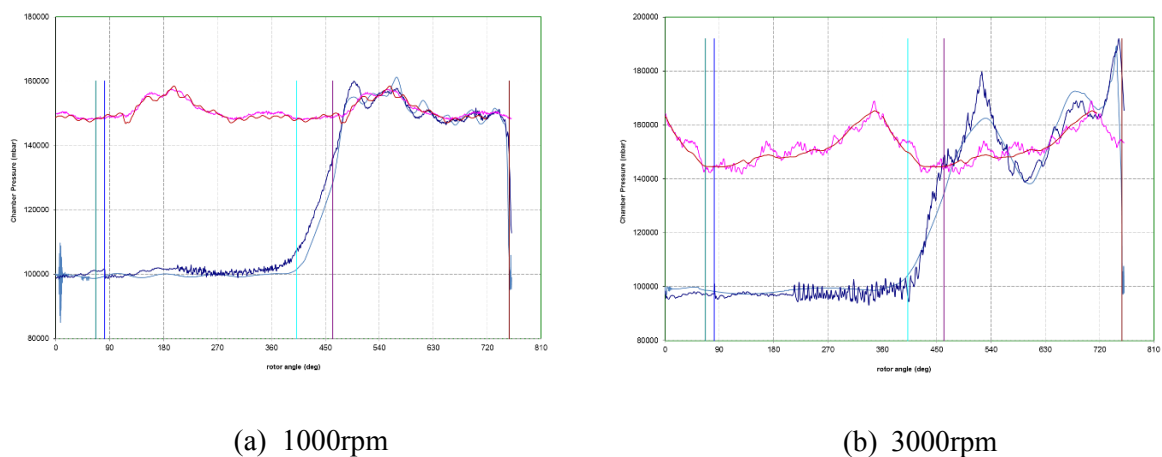
Figure 9. Predicted and measured pipe pressure (pink) and chamber pressure (blue) at 0.5 bar back pressure. Smoother line in both cases is the simulated result.

It can be seen that the gas dynamic simulation tracks the major oscillations in the outlet pipe and their effect on the chamber pressure. In some cases the simulation under predicts chamber pressure during compression and outlet opening, and this is thought to be due to spatial variation of pressure within the working chamber that the simulation cannot model. Lontra is also working on implementing silencer modelling which could help to improve the accuracy of the prediction of intake pressure waves.

7. Conclusions

The Lontra blade compressor is an innovative design of positive displacement compressor. As with all new designs, it is necessary to understand the operation of the device in detail in order to improve and develop the product effectively. The simulation software that has been developed at Lontra is able to provide insight into the physical processes involved during compressor operation and can effectively inform design decisions. It is also possible to accurately predict mass flow for a given set of clearances, and accurate pressure predictions allow trends in indicated torque and efficiency to be evaluated. With new products being developed at Lontra, the simulation software has become an invaluable tool in evaluating new concepts while at the same time speeding up the development process and saving valuable resources that would otherwise have to be spent on a greater number of physical prototypes. It has also allowed Lontra to help our licensees with "what if" simulations, to quickly assess the effect of production changes (casting and machining tolerances for example) on machine operation and efficiency forming a very valuable tool for production cost management.

Nomenclature

m = mass (kg)
U = internal energy (J)
H = Enthalpy (J)
Q = heat transfer (J)
W = work (J)
p = pressure (Pa)
v = volume
Nu = Nusselt number
Re = Reynolds number
Pr = Prandtl number
ρ = density (kg/m^3)

s = shear stress (Pa)
u = velocity (m/s)
μ = viscosity (Pa.s)
R = specific gas constant (J/kg.K)
T = temperature (K)
b = breadth of slot (m)
c = slot clearance (m)
γ = ratio of specific heats
A = area (m^2)
$\dot{m}$ = mass flow rate (kg/s)

References

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